SAE Journal

Published Monthly by The Society of Automotive Engineers, Inc.

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■ *Since 1938,* MARTIN A. ELLIOTT *has been employed as a gas engineer by the U. S. Bureau of Mines to study possible hazards that might attend the use of diesel engines underground. A 1930 Johns Hopkins graduate, he stayed on as instructor in gas engineering at the University, received his Ph.D. in 1933. Mr. Elliott went with the Consolidated Gas Electric Light and Power Co. of Baltimore, in 1934, as technical assistant to the superintendent of gas manufacture, spent four years there, and then stepped into his present job.*

■ R. M. GUERKE *received his M.S. in aeronautical engineering upon graduating from M.I.T., in 1938. He had majored in propeller vibration engineering, a subject that has been the keynote of his research career. Today, he is chief vibration engineer, Curtiss Propeller Division, Curtiss-Wright Corp., developing and testing both aluminum alloy and hollow-steel bladed, electrically-controlled "props."*

■ FRED E. WEICK *(M '29) took his present position as chief engineer, Engineering & Research Corp. early in 1937. Prior to that he helped in the development of the NACA cowling; cooperated with NACA engineers in designing experimental planes used by U. S. Department of Commerce as a part of its program on airplanes for private use.*

■ JAMES C. ZEDER *(M '23), chief engineer of Chrysler Corp., graduated from the University of Michigan a B.S. in 1922. The following year he went with Timken Roller Bearing Co. as service engineer. Two years later he joined the Maxwell-Chrysler Motor Corp. as experimental engineer. In 1929, he was named chief engineer of Plymouth and DeSoto Divisions; in 1931, assistant chief engineer of Chrysler Motors Corp.; and in 1937, advanced to his present position.*

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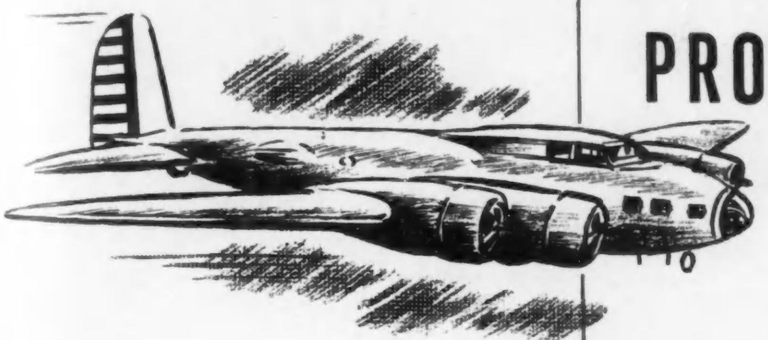

PROGRESS TRAIL for War and Peace Blazed at AIRCRAFT PRODUCTION MEETING

by Charles F. McReynolds

WITH aircraft production now concentrated almost exclusively in the military field, and with subcontractors from every branch of American industry cooperating to boost the production of fighting planes, there was an air of grim determination about the 1941 National Aircraft Production Meeting. Throughout the sessions one sensed the fact that engineers and executives had "got their teeth into" the production problem and were going after it in deadly earnest. Despite the present pressure on production men resulting from the ominous progress of military events in Europe, attendance at the sessions was fully up to the high mark established at previous meetings. There was even some indication that the caliber of the men in attendance was higher than ever. Men who meant business were everywhere in evidence. Interest in the Engineering Display was greater than ever before. Exhibits had been prepared with unusual care and received the closest attention from those in attendance.

Vital features of production, such as weight control, quality control, standardization, engineering liaison and others, were discussed at length in the discussion periods held in the course of each session. High point of the meeting was the Friday evening session in which SAE President A. T. Colwell presented all-out production of the weapons of war as the one sure method to stop blitzkrieg; and T. P. Wright gave a heartening report on production progress to date under the Office of Production Management.

Total registration at the meeting, again held in the Biltmore Hotel, for the three days, Oct. 30, 31, and Nov. 1, was approximately 1000, which equaled last year's high level. Almost all of the sessions were heavily attended, with the hall packed to capacity for all evening sessions. Total attendance at all sessions must have exceeded 3000 persons, and this does not take into account the additional visitors who thronged the Engineering Display during the three-day Show.

All four Pacific Coast Sections of the SAE cooperated with the National management in staging the Production Meeting, as in the past. SAE President A. T. Colwell was in attendance and took an active part throughout the three days. Mac Short, chairman of the Southern California Section; C. F. Becker, chairman of the Northern California Section; E. A. Marks, chairman of the Oregon Section, and T. C. Howe, chairman of the Northwest Section, all collaborated on arrangements with John A. C. Warner, SAE general manager. Robert Insley, vice president of engineering, Menasco Mfg. Co., served as general chairman of the meeting, together with Mac Short. J. H. Kindelberger acted as chairman of the Aircraft Engineering Display Committee. Banquet and Grand Ball arrangements were handled by F. C. Patton, with all tickets sold out more than a week in advance of the event. Numerous other SAE officers and members, both national and local, helped to make the meeting an outstanding success.

Opening the first session promptly, A. T. Colwell, SAE

4 MEETINGS

POINT SAE PROGRESS ON ALL FRONTS

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At the President's Dinner, where President A. T. Colwell presided, praise of the industry's great part in National Defense and congratulations to those present for their intelligent attitude and energetic activity in promoting the SAE program was extended by F. C. Crawford. Honorable Robert Hinckley, Assistant Secretary of Commerce, spoke as did Lt.-Col. D. G. Lingle. Col. Lingle arrived too late to be included in the group picture

Left to right around the outer side of table: Mac Short, Vega Airplane Co.; Eddie Molloy, Ryan Aeronautical Co.; Gunnar Edenquist, Kinner Motors, Inc.; H. J. Stoneburner, Civil Aeronautics Administration; C. E. Stryker, OPM; William C. Lawrence, OPM; Marion P. Crews, Civil Aeronautics Administration; I. M. Laddon, Consolidated Aircraft Corp.; Arthur Nutt, Wright Aeronautical Corp.; Honorable Robert Hinckley, Assistant Secretary of Commerce; SAE President A. T. Colwell, Thompson Products, Inc.; F. C. Crawford, Thompson Products, Inc.; R. R. Teetor, Perfect Circle Co.; W. B. Stout, Stout Engineering Laboratories, Inc.; T. P. Wright, OPM; J. L. Atwood, North American Aviation, Inc.; J. T. Gray, Aeronautical Chamber of Commerce of America, Inc.; Arthur Raymond, Douglas Aircraft Co.; C. L. Johnson, Lockheed Aircraft Corp.; B. Clements, Wright Aeronautical Corp.; J. B. Johnson, Wright Field

Left to right around the inner side of table: B. C. Boulton, Lockheed Aircraft Corp.; S. A. Bell, Hughes Aircraft Co.; R. W. Palmer, Vultee Aircraft, Inc.; J. D. Redding, SAE; Harold Adams, Douglas Aircraft Co.; H. J. E. Reid, NACA; J. K. Northrop, Northrop Aircraft, Inc.; John A. C. Warner, SAE Secretary and General Manager; Val Cronstedt, Pratt & Whitney Aircraft; Carl Squier, Lockheed Aircraft Corp.

president, welcomed members and visitors, emphasized again the life and death urgency of aircraft production in ever increasing volume, stressed the fine work of the standardization committee, and introduced Jack Gray, Aeronautical Chamber of Commerce. In a brief, timely, and humorous statement, Mr. Gray stressed the fact that what the aviation industry produces at this time is much more important than how it is done. He gave approximate figures since military authorities now frown on exact production statements, to show that our output of aircraft has been quadrupled in about 16 months. He predicted that total 1941 plane production would exceed 18,000 planes in this country, as against only 2400 in 1939, and about 6000 in 1940. He closed by stating that in the first seven months of this year more than $284,000,000 worth of aircraft and aeronautical equipment had been shipped to the British. Following this heartening report the first aircraft-engine session was called to order. In the absence of Robert Insley, due to illness, Mac Short introduced H. H. McMahon, sales engineer, Aircraft Accessories Corp., as the session chairman.

AIRCRAFT-ENGINE SESSION

H. H. McMahon, Chairman

Both papers presented during this session dealt with the problems of engine induction systems. Much new information was developed in the course of lengthy discussions which carried the session far beyond the normal closing time.

Aircraft Carburetor Airscoops and Their Effect on Fuel-Air Metering in Flight – F. C. MOCK, Bendix Aviation Corp.

Opening with an allusion to ignorance as being the customary condition of most engineers, Mr. Mock listed the three main stages of engineering ignorance as: (1) thinking we know what we don't know; (2) finding out that we don't know what we thought we knew and proceeding by trial and error methods while we seek a more acceptable solution; and (3) developing an intelligent research program in order really to solve the problem.

Proceeding from this introduction Mr. Mock stressed the fact that we had now learned that we didn't even know where to put the carburetor airscoop with respect to the engine cowling. He cited German practice on a recent military plane which carries the scoop out several inches beyond the cowling where the drag must be great, yet there are probably compensating benefits. Excellent moving pictures were shown to illustrate the work that has been done by Bendix in determining what factors control intake airscoop form and location. By using airscoops with transparent walls and directing kerosene fog into the air stream, it was possible to demonstrate graphically how the air behaves in various scoops and why fuel-air metering is disturbed and unpredictable with certain scoops. Pulsations from the propeller blades also carry into the intake airscoop and, in fact, right through to the individual engine cylinders. This effect seems to hold within about 5 to 8% of total variations from normal. Much investigation has already been devoted to propeller effect but no solution is in sight. One suggestion is that the scoop be narrowed and widened, or even split, so as to spread the propeller blade effect over a longer period. In conclusion, Mr. Mock gave his opinion that scoop problems are not as difficult as some have thought and that scoops have been unfairly blamed for many difficulties arising elsewhere. With continued research, including a substantial government program, he feels that complete solution of scoop problems is in sight.

DISCUSSION

Additional test data were made available in the form of a prepared discussion presented by L. S. Wait, North American Aviation, Inc. Mr. Wait's discussion was illustrated to show the results obtained in extensive flight testing of various intake scoops tried on the NA-73 fighter airplane. A serious carburetor condition developed in early flight testing of this model. In shallow dives the engine would run extremely rough and, as the dive steepened, stop entirely. Various modifications of the automatic mixture control gave some improvement, but not enough. Eventually the difficulty was eliminated by extending the air intake scoop forward until it overlapped the propeller spinner. This eliminated the carburetor trouble and also a serious vibration condition which had been experienced due to the fact that the normal air vibration frequency of the scoop had been the same as that of the altitude compensator unit.

James B. Kendrick, Vega Airplane Co., asked Mr. Mock if the scoop difficulties he had described would also apply in the case of aircraft of very high speed, approaching the compressibility condition. Mr. Mock replied that there was not enough knowledge to make a safe prediction, other than to say that variations would probably increase with speed.

Dr. A. L. Klein, California Institute of Technology and Douglas Aircraft Co., suggested that there is much literature available on other investigations which have been made, into the flow of air through scoop systems. He referred particularly to investigations of a centrifugal air pump system undertaken some time ago at the California Institute of Technology. Research on that problem has developed centrifugal air pumps with efficiency as high as 91-95%.

To a question concerning the effect of air intake surge on the supercharger impeller, Mr. Mock replied that it is considerable, and is a factor of the frequency of the air column back of the impeller.

Victor J. Skoglund, Pratt & Whitney Aircraft, stressed the importance of elimination of icing hazards as an uncompromising feature of all intake airscoop design. He further commented that certain theoretically poor scoops, such as those on the Douglas DC series airplanes, are actually working very well in practice. Mr. Mock remarked that empirical methods would certainly develop practical solutions to this problem in the present stage of knowledge of air-scoop problems, when theoretical efforts might not.

Icing Problems in Aircraft-Engine Induction Systems – L. B. KIMBALL, National Bureau of Standards.

This paper was liberally illustrated with photographic slides illustrating air intake icing developed in laboratory tests. The ice formations were rather terrifying to see, especially as it was stressed that dangerous ice accumulations might occur within a very few minutes. The point seemed particularly well taken in the light of word appearing later in the day that within this very 24-hr period two transport airplanes had crashed, both under conditions where ice may have been a factor.

Two points were especially stressed by the author: (1) We need to make use of protected air intakes for icing conditions, drawing the intake air from inside the engine compartment and in rear of the engine. (2) We need better ice indicators and, more particularly, indicators of icing conditions before ice has actually started to form.

Three specific design recommendations were put forward with respect to the induction system: (1) There should be no change in the aspect ratio of sections of the intake below the carburetor which would tend to prevent escape of ice formations which had been loosened from the metal surfaces. (2) There should be a slight draft in the downstream direction so that ice may fall off the passage walls more readily. (3) There should be no nuts, bolts, bulbs or other protuberances into the air passage which might tend to hinder removal of ice formations. A flush type intake air thermometer bulb has been developed and was shown.

The Aeronautics Division of the SAE Standards Committee (shown above) adopted a large group of standards as presented by the subdivisions during the three-day Aircraft Production Meeting. Approved, also, by the Division, was the Aircraft-Engine Drafting Room Manual (see p. 41)

Standing, left to right: C. E. Stryker, OPM; T. P. Wright, OPM; John A. C. Warner, SAE Secretary and General Manager; and Lt.-Col. D. G. Lingle, The Aeronautical Board

Seated, left to right: J. T. Gray, Aeronautical Chamber of Commerce of America, Inc.; William Littlewood, American Airlines, Inc.; B. C. Boulton, Lockheed Aircraft Corp.; Harold Adams, Douglas Aircraft Corp.; J. B. Johnson, Wright Field; J. D. Redding, SAE Staff Representative, Aircraft and Aircraft-Engine Activities; Arthur Nutt, Wright Aeronautical Corp.; SAE President A. T. Colwell, Thompson Products, Inc.; L. S. Hobbs, Pratt & Whitney Aircraft; Val Cronstedt, Pratt & Whitney Aircraft; L. D. Bonham, Lockheed Aircraft Corp.

DISCUSSION

Prepared discussion presented by Victor J. Skoglund, Pratt & Whitney Aircraft, reported on a parallel study of icing problems as conducted by Pratt & Whitney, in close cooperation with the Bureau of Standards. Mr. Skoglund pointed out that there is no objection to liberal use of heat in preventing ice formation up to the engine's critical altitude, since power loss resulting from use of the heat can be compensated for by further opening of the throttle. He confirmed the need of a protected air intake to avoid formation of impact ice. His research had not found that water droplet size has much influence on the rate of ice formation. He thinks the suggestions made by Mr. Kimball as to maintaining intake passage aspect ratios without change are not possible in actual practice. He also stressed the importance of preventing formation even of the thin layer of "wolf" ice which is considered quite common and relatively harmless. A description was given of an entirely new fuel metering method in which the liquid fuel is fed directly into the body of the impeller and sprayed out through holes in the vanes. It is believed that the system has marked advantages and research is continuing.

In reply to a question from the floor as to whether formation of intake ice could be prevented by heating the carburetor adapter throat, Mr. Kimball replied that the mixture impinging on a warm metal surface has a tendency to insulate itself and build up a solid coating of ice not actually adhering to the metal. This then tended to break off in chunks.

Homer J. Wood, Lockheed Aircraft Corp., asked Mr. Skoglund if ice ever formed directly on the impeller. The reply to this question was that while no ice had ever been found on the impeller, it took from 20 to 60 sec to remove the carburetor for an inspection of the impeller and that some ice might have formed and then melted off in this time interval.

Donald Douglas, Jr., Douglas Aircraft Co., asked about the possibility of an ice bridge forming across the cold air intake door and blocking it while warm air is being fed to the engine. Mr. Kimball thought it was up to the pilot to feed warm air before ice actually started to form, that if the door did freeze over it would probably not be possible to open it again while iced up. He further commented on a question concerning keeping rain out of the induction system, that much work remained to be done on this. Designers have usually relied on inertia of water droplets to carry them out of the air stream at bends in the scoops, and into collecting vanes from which they could be drained off through holes, he said. These holes are usually much too small and, therefore, not very effective.

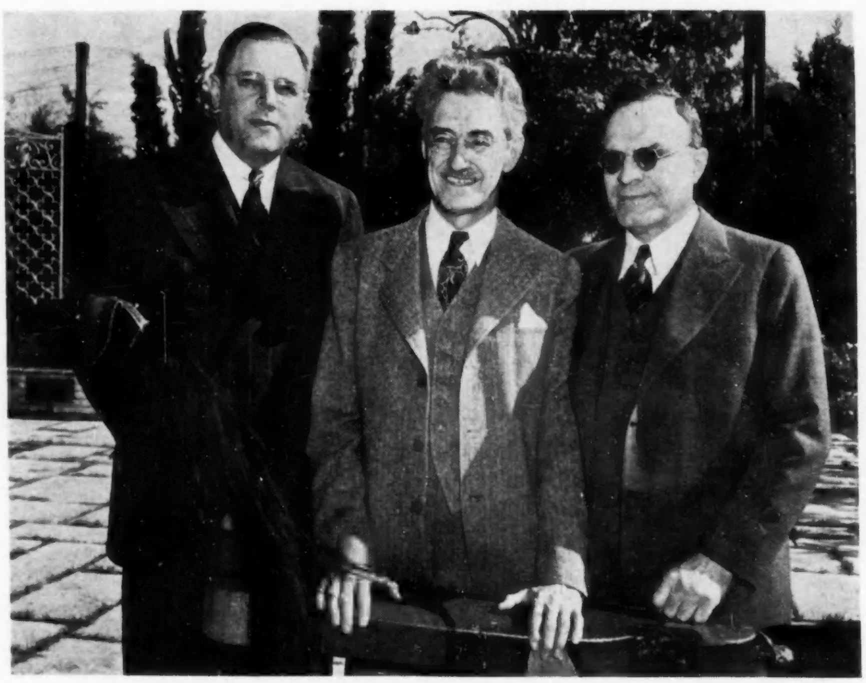
AIRCRAFT-ENGINE SESSION

Gunnar Edenquist, Chairman

Production and design problems vied for attention at this session, one paper dealing with engineering liaison and production control, the other referring to the problems of fuel feed at high altitude, with special emphasis on the application of centrifugal pumps to retard fuel line vapor lock. As with the morning session, this meeting was well attended and the discussion was lively, carrying past normal quitting time without any diminution in interest on the part of the audience.

Engineering Liaison and Production Control – DONALD U. KUDLICH, Wright Aeronautical Corp.

Remarking rather humorously that in popular writing the engineers are always stepping out into the shop and solving production problems on the spur of the moment, and vice versa, Mr. Kudlich expressed a wish for such simple solutions of the mutual problems of engineering and production people. He could admit of only two possible solutions, either to eliminate the gulf between production and engineering people by training them to think and act as one, or to bridge the gulf between them by establishing special liaison through a small number of specialists. The latter solution seems the only practical one since engineering and production personnel are widely separated by training, experience and general attitude. The Engineering Liaison group at Wright Aeronautical has proved quite successful, using freely the facilities of the engineering, assembly, production, and test departments. Directly responsible to the chief engineer, the liaison engineer investigates and coordinates change requests and production difficulties with the project engineers of models affected, as well as with the engineering and manufacturing departments. Modifications requiring basic design changes are referred to the project and experimental engineers, whereas all other types of modifications are investigated on production engines. Three general types of faults are covered: seasonal, chronic, and isolated. Piston-ring scuffing is an example of a seasonal fault. Much work has been done along this line and most of the trouble eliminated. Chronic faults arise from the lubrication system, amount of chamfer used on certain parts, etc. Detailed changes usually eliminate specific faults that arise, but others will develop in this category. Isolated faults arise in unexpected places, such as a combination of accumulated tolerances resulting in serious interference in a final assembly. Such faults are investigated on the spot and eliminated as they arise. Engineering liaison is especially important in maintaining production control under present conditions of vast expansion of production program.



Snapped outside the world-famous Los Angeles Breakfast Club during the meeting were President A. T. Colwell, Past President W. B. Stout and Past President R. R. Teetor. Mr. Stout was the principal speaker of the hour-and-a-half Breakfast Club program and delivered a talk entitled, "Where Are We Going in Aviation?" He was also an important speaker at the Aircraft Production Meeting itself

DISCUSSION

Arthur Nutt, Wright Aeronautical Corp., contributed a story to illustrate the complete absence of the liaison engineer's function in the early days of aircraft-engine development. To study a case of crankshaft failure he had microphotographs made of the cross-section at the point of failure. The officials of the forging company understood these photos so little that they were inclined at first to believe Mr. Nutt's stories that they were pictures of ginger snaps.

A. T. Gregory, Ranger Aircraft Engines, suggested that the project engineer normally does the liaison job and can frequently smooth out production difficulties before they arise by allowing variations from the original specifications.

Mr. Kudlich replied that mass production had now reached a stage which did not permit the project engineer to do both jobs. Engineering contact in the production department must be active before the final assembly stage since it may be too late to make vital changes if they are not considered before then.

Mr. Nutt pointed out that this function is partially filled by the special inspection review department which reviews all rejected parts before they are scrapped.

Gunnar Edenquist, chairman, asked where the findings of the service department would fit into the liaison engineering system. Mr. Kudlich replied that service reports always should feed back through the project engineer, leaving the liaison group free to work directly between engineering and production.

(Continued on page 51)

". . . few people realize how enormous, how complex, how responsible, how different from its normal operations are the defense jobs which the automobile industry has undertaken to do." – K. T. Keller, president, Chrysler Corp., at the Army Industrial College, Sept. 17.

DEFENSE IMPACT ON INDUSTRY

"Without the loyal cooperation of such civilian groups as the SAE, our task of building, training, and equipping our modern Army from scratch would be vastly more difficult." – Major-Gen. E. B. Gregory, The Quartermaster General, U. S. Army, at the National T&M Meeting, Nov. 13, at Cleveland.

Metal Substitute End-Point Is Near, OPM Data Indicates

• A Round-Up on Materials Situations

WE have reached a point where further attempts to substitute one metal for another can't do much good. From now on it will be necessary to substitute a non-metal or reduce the use of the critical metal. That is the only realistic way to summarize a day-full of data given by OPM experts to the Engineers' Defense Board at a Conference on Conservation and Substitution of Metals held in Washington on Nov. 13 under the chairmanship of Dr. C. K. Leith, OPM Conservation Division. Participating in the Conference along with the Engineers' Defense Board was the Technologic Committee of the National Academy of Sciences. Five of the six SAE members of the EDB were present at the conference, those in attendance being: C. L. McCuen, vice president, General Motors Corp.; James C. Zeder, chief engineer, Chrysler Corp.; Frank W. Caldwell, director of research, United Aircraft Corp.; C. E. Frudden, executive engineer, Tractor Division, Allis-Chalmers Mfg. Co.; and Norman G. Shidle, executive editor, SAE Journal.

Distressing figures portending serious shortages on almost every metal came from one OPM speaker after another throughout the day-long Conference, giving little hope for relief of already-crippling civilian deficiencies until after 1942 at least. Only tin, zinc, and tungsten did government representatives fail to "view with alarm," a note of uncertainty appearing even in their discussion of the relatively plentiful metal, molybdenum.

Steel

Few bright spots appeared in the story on steel as presented to the Conference by OPM representatives H. Leroy Whitney, Stanley B. Adams, Charles Halcomb, and R. C. Allen. Highlights included the following:

In 1942 about 14 million tons of alloy steel will be needed – and it looks now as though there would not be enough even for defense needs. The Army alone will need 51.4% of the optimum estimated output . . . and the Navy will probably need an equal amount.

We are facing shortages on all steel alloys – including molybdenum (a Nov. 30 order is aimed at channeling into direct defense production all magnesium in the country not now being so used).

Steel is the "end-point" in metal substitutions. That is one reason that a terrific load is being thrown upon it now. Everybody finally turns to steel as other metals become less and less available.

The demand load is developing beyond capacity in structural steels and in rails.

An acute shortage of steel scrap now existing may result in production of less

(Continued on page 47)

Transition

PATTERNS are hard to trace in the flux that is OPM, but an interesting transition is taking place through the 30-odd industry committees, a structure which may remain a part of the American way of life for a long time to come.

Industry committees were set up as purely advisory groups to OPM executives. However, they are becoming more and more the channel through which industry brings problems to the government. Far more of such problems are being brought to the government today than are generated by the orders issued to industry through these committees.

Thus the industry advisory committees, taking the form of Interstate Commerce Commission procedure, may well become a cornerstone in whatever peacetime government control philosophy is born of the present emergency.

Cobalt Is Under Direct Allocation

Cobalt was put under control by OPM in general preference rating order M-39 (Nov. 4) because of shortages and uncertainty of imports. Termination date was March 31, 1942, and in view of the estimated 400-ton shortage for next year, extension of the order may be expected.

Vital for high-speed tool steel, the ore comes from Africa, with small, marginal production in Pennsylvania. Canada's output is absorbed by Canada and Great Britain. About 1700 tons is the estimated 1942 U. S. supply. A 50% increase in refining capacity will be ready about the middle of next year at a Niagara Falls project, and some low grade deposits may be worked in Canada to supply this refinery next year.

Monthly requests for the metal and chemical derivatives will be made to OPM, and requirements will be allocated on the basis of defense needs.

Materials Allocations Throughout Industry Puts U. S. On War Footing

ALLOCATION of all critical materials in 1942 was announced by the Supply Priorities & Allocations Board, putting American business on an all-out war economy basis.

1. Executive Director Donald M. Nelson of SPAB requested Director General W. S. Knudsen of OPM to obtain detailed production programs, industry by industry, for 1942. Each program will show monthly requirements of critical materials needed to produce:

- Military products and equipment,
- Essential public services,
- Industrial supplies,
- Civilian goods,
- Repair parts, and
- Requirements for capital expenditures.

2. OPM has issued an administrative order setting up the machinery by which the whole allocations program will be developed.

3. OPM has set up a new system for handling preference ratings to fit the nation-wide allocation program.

How the allocations program will work:

- The Automotive, Transportation, and Farm Equipment Branch of OPM, for example, will develop requirements programs for each of its industries. Requirements for the automotive industry will be set up by the Automotive Section of the AT&FE Branch through consultation with the Automotive Industry Advisory Committee and the military services involved.
- The AT&FE Branch will then take up the question of each material required with each of the materials branches involved to determine the amount of materials which can be spared from defense.
- Each allocations program will be subject to further revision, depending upon greater or lesser estimated military requirement for each material.
- Careful review of each of the scores of materials required for the automotive industry for 1942 will be made to see if further reductions might be achieved through:

(a) Simplification and reduction of the number of car models,

(b) Further substitution of materials,

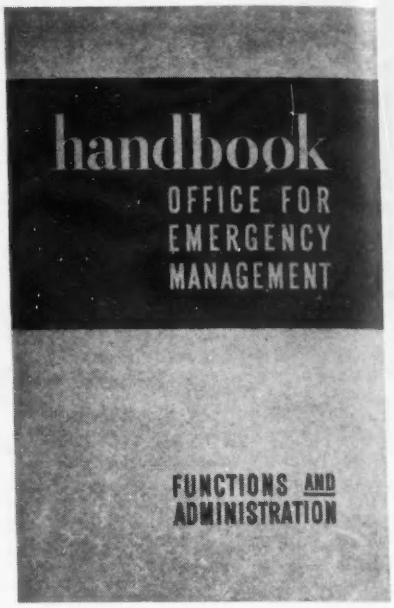
(c) Materials reduction through redesigning, and

(d) Segregation and collection of shop scrap.

- Executive Director Nelson of SPAB will then coordinate the automotive materials program with those of other industries.
- SPAB then reviews the program, having authority for modifications, rejection, or approval.
- Priorities Division of OPM, headed by Mr. Nelson, will issue priorities ratings, or make allocations, to insure that the needed quantities of materials will be available to automotive manufacturers – if there is enough material left over from Army, Navy, Lend-Lease, or other defense agency.

Functions of the industrial branches in setting up their respective programs were outlined in Administrative Order No. 29, signed by Mr. Knudsen. It provides that:

- Each industrial branch shall immediately set up 1942 requirements for its respective industries,
- Each industrial branch shall be responsible for the final disposition of the materials to its industries,
- Effective Dec. 1, every application for preference rating (PD-1) from a company shall be routed to the industrial branch of the industry concerned. The branch will:
- Review the relative importance and general desirability of obtaining the applicant's product or service,
- Indicate the preference rating to be assigned to the company,
- Indicate the approximate date on which materials should be delivered to the company, and
- Pass the application over to the proper materials branches for review. The materials branches may then question the amounts of materials asked for. If the industry branch and the materials branches cannot agree, both recommendations are sent to the Priorities Division which will assign a rating.



Birds-eye view of OEM, top government agency coordinating national defense, is given in this new handbook. Available from local OPM offices and Government Printing Office, Washington: 15¢

Appeals Guide Copper Changes

BECAUSE of the vital importance of copper to automotive manufacturing, revisions and amendments to that sweeping document, Copper Order M-9-c, must be watched from day to day. It may be amended or exceptions allowed on appeal.

In general, whatever favorable revisions that are made automotive-wise, will result from specific appeal to the Automotive, Transportation & Farm Equipment Branch of OPM. While manufacturers of non-defense goods are making their appeals, however, the Copper Branch of the Materials Division is likely to disclose increased defense needs of the metal, or have some bad copper production news to tell.

Considerably more drastic restrictions are being predicted freely in Washington, the most important of which might be the virtual prohibition of copper for radiators. At an informal conference of radiator engineers and OPM officials in Washington on November 10, information was developed which was interpreted by the OPM as conclusive evidence that substitution away from copper is feasible. It was brought out, however, that steel radiators would be from 6 to 20% less efficient, from 25 to 50% heavier, and that corrosion problems of steel radiators were still unsolved.

While automotive manufacturers are appealing for individual relief from specific provisions of the existing Copper Order and its commandments, makers of electrical appliances and machinery, and scores of other copper-using industries will be appealing also through their respective industry branches for specific liberalization of the order.

Furthermore, since the available copper is being zealously guarded by the Copper Branch, appeals must be supported with all available facts if they are to be successful. In general, the SAE Journal has found that OPM officials appreciate the efforts made by automotive engineers to stretch critical materials as far as possible. They have made a favorable impression, when compared with spokesmen of some of the other large industries of the country.

Parenthetically, some OPM officials are appalled to discover that some of industry's spokesmen are not conscious of the fact, as they put it, that "we *are* at war."

Experts Will Review "Needs"

In appealing any detailed provision of the copper order, automotive manufacturers will discover that the materials branches are manned by technical experts who have been charged with seeing that military production is not crippled through lack of materials. To some of these men such considerations as economic dislocations through a "priorities famine," with a resulting loss of jobs, is unimportant as compared with a materials shortage that would hamper defense production. Some of these materials "watch dogs" resent the 1942 automobile models as symbols of defiance in the face of acute shortages in plating materials for bright work, for example.

Several industry branch chiefs have been called "up stairs" to explain their apparent loss of perspective, or an occasional lowering of sights on the defense objective. Yet it could hardly be said that even the most lax of these chiefs have been guilty of pampering a non-defense industry.

Because the copper order will be administered by each industry branch on the basis of specific appeals, the whole structure will soon be cumbersome and spotty. However, because of the complexity of most manu-

…actured products, few generalities covering more than one case will be probable — *few exceptions granted for one manufacturer will apply to others who failed to appeal.*

Automobile manufacturers are appealing (with good hopes for success) for an exception for the "keys and locks" item on List A of the order, and bus makers, for an exception to "heaters" on the same list. Since the OPM has indicated that sufficient copper for essential electrical parts of heater will be allowed, a number of passenger-car companies are planning to substitute steel for copper heater cores to comply with the order. At the Nov. 4 meeting of the OPM Technical Advisory Subcommittee, clarifications of the copper order were requested with respect to "miscellaneous fittings and trim" item of List A. If copper of brass brake tubing is interpreted as "miscellaneous fittings," a number of companies will appeal for an exception, especially truck and bus makers using air brakes, on the basis that alternates would lower the safety of the brake systems.

Defense-O-Grams

BEARING INSPECTION PROCESS, DEVELOPED BY ALLISON, BROUGHT THIS WIRE FROM BRIG.-GEN. G. C. KENNEY, ARMY AIR CORPS: "CONGRATULATIONS . . . IMPORTANCE CANNOT BE MEASURED IN WORDS . . . MAKING IT AVAILABLE TO OTHER MANUFACTURERS IS FINE EXAMPLE OF TEAMWORK THE NATION SO SORELY NEEDS IN THIS EMERGENCY." THE PROCESS? FOR TIME BEING A CLOSE MILITARY SECRET.

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PERMANENT AUTO TAGS ADVOCATED BY CONGRESSMAN GWYNNE, IOWA, ESTIMATING 550 TON SAVING IN STEEL AND $44,060 TO HAWKEYE TAXPAYERS ALONE. NATIONAL STEEL SAVING WOULD BE 22,200 TONS, ABOUT $1,750,000 SAVING TO U. S. TAXPAYERS.

★

HIGHWAY TRANSPORTATION WAS OVERLOOKED IN COMPILATION OF COMMERCE DEPARTMENT OF 123 NATIONAL DEFENSE RESEARCH PROJECTS, HIGHWAY USERS CONFERENCE POINTS OUT. OF 146 PROPOSED PROJECTS, ONLY ONE WAS ABOUT HIGHWAY TRANSPORTATION.

★

NATIONAL DEFENSE HAS UPPED ALL MOTOR TRANSPORT TO NEW HIGHS, COAST-TO-COAST SAMPLINGS SHOW.

★

AUTOMOTIVE EQUIPMENT IN THE FRONT LINES: GASOLINE-POWERED AIR COMPRESSOR UNITS IN HANDS OF FEW SOLDIERS CAN DO WORK OF 50-MAN PLATOON IN ENGINEERS CORPS. TYPE OF JOBS: SAWING TIMBER, DRILLING HOLES IN ROCK

Why and how billions of your dollars are being spent is graphically portrayed in this booklet, free to all from Office of Emergency Management, Washington, D. C.

FOR BRIDGE AND CABLE LINE FOOTINGS, DIGGING TRENCHES, BORING, PUMPING, BACKFILL TAMPING, ETC.

★

QUARTERMASTER CORPS HAS BEEN ASKED BY CAVALRY TO BUY 1000 TRAILERS TO TRANSPORT HORSES AND MULES TO KEEP UP WITH NEW FAST-MOVING ARMY.

★

TO AID ARMY IN OPERATION AND MAINTENANCE OF FORD-BUILT MILITARY EQUIPMENT, FACTORY-TRAINED SERVICE STAFFS ARE NOW STATIONED AT ALL 34 FORD BRANCHES. COMPANY HAS THREE TRAVELING CLASSROOMS, TAKING MAINTENANCE COURSES TO OFFICERS AND MEN IN CAMP.

★

GOVERNMENT CIVILIAN EMPLOYMENT HAS REACHED ALL-TIME RECORD OF 1,444,985, MORE THAN 1% OF TOTAL U. S. POPULATION. MONTHLY PAYROLL: $217,772,054, AVERAGE ABOUT $151.

★

SINCE KITTY HAWK, 37 YEARS AGO, WHEN THE WRIGHT BROTHERS FLEW FIRST, 75,000 AIRPLANES HAVE BEEN BUILT IN THE U. S. INDUSTRY IS ASKED TO MAKE 2/3 THIS TOTAL IN ONE YEAR, A TASK EQUIVALENT TO THE AUTO INDUSTRY BUILDING 53,000,000 VEHICLES PER YEAR, OR ABOUT 12 TIMES THE 1940 AUTOMOBILE PRODUCTION.

Defense Progress

HOW'S DEFENSE?

Here is what Government officials say:

Defense Program:

The whole program has now reached the $64 billion mark. This vast sum, equal to more than three times the nation's 1929 total manufacturing output, is already being looked upon by some authorities as inadequate, and with doubling of some specific programs the $100-billion goal appears imminent.

To date, the automotive industry has been awarded $3½ billion in contracts and subcontracts. This compares with the $3.4 billion (wholesale value) of cars and trucks produced in the U. S. industry in 1929, a little more than $3 billion in 1940.

Defense Plant:

Of the $7,958,203,000 defense plant program, nearly one-half has been completed, and about a quarter is half way completed or further than half way along. This leaves less than one-quarter to be started. Private financing for defense nears the $1 billion mark. Here is how the plant and equipment program looks:

U. S. DEFENSE PLANT PICTURE

(These data do not take into account private financing for expansion required by companies to meet their defense contracts.)

Type of Plant	$ Million Projected	Number of Projects
Iron and steel products	332	52
Non ferrous products	175	21
Chemicals, explosives	578	33
Aircraft, engines, etc.	810	139
Tanks and vehicles	30	12
Cannon and gun plants	279	71
Ammunition, bombs, etc.	791	97
Shipyards	571	68
Machinery	116	76
Electrical machinery	30	22
Miscellaneous	555	37
TOTALS	4267	628

Defense Products:

Aircraft: The bomber program is ahead of schedule, "With most of the designs for the four-engine types shaken down and ready for mass production," (W. H. Harrison, OPM Production Division director). Fighter planes also ahead of schedule, with increased speeds and firing power. Total backlog of aircraft, engines, and accessories, reported by manufacturers, $8 billion. Government-owned bomber plants at Tulsa, Fort Worth, and Ypsilanti, Mich., (Ford) will be in operation by summer of 1942. The U. S. bomber plant at Kansas City is about ready for operation, and the Omaha plant is expected to be running about Jan. 1. Besides Ford, several operating units of General Motors Corp., Hudson Motor Car Co., Nash-Kelvinator, Briggs Mfg. Co., and Murray Corp. are among automobile manufacturers in the aircraft and aircraft parts effort.

Aircraft Engines: Production is ahead of schedules. Backlog: $1½ billion. G. M's Allison backlog is aout $358 million; Chev-

rolet and Buick, $219 million, miscellaneous engine parts about $38 million, bringing the corporations aircraft-engine stake to something like $615 million. Ford's total aircraft, parts, and engine backlog is $737 million, a substantial part of this being for the 2000 hp Pratt & Whitney engine. Packard's Rolls-Royce "Merlin" aircraft-engine backlog is about $167 million, and Studebaker's Wright engine contracts total more than $100 million. Both Hudson and Graham-Paige Motors Corp. have contracts for aircraft-engine parts.

Tanks: Medium (30-ton) tank program was recently doubled from 1000 to 2000 a month, and the heavy and light tank programs have been upped, calling for additional automotive capacity for parts. To date, Chrysler Corp. and American Car & Foundry Co. (acf buses) are leading automotive manufacturers in the manufacture of tanks. Both General Motors and Ford officials have been discussing tank production with OPM, however, and several hundred automotive manufacturers are subcontracting parts for this equipment. Chrysler is doubling its tank capacity.

Trucks: Whole program is ahead of schedule, with General Motors, Ford, Dodge and other Chrysler units, Studebaker, White, Autocar, and Diamond T, among the builders. The last three companies named have coordinated their purchasing programs, a new "emergency" technique thus being born. Willys-Overland Co.'s already-famed "jeep" is under a heavy production schedule, and Ford has agreed to go into production of this ¼-ton Army vehicle.

Guns: General Motors, Chrysler, and Hudson have begun to turn out anti-aircraft and other guns in quantity, as well as undertaking large orders for cartridge shells, gun control equipment, and artillery accessories. The schedules are "well abreast of schedules" one OPM spokesman said recently.

OPM Materials Orders Show Increasing Firmness

OF the 38 raw materials controlled by OPM since March 22, 19 are of vital interest to the automotive industry, and a few others, such as cork, are important although the industry does not use a major portion of the national supply. These 19 "M" orders cover so many automotive materials that they have closed the door to any important substitution.

Other "M" orders due soon are on tin, molybdenum, manganese, cadmium, and ferro silicon, and possibly beryllium and mercury.

These orders regulate the distribution and flow of the metals and materials covered. During the past eight months many succeeding orders have been stiffer than their predecessors. This stiffening, together with supplemental orders and allocations, were caused by upping the defense program on the one hand, and the discovery that the materials covered were actually more scarce than the earlier estimates indicated.

These materials orders (hence the "M" designation) will continue to follow this general pattern:

- "Defense orders" are defined;
- Defense orders are given preference over non-defense orders;
- Rules governing the distribution are set up;
- Stiffening amendments are provided;
- Often complete inventory information and detailed reports on delivery of the material are required.

Automotive materials now under control for defense are:

Aluminum (M-1, March 22) with three supplemental orders, including M-1-c, to direct the distribution of scrap and secondary aluminum. (M-1-a, March 22, M-1-b, April 11, M-1-c, June 10).

Tungsten (M-3, March 26) with two supplemental "M" orders. (M-3-a, June 3).

Neoprene (M-4, March 28) which expired on June 30, but was included in the synthetic rubber (M-13, June 9).

Nickel-bearing steel (M-5, April 10) with two supplemental orders. This was revoked by M-21-a, Sept. 16.

Nickel (M-6, May 15) was the first order to include the mechanics for allocation in M-6-a, on primary nickel, Sept. 30, revoked M-6.

Copper (M-9, May 29) with two amendments and three supplemental orders (M-9-a, Aug. 2; M-9-b, Oct. 1; and M-9-c, Oct. 21; interpreted on page 25, November SAE Journal). Amended on Nov. 1.

Zinc (M-11, July 1) with an amendment, and four supplementary orders, one of which was interpreted two weeks later. The metal was completely allocated. (M-11-a, July 1; M-11-b, Aug. 1; M-11-c, Sept. 1; M-11-d, Oct. 1; Amendment Oct. 16; M-11-e, Nov. 1).

Synthetic rubber, including neoprene (M-13, June 9).

Tungsten high-speed steel (M-14, June 11).

Rubber (M-15, June 20) with a supplemental order and two amendments – one of which prohibited the use of white sidewalled tires (Aug. 8).

Pig iron (M-17, Aug. 1) required that producers must submit specific shipping schedules to be approved by OPM within a stated period. Amendment Oct. 13.

Chromium (M-18, July 7) with an amendment tightening the control on Aug. 22. This order is not very tight, but the Director of Priorities can take over the metal for defense.

Calcium silicon (M-20, July 29) provided for complete allocation on the basis of monthly schedules.

Steel (M-21, Aug. 9) with two supplementary orders, one relating to alloys, and one to warehousing procedures. This order provided an allocation procedure. M-21-a revoked M-5, M-5-a, and M-5-b, on nickel-bearing steel.

Vanadium (M-23, Aug. 14) which resembled the M-18 order on chromium.

Iron and steel scrap (M-24, Oct. 11).

Formaldehyde, paraformaldehyde, hexamethylenetetramine and synthetic resins (M-25, Aug. 23) with an amendment on Aug. 28, amendment No. 2, Oct. 1.

Tungsten (M-29, Aug. 31) which revoked the M-3 and M-3-a orders issued March 26. This order completely allocated the ore, ferro-tungsten and tungsten powders. All tungsten to be used in steel making was allocated. (M-29-a, Oct. 13).

Lead (M-38, Oct. 4, M-38-a, Nov. 1) provided for allocating the entire Metals Reserve supplies, and refiners were to set aside about 85% of monthly output to fill defense orders under Regulation No. 1.

Cobalt (M-39, Nov. 4) was similar in language and interpretations to the tungsten order, M-29.

In general the M orders tell how a material may be distributed whereas the P orders (limited blanket ratings) serve to help the manufacturer to get needed non-defense orders.

A further check is provided by the P orders, however, which are given only upon study of the necessity of the product in the whole defense picture, and the issuing of a rating, such as A-1-a, in each case.

The question "How does a manufacturer get materials or supplies for non-defense manufacture, even if assigned a priorities rating?" is answered by OPM's estimate of defense needs. If there is enough of a surplus of the needed materials he is protected by his relatively high rating. The fact remains that a high rating does not guarantee delivery: *C'est la guerre!*

Maintenance Gets A-10 Priority

HUNDREDS of thousands of the nation's automotive and other industrial plants, big and small, were granted the use of an A-10 priority rating to obtain maintenance and repair materials, in line with a recent SPAB policy to keep the national economy in good running order.

The rating, granted by the Priorities Division of OPM, also can be used to obtain operating supplies which are used up in manufacturing. The original P-2 order, issued Sept. 9, was amended Oct. 16, was interpreted in two orders on Nov. 5, and was amended Nov. 10.

Fleets used to move petroleum products are now included, and priorities of A-10 are now extended to machine and repair shops servicing this type of trucks. Maintenance of public utilities fleets are covered by the P-46 order, Sept. 17.

Granted the use of the A-10 rating, which is a defense rating, include:

Plants manufacturing cars, trucks, trailers, buses, tractors, and automotive parts.

Warehousing: maintaining warehouses for storage or distribution of any material or product.

Wholesaling: distributors of automotive products sold to manufacturers, wholesalers, retailers, or other persons not consumers. Dealers are not included for the time being.

Automotive carriers: urban, suburban and interurban common or contract carriers of passengers or freight.

If a manufacturer needs a repair part, for example, he simply places his repair order

with a supplier and on the face of the order and all copies signs the following statement:

"Material for Maintenance, Repair, or Operating Supplies – Rating A-10 under Preference Rating Order P-22, as amended, with the terms of which I am familiar."

This constitutes legal use of the rating. Since the A-10 rating denotes a defense need, the order placed must be accepted by the supplier under the terms of Priority Regulation No. 1.

The supplier may extend the rating in the same manner if necessary to obtain materials going into the producer's order.

The order emphasizes the fact that the rating granted cannot be used to obtain anything except maintenance, repair and operating supplies *as these are defined in the order*.

The phrasing of the certification to be placed on all purchase orders for such materials makes it mandatory for those using the order to be familiar with *all its terms before using it*. Write Priorities Division, OPM, for an official copy.

Stipulations:

1. Purchase orders for repair, maintenance and operating supplies must be made up separately from all other orders, if the rating is used.
2. The rating must not be used if the material can be obtained without a rating.
3. Producers using the rating may do so only to obtain materials in quantities which are not above certain 1940 levels as defined in the order: provided, however, that the Director of Priorities may permit larger quantities of materials to be ordered and used in proportion to any increase in operations over last year's levels.
4. Misuse of the plan may result in direct punitive action.
5. Utilities and mines covered by separate repair orders are not covered by the present order. However, the plan does apply to all other establishments previously covered by Preference Order P-22, now revised.

OPM Finds Auto Makers Know Simplification

Cursory investigation of automobile models was a shock to several OPM conservation consultants who had assumed that reduction of the number of styles offered would, per se, release more materials needed by the defense program.

The basic premise of mass production, i.e., interchangeability of components and parts, on which the automotive industry has been working tirelessly for nearly a quarter of a century, has brought about a degree of simplification that astounded several investigators who disclosed their efforts to the SAE Journal.

Taking studies made by Dr. Edwin W. Ely, chief of the National Bureau of Standards division of simplified practice, conservation experts noted omission of automotive examples, and wondered why. The answer: Competition has forced all car makers to adopt, long ago, simplified practices and careful standardization both as company policies and industry-wise through the SAE committees.

Metal of Mars Under Control

Implementing numerous recent warnings by OPM officials, three drastic moves were set into motion to control the civilian use of steel all within a week's time.

On Oct. 31, a nation-wide survey was launched by the Compliance Section, OPM's "priorities policeman," with the cooperation of the Federal Trade Commission's examiners.

On Nov. 1 SPAB asked OPM to develop an allocation system for steel, which was put under mandatory control on Aug. 9 by M-21, later amended to tighten the control.

On Nov. 7, SPAB announced the allocation of all materials to every user industry (See "Materials Allocations Throughout Industry Puts U. S. On War Footing" p. 20.)

First Suspension Shows Widespread Compliance

Top OPM officials indicated to the SAE Journal that compliance with orders issued have been generally widespread, and "policing" of orders had for the most part been "educational" rather than punitive.

The drastic action taken by Donald Nelson, director of priorities against the Central Pattern & Foundry Co., Chicago, had been preceded by warnings and urging by E. L. Martin, former assistant to the president, Thos. A. Edison, Inc., and head of OPM's Compliance & Field Section.

The suspension order, the first punitive action of OPM, shuts off all aluminum operations of the company for four and a half months for failure to comply with the famed Order No. M-1, which put aluminum under full priority, and Orders Nos. M-1-a and M-1-c, detailed amplifications.

Many other examples have been cited by OPM chiefs of companies agreeing to make amends for violations. Example:

One of the largest iron and steel scrap brokers in the country agreed to refund to buyers all amounts in excess of ceilings.

Laboratory Supplies Aid Is Extended

To further implement assistance given to research and development laboratories under preference rating order P-43 (Aug. 28), when an A-2 rating was given for scarce materials and equipment, OPM issued an A-5 rating (P-62, Nov. 15) to equipment and supply manufacturers for materials required for packaging such equipment for delivery.

Nearly 1200 laboratories took advantage of the P-43 order, and more are coming in every day. In applying, simply ask for the P-43 form, fill it out, and return to OPM.

Scrap Reporting Is Now Merged

The United States Bureau of Mines, OPM, and OPA have merged their respective iron and steel scrap reporting projects.

Forms PD-149, 150 and 151 cover the entire field of iron and steel scrap reporting. All three forms are returnable to the Bureau of Mines, at Pittsburgh, and not to either the OPM or OPA as heretofore.

Questionnaires sent out during November are designed to cover the October operations of scrap producers, dealers, brokers and consumers. Further monthly report forms will originate in the Bureau of Mines.

Purposes of these reports are two-fold:

- To develop a general policy for the distribution of scrap under General Preference Order M-24.
- To assist in price control of scrap.

The current serious shortage of scrap and the urgencies of the defense program dictate that the forms be given prompt and serious consideration.

Form PD-150 applies to consumers. It combines the former OPA Form 104:8 revised and the Bureau of Mines Form 6-830a MC. It calls for figures on stocks, production, receipts, delivered prices and consumption, all by grades.

The one-time survey of alloy steel scrap provided in Form 144 issued by the OPM has now been put on a monthly basis by including all alloy scrap grades in the revised questionnaires.

Failure to receive forms, it was announced, does not mean exemption from reporting but rather places an obligation on the producer, broker or consumer to obtain them on his own initiative. They may be obtained from the office of the Bureau of Mines at Pittsburgh, or from any Field Office of the Division of Priorities, Office of Production Management.

24-Hr Day In Lead Mines Sets Pattern

CRYSTALLIZING several off-the-record opinions expressed by top OPM executives during the past few months, two official statements to the lead industry brought the national defense needs into clearer perspective.

Two days after Leon Henderson, price administrator, spiked rumors of an increase in lead prices, Director General Knudsen and Associate Director Hillman sent joint telegrams to lead mine operators urging them to work their properties at maximum productive capacities . . . "24 hours a day, six days a week and where possible, seven."

Mine operators were requested to:

- Take all necessary steps to this end, and
- Acquire any additional equipment needed.

"Please advise us if we can be of any assistance," the telegram added.

Control of lead distribution became effective in October under the lead order (M-28) and under several limited blanket ratings issued to facilitate acquisition of the metal. The two new lead statements fur-

(*Continued on page 46*)

Aviation Program Under Full Steam, Fuels & Lubricants Meeting Is Told

AMERICAN aviation expansion is driving hard with the throttle wide open, SAE President A. T. Colwell told 200 automotive and oil technologists who attended the National Fuels & Lubricants Meeting on Oct. 23-24, at Tulsa, Okla. Thoroughly acquainted with everything that is going on and planned, full of his subject, Mr. Colwell's machine-gun speed and precision kept his audience at keen attention for an hour and a half. At other sessions, nine technical papers were contributed by 14 authors. A debate of student engineers closed the program.

Mr. Colwell decried defeatists who say that only in a totalitarian state can Germany's amazing mass production of armament be equaled. The experience and technique of American industry, he declared, already have made possible potential production that will outstrip anything that has been accomplished overseas.

"All our technical and natural resources are being tapped to compete with the whole German economic system which was built for and applied to armaments. I believe," he declared, "we can achieve the goal without seriously dislocating our national economy."

Germany followed the accepted American practice of developing parts specialists who make only rings, pistons, springs, valves, castings, etc., he said. By gearing their entire industry to war production, they were able to obtain the then mythical figure of 2000 aircraft engines per month at the start of World War II. American production starting two years ago totaled only 300 engines per month, has today reached an actual production of 5000 reliable 2000 hp engines per month and in one year will exceed 10,000 per month.

Tool Capacity Soaring

Mr. Colwell said American machine tool capacity in the past year in defense production is equal to the capacity of all previous machine tool output for all time. He pointed out that our tank program alone is equal to all of General Motors output for one year.

Turning to equipment, Mr. Colwell revealed that the American bombsight is today accurate enough to hit a 25-ft circle at 18,000 ft, with the result that millions of dollars are being spent by the Navy to protect battleships. It is paradoxical, he continued, that a manufacturer is called upon to make an armor plate that will withstand a gun of a given caliber, and then gets an order to develop a projectile to pierce the armor plate. But, he said, that is the essence of engineering and American manufacturing industries have been making similar paradoxes commonplace for years.

Lubricant Values Analyzed

In their paper on "Changes Occurring in Oils and Engines from Use," Frank A. Suess, W. A. Jones, H. C. Baldwin, and Bert H. Lincoln, Continental Oil Co., showed that the accuracy of the indications obtained from a comprehensive engine laboratory and road test program regarding the relative merits of different lubricants had been proved by more than a year's commercial experience.

Four oils were tested in engines selected for a wide range of engine materials and design features, and good correlation between laboratory and road results was obtained. "The confirmation of road test data by chemical and mechanical laboratory justifies more positive conclusions from road tests," the authors stated. They showed how the oils were rated in descending order of merit by a meticulous test routine, and how composite ratings were arrived at by considering the oil performance on the basis of several important factors, oil condition, engine condition, oil and fuel consumption, etc.

Oil stability engine testing is the most expensive laboratory means of evaluating oils, yet it is the only one for which no standard method of expressing test results has been worked out, H. L. Hemmingway, Kendall Refining Co., said in concluding his paper on "The Value of Engine Rating Systems in the Correlation of Oil Stability Engine Tests." He made a strong plea for interlaboratory cooperation to develop a standard method of measuring engine dirtiness, and explained a demerit system used in the Kendall Refining Co. laboratory for rating engine and oil condition.

Independent observers rate the engines after test, estimating the condition of each part on a zero-for-perfect to 10 for extremely dirty condition. Two teams obtained excellent checks in rating the same engines independently, he showed, and by means of this system excellent correlation between laboratory and engine tests is made possible. Above 35 demerits the laboratory engine is usually found dirtier than the road engine, while below a 35 rating the road engine is usually dirtier, Mr. Hemmingway has found.

In a paper pointing out the possibilities for extending the use of the Sligh Equilibrium-Air distillation apparatus to solve other time-consuming and expensive test problems, R. C. Alden, T. W. Legatski, and H. M. Trimble, Phillips Petroleum Co., summarized studies made in the laboratories of this company. Title of the paper was "Some Recent Work on the Sligh Equilibrium-Air Distillation Apparatus."

Three base fuels, having widely differing distillation characteristics, were used to make various blends whose equilibrium-air distillation characteristics at an air-fuel ratio of 12 to 1 were studied in the described apparatus. By cross plotting equilibrium-air distillation test results, the necessary compositions in terms of the three base fuels to yield a constant percentage vaporized at various predetermined constant EAD temperatures were determined. These compositions were determined for a range of EAD temperatures and for conditions of 20% vaporized, 60% vaporized, and 100% vaporized.

The authors reviewed known data on the correlation between EAD results and the corresponding results obtained in the fuel induction systems of engines, pointing out that items of motor performance such as starting, warm-up, acceleration, etc., are all related to certain air-fuel percentages vaporized on the equilibrium-air distillation curve at some air-fuel ratio. However, although any one item might check, another could vary widely in relative behavior on other items; but the authors concluded "the equilibrium-air distillation test, if conducted on a sufficiently wide range of base fuels and mixtures would provide many short cuts on problems involving starting, chokes, warm-up adjustments, stalling, etc."

Octane-Volatility Limits Urged

Discussing this paper, Dr. Ulric B. Bray brought out the necessity for development of tests of this type which would supply a simple means of making desired determinations quickly and accurately. He emphasized the economies which would result both in car operation and conservation of natural resources. He suggested setting a limit for both octane number and volatility requirements, remarking that the oil industry cannot afford to continue both upward trends indefinitely. He advocated that the oil industry write top or economic limit specifications, and develop tests which would aid in making them easy to maintain.

R. D. Best, Continental Oil Co., commented on using the Sligh apparatus as a substitute for road testing and for octane testing. He said the apparatus must be greatly enlarged for this purpose and should cover a greatly increased range.

W. G. Ainsley, H. D. Young, and M. L. Hamilton pointed out in their paper, "A Cetane Number Study of Diesel Fuels," that

(*Continued on page* 35)

SAE T&M Engineers Gird For Defense Tasks at Cleveland NATIONAL MEETING

AUTOMOTIVE TRANSPORTATION'S defense role was vividly portrayed at the National Transportation & Maintenance Meeting, Nov. 13-14, in the Hotel Statler, Cleveland, to more than 300 of the country's leading transportation engineers from coast to coast. Four technical sessions, two based on SAE T & M fact-finding projects, were presented at the two-day session.

"We are only in the beginning of this defense program, I believe," Major-Gen. Edmund B. Gregory told his audience at the Banquet, Nov. 13. "Anything less than all-out preparedness would be to invite disaster.

"In our continued efforts to achieve our goal of complete military preparedness – military perfection, if you please – we need and will continue to require the very splendid talents and capabilities of the SAE . . . I am convinced that between us we can build a modern Army whose mobility and striking power will be more than a match for any other in the world," he concluded.

Cutting through a mass of new information gathered during the past few months, F. K. Glynn, American Telephone & Telegraph Co., outlined a clear pattern for intensifying the use of motor equipment throughout the country if this should be required by total warfare. Assisted in this pioneering T & M survey by G. D. Gilbert, Jean Y. Ray, E. W. Hubbard, and Fred B. Lautzenhiser, each of whom presented basic phases of the problem and offered solutions, Mr. Glynn opened the technical session Thursday morning by setting the meeting's theme – Transportation and National Defense.

Lack of response from other types of fleets forced the study along the lines of public utility experience in pooling motor vehicles. However, the analysis showed other operators how they might get more miles out of equipment, and more transportation from dollars spent. The paper showed clearly that public utility operators already have been pooling passenger cars to a considerable extent as a peacetime economy policy. Hence, Mr. Glynn said, a sound background of experience has already been built in this practice. Highlights of car pooling experience:

Each automobile served several men,

Fewer cars were needed,

Sharp increases of mileage per year, per car,

Rented vehicles reduced or eliminated,

Joy riding was curbed.

G. D. Gilbert, Illinois Bell Telephone Co., discussed the operation of passenger car pooling. His experience showed:

Garages of a pooling plan must be located near the headquarters of those using the pool;

Half-yearly revision of lists of persons authorized to use the pool must be made;

These names should be kept alphabetically on card indexes at the pool garages to eliminate all chances of unauthorized use of cars;

Often night shifts are required, both to keep the pool open and for repair and maintenance work;

Management and employees must be completely sold on the plan;

Assignment routine requires rigid administration to prevent favoritism;

Record of assigned time for use of the car must be kept to check against actual time out.

Jean Ray, Virginia Electric & Power Co., a pioneer in pooling, noted considerable partial pooling among fleet operators. Greatest efficiency, he said, can come only from complete pooling, carefully planned and rigidly enforced.

He was not hopeful of any early success or ease in initiating a pooling plan, because such plans must be fitted into the needs of the specific business, and the fitting will be fraught with some discouragements.

In many fleets, company-owned pooled vehicles are supplemented by employee-owned cars and employees are reimbursed. "These arrangments are better than nothing," he said, "and it appears that successful experience is almost unanimous."

In starting a pool, Mr. Ray suggested these guide posts:

(a) Be sure of top management's support of the idea. Enough cost data are available to make a sound case for pooling.

(b) Study costs of your own company's automotive operations, preferably over a five-year period.

(c) Determine from department heads probable demands for cars from the pool. Discount requirements on the basis of the degree of disfavor each executive holds toward the pool.

(d) Estimate the probable use each car will be put to. Whereas in some localities an automobile can be expected to run up 2000 miles per month, in others 1200 may be the limit.

(e) Employees should be required to use streetcars and buses as much as possible. This will reduce the investment for fleet, the cost of maintenance, and overall operating cost.

(f) Provide for handling peak-loads. Usually about 5% additional units will give a safe margin.

(g) Preferences can be eliminated by having all of the cars in the pool of about the same type. Luxury cars belie the whole purpose of pooling – which is *economy and intensity of use.*

(h) Failure of the whole plan will come of family use of cars in a pool. Hence, specific assignment of cars to a foreman in case of some future emergency should not be tolerated.

In introducing the subject of night repairs, Mr. Glynn reported that only one utility fleet replying to his questionnaire did not service at night. Others did so to gain the utmost in intensity of use. Many reported they have no daytime mechanics at all, others only a few.

Some build up replacement units by day, and install them at night.

Some companies believe there might be economic advantage in pooling repair work with other fleets. However, the immediate difficulty in obtaining parts and competent mechanics forces some to fear such a plan, although others have hopes for such a plan.

Speaking specifically on the point of pooling repairs, E. W. Hubbard, General Motors Corp., believed that of the 47,000 automobile dealers in the country facing serious curtailment of automobiles, many would make an effort to get some of the

(*Continued on page 33*)

WEST COAST T&M Sessions Stress Problems of the Immediate Future

by Trescott S. White

WHAT will transportation and maintenance men be up against in the months to come? That was the underlying question which drew some 150 operating men to the West Coast Regional T & M Meeting, Fairmont Hotel, San Francisco, Nov. 5-6. What are the new developments and practices which the alert fleet man should be preparing for, asked the operators – and answers dealing with heavy duty bearings, choice of lubricants, taxicab maintenance, chromium plating, compulsory inspection, butane equipment, supercharging, and magnaflux inspection, were provided by the papers presented. A movie tour of the Wright engine plant, a round table discussion, and a banquet meeting finished up an informative two-day program.

A. T. Colwell, Society president, fired the enthusiasm of every one present by his stirring banquet dinner address, "Behind the Scenes in National Defense Engineering." This country is nearing the end of its "tooling-up" period and is already well started on the most gigantic production effort ever conceived by man – an effort, which with irresistible force will enable us to meet whatever demand is put upon us, stated Mr. Colwell.

John A. C. Warner had good news to report from SAE members who are taking in national defense. Work of SAE committees particularly in regard to standardization is the best possible example of a working democracy, Mr. Warner stated, adding that we should be proud of the opportunities now offered the SAE to serve our country.

Credit for arranging the series of meetings goes to Charles F. Becker, Tide Water Associated Oil Co.; Sidney B. Shaw, Pacific Gas & Electric Co.; and Peter Glade, Purity Stores, Ltd. Carl W. Spring, Shell Oil Co., and William S. Crowell took care of publicity and secretarial work respectively. Fred C. Patton, manager, Los Angeles Motor Coach Co., representing the Southern California Section, presided at the Wednesday afternoon session; Earl A. Marks, chairman of the Oregon Section, presided at the Thursday morning session; and Lee Ketchum, Six Robblees, Inc., representing the Northwest Section, presided over the

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Tanks and Defense Effort Discussed at Closed Meeting

▪ Detroit

AT a closed meeting held in the Hotel Statler on Oct. 27, Lt.-Col. J. K. Christmas, chief of tank and combat vehicles, Ordnance Department, U. S. Army, and A. W. Herrington of Indianapolis, president of the Marmon-Herrington Co., Inc., addressed members of the Detroit Section.

Col. Christmas presented a technical review of the development of the fighting tank and armored cars, illustrated some technical aspects of tank design with slides and showed motion pictures of testing of tanks at Aberdeen Proving Grounds. His only statement for publication was that tank production is going very well nationally and "fine in Detroit."

Mr. Herrington, whose experience with these types of military vehicles dates from his motor transport service in World War I, and who also has been a manufacturer of tanks and military vehicles for the use of other nations' armies since 1930, spoke principally on the kind of morale needed to spur the United States' defense program along. He stated in an interview before the closed meeting that in his opinion much of the criticism being directed at the defense effort was unwarranted. Especially he singled out the criticisms in a national news magazine directed against the head of the Army Ordnance Department (General Wesson). He stated that this department of the Army has been increased 2900% in size in the past two years and that its work is very creditable.

Sniping criticisms directed at OPM are both unjustifiable and unpatriotic, he said. Speaking as a small businessman himself (he defined a small businessman as one whose annual volume is less than $5,000,000), he stated that this group refused a year ago to recognize the very real threat of priorities and materials allocation. They adopted the attitude "it can't happen to me," Mr. Herrington stated, and as a result these businessmen find themselves seriously hurt by restrictions that exist today. Such small businessmen have been seeking defense work by going to their Congressmen, he charged, but it would do better to face the facts of the situation and seek defense contracts in a normal manner, making allowance for the time required to acquaint themselves with the task proposed and retool for necessary production.

Mr. Herrington also accused labor of "asking too much" and warned both labor and management that "it was a selfish attitude of labor and small business which brought France's downfall."

Engineers Equipped For Defense By Peacetime Training Programs

▪ So. California

THE U. S. Army Engineers Corps and land transportation in National Defense were key subjects of a two-speaker program held before 225 members and guests of the Southern California Section, Oct. 17, at the Elks Club, Los Angeles.

Col. Edwin C. Kelton, U. S. Army District Engineer and president, Southern California Post, Society of American Military Engineers, gave a brief résumé of the U. S. Engineers and their technical society, laying emphasis on the rapid growth of the Army Engineers Corps with 231 officers in 1916, to 18,000 officers in 1918. He pointed out how the huge navigation, irrigation, and flood control program established by Congress in 1928 provided large-scale peacetime civilian jobs which train the men for the vast undertakings encountered in time of a national emergency. He emphasized these points by discussing the number and size of the airport and construction jobs they are now undertaking to build.

Dr. Hampton K. Snell, professor of transportation, University of Southern California, talked on "Land and Air Transportation in National Defense," using rail transportation as his chief topic for discussion.

Dr. Snell gave the following five methods of transportation and their percentages of the total freight carried:

1. highways – 5½%
2. pipe lines – 14%
3. inland waterways – 14%
4. airlines – 1½%
5. rail – 65%

In discussing the situation of freight cars in use and needed for future use, it was pointed out that the critical ratio, below which a shortage must be felt, of cars to weekly loadings, is 1.75 to 1. With this in mind, it was also pointed out that during 1940, the ratio was 1.72, and headed toward about a 1.64 ratio in 1941.

Mr. Snell dwelt briefly on the diesel engine in transportation, telling of its revolutionary effect on rail service, both freight and passenger; and also showing how it had proven the most economical power for all motive operation.

In summing up, the following ten items were listed as "must be done" by the rail companies.

1. buy all cars that can be built
2. improve car utilization
3. cut out circuitous routing
4. cut out fall season peak
5. order more locomotives
6. pool facilities
7. awaken labor
8. pool passenger runs
9. coordinate with trucks
10. (above all else) abandon the present competitive set-up for the duration of the emergency.

Sees Alternate Material Parts As Good As Ones They Replace

▪ Chicago

USE of alternate materials in 1942 cars has not affected in any way performance or riding qualities of the new models. The redesigning of parts to meet the new conditions imposed by defense curtailment has, in many cases, resulted in improved performance, efficiency, and/or appearance of these parts. Alternate material change-over is, however, contributing to the higher costs of cars. This is due to the greater expense of making such change-over.

The foregoing is part of the conclusions set forth by Thomas A. Bissell, technical editor, SAE Journal, in his talk "Alternate Materials and Design Trends in 1942 Passenger Cars," delivered before 119 members and guests of the Chicago Section, Oct. 14, at the Hotel Knickerbocker. The paper reported a study, conducted by Mr. Bissell, of developments now under way in automotive engineering departments to achieve conversion to alternate materials.

With the aid of slides, Mr. Bissell gave a behind-the-scenes analysis of what car makers are up against in making changes to new materials; showed how their research efforts are substantially contributing to defense by reducing the amounts of aluminum, nickel, magnesium, zinc, chromium, copper, tin, and other critical materials without a letdown in car performance or efficiency; indicated trends in the material situation in the light of existing OPM edicts.

(Mr. Bissell's Chicago Section paper brought up to date the material contained in his July SAE Journal article "Designing for Alternate Materials" p. 249-259).

In the discussion period which followed, priority problems were frequently cited from the floor. H. G. Smith, executive engineer, Buda Co., pointed out that the OPM decisions were primarily on a passenger-car basis which failed to take into consideration the special problems of the diesel-engine producer. He cited particularly the problem facing manufacturers on the new-type diesel-engine bearings. Discussing substitute materials for cork gaskets, A. J. Aukers, engineer, Victor Mfg. & Gasket Co., said that research indicates that compound materials, made on a synthetic base, will furnish satisfactory alternate materials but that the cost will be from 5 to 10 times that of cork gaskets. E. R. Barnard, Standard Oil Co. of Ind., said eliminating extra "tinware" on new cars was a much desired development. It would tend to place the emphasis on utility, where it belonged, he said. Mr. Barnard illustrated his point by citing the case of a recently-issued shop manual, whose contents were devoted more to servicing the extras and gadgets than to the car itself.

Howard Pile, Chek-Chart Corp., called attention to specifications for lighter oils recommended on some of the new cars using cast-iron pistons. A plea for centralized distribution of priority data and material changes for the benefit of manufacturers and SAE members was made by H. G. Smith, Buda Co.

Fatigue Testing Methods Charged Inadequate by Almen

▪ Detroit

AN evaluation of fatigue testing methods as they apply to automotive testing procedure was offered to the Detroit Section by J. O. Almen, mechanical engineering department, Research Laboratory Section, General Motors Corp., in a paper presented to members and student members at the General Motors Proving Ground auditorium on Oct. 13. Mr. Almen's paper, "Facts and Fallacies of Stress Determination," was heard by an audience of nearly 500 at an evening session. In the afternoon, General Motors Corp. permitted the members and students to make a thorough tour of the Proving Ground. E. E. Wilson, director of the Proving Ground, acted as host.

Mr. Almen led his discussion with the statement that "no means are yet available for reliable determination of stresses in dynamically loaded, highly stressed, machine parts." Stress calculations, he said, are entirely inadequate in most cases; photo-elastic analyses, he stated, can be made to reveal major stress patterns but do not provide useful quantitative values of stress. Similarly, he said, extensometer readings, brittle lacquers, etc., show major stress patterns but do not provide quantitative values. Lack of adequate information, he said, leads to the

"overdesign" of such machine parts by the incorporation of factors of safety.

After outlining his views on the extent of applicability of various methods, Mr. Almen said, regarding fatigue test data, that they "are not directly applicable to the design of light-weight, high-output machine parts because, with few exceptions, they assume: (1) that stress can be calculated, (2) that a machine part must be stressed below the fatigue endurance limit to be successful, (3) that laboratory test specimens are representative of a material when that material is used in a machine part, and (4) that a representative fatigue curve can be constructed from a dozen or less specimens."

He showed how S-N diagrams can be plotted and gave some suggestions regarding interpretation.

The life requirements at maximum stress of a few typical machine elements (automotive parts) as determined by service experience were cited by Mr. Almen. These life requirements, he said, are fixed by the abusive type of user and are sufficiently high so that failure rarely occurs during the lifetime of the automobile except in the case of an accident. He stated that obviously it would be economically wasteful to design such parts for infinite life at maximum stress. The figures he quoted follow:

Automobile rear axle gears	100,000	cycles
Automobile transmission low gear	100,000	"
Auto chassis springs	100,000	"
Automobile transmission second gear	300,000	"
Truck rear axle	500,000	"
Bus rear axle	1,000,000	"

Mr. Almen charged that "published fatigue curves without exception are based on an insufficient number of test specimens." Wide scatter of test points results, and this greatly decreases the value of the data, he indicated. On this score, he declared that "fatigue data should be regarded as mortality data, particularly in the finite life region of the diagram."

From test data available, Mr. Almen drew the conclusion that if a sufficient number of specimens are tested and if the stress scale proportionality is the same for all specimens, it is probable that lines for a series of tests would tend to converge toward a point. He displayed a chart, by means of a projected slide, in which this convergence might occur at about 1000 cycles and 90,000 psi, the latter value being approximately the ultimate strength of the material. He suggested that greater knowledge of the characteristics of fatigue curves at high stress would be valuable in industry and that such tests could be evaluated in terms of the slope of the fatigue curve.

"A large variety of machine elements are constantly being tested for relative durability in the laboratories of industries engaged in the manufacture of light-weight, high-output machines," he said. "In most cases these fatigue tests are intended to compare one design, material or process with another design, material or process. It is axiomatic that nothing can be learned in regard to limiting loads except through tests to destruction. Therefore, the fatigue tests for practically all parts are run to failure and the comparison is made on the number of stress cycles at constant load that each part will withstand. This procedure is followed regardless of whether, in practice, the part in question is stressed below the fatigue limit or whether

WEST COAST BRANCH OFFICE Established by SAE . . . .

A WEST COAST Branch Office of the SAE is being established to better serve the growing needs of the four Pacific Coast Sections of the Society and to provide the most efficient possible liaison for all Society services with that region.

In charge of the newly established branch office will be the SAE Assistant General Manager Edward F. Lowe, whose broad knowledge of Society affairs and experience in operation of the Membership, Section, and Placement work insures maximum effectiveness for the new project. Los Angeles will be headquarters.

Edward F. Lowe

SAE Assistant General Manager who will head new West Coast Branch Office of the Society

Announcement of establishment of the West Coast Branch Office, and of selection of Mr. Lowe as its administrator, was greeted with enthusiastic acclaim by Pacific Coast Section members when made by SAE Secretary and General Manager John A. C. Warner and SAE President A. T. Colwell during their recent tour of the Pacific Coast.

Mr. Lowe has spent many months in the last two years on the Pacific Coast assisting in the development of the rapidly increasing SAE activities in that area and laying the ground-work for the more extensive activities now to be undertaken.

HOLLISTER MOORE, SAE Membership department manager, will expand his duties to include Sections development as a result of Mr. Lowe's assignment to the West Coast, becoming manager of the Membership and Sections departments.

it is a part requiring relatively short life at maximum stress."

The method of evaluating test results is subject to serious error, he declared.

Mr. Almen included in his paper a number of illustrations based on cataloging of leading ball and roller bearing manufacturers, together with interpretations of the curves.

A striking statement in Mr. Almen's talk concerned the common belief that "we cannot conduct reliable tests in the laboratory by reproducing the conditions of service." He declared that "by the time the laboratory investigator has provided for all the conditions that occur in service, he will, in the case of automobile parts, find himself on the road with the complete automobile, and even then he will not represent the type of driver who most severely taxes the strength of the machine."

Costly misinterpretations have occurred as a result of using some of the available test data, he said. He cited particularly "the fiction that a carburized part should have a hard case to resist wear and a tough core to resist breakage which arose from laboratory impact tests." Speaking of such tests on gear teeth he said that if "the gear tooth had been examined after the first impact, the tooth would have been found bent and, therefore, ruined. Hence it would make no difference how many more blows were required to fracture the tooth." He pointed out that eventually engineers have come to realize that gear teeth fail by fatigue and that fatigue failure, for the usual depth of carburization, always originates at the surface of the case.

Similarly, he said, steels for many parts have been selected by false standards and this has resulted in the payment of premium prices for alloy steels for fancied advantages when "fatigue tests on actual machine parts correlated with service records have shown that there is no detectable difference between the high-priced steels and the low-priced steels when used in many machine elements."

Discussions of Mr. Almen's paper were offered by Mr. R. E. Peterson, Westinghouse Research Laboratories, Dr. H. W. Gillett, Battelle Memorial Institute, Dr. John M. Lessells, Massachusetts Institute of Technology, and E. Chester White, Ford Motor Co. Mr. White presented a series of slides which showed recent tests on airplane engine crankshafts with brittle lacquers.

Fatigue Testing

Mr. Peterson offered comments on the fatigue testing of railroad axles, wherein a cam applies a series of loads to a rotating cantilever fatigue specimen "in accordance with the spectrum of stresses obtained in service as measured by strain gages." He said that more research work needs to be done on the net effect of several stress levels applied to the same structural member. He said that Mr. Almen's reference to the convergence of lines on the chart which he had displayed represent a convenient procedure "as a working rule" but that a "consideration of the mechanics of failure makes one think that from a scientific standpoint the picture is not a simple one. . . . The mechanics of failure are so different in the case of tensile and fatigue specimens that one would hardly expect test data to be connected by a continuous function of simple form. What is needed is tests to failure at 1000 cycles, 100 cycles and even 10 cycles. . . . I would like to make it clear that I am not objecting to Mr. Almen's procedure; we need methods for design which are convenient and which give reasonable results."

Comments of Dr. Gillett and Dr. Lessells were read, in their absence. Dr. Gillett agreed that the high-stress end of the S-N curve "has been sadly neglected" and that the engineering lesson of the S-N curve is given by the lower boundary of the scatter band, hence that enough specimens should be tested to show the scatter. He said that the width of the scatter band would make a very good criterion for uniformity or non-uniformity of a metal. He disagreed with "the implied point of view that fatigue testing should be done only at that stress which produces failure in the same number of cycles as in service," adding that service conditions will arise in which the test of a virgin specimen at high stress will give misinformation or inadequate information. Dr. Gillett affirmed his belief in the statements made by Mr. Almen about brittle carburized case and tough core.

Research Engineers' Philosophy

Taking issue with Mr. Almen, Dr. Lessells asserted that the philosophy and outlook of the research engineer is influenced by the type of product made by the company which employs him. "It is cheaper to test 100 connecting rods, costing 50 cents each, in service than to spend time and money on thorough analysis of a particular design. . . . These conditions are not possible when we turn to steam turbines, ships, aircraft or their engines. . . . The appraisal of rational methods using laboratory results will be differently made, depending on whether the assessor is interested in one installation which costs $2,000,000 and which, to be out of service for even a short time due to breakdown, means great economic loss, or whether interested in a commodity like our present automobile, which may be replaced every two or four years and where we have . . . half a million guinea pigs."

Dr. Lessells asserted that photo-elasticity is a quantitative tool as well as a qualitative one, and that the results are in general substantiated by mathematical theory for all cases where analyses are available. He declared his belief in the possibility of obtaining fatigue curves with little scatter between the points, provided the steel is more or less homogeneous and care is taken in testing. Graphs projected on the screen showed samples of such careful tests. He warned of the necessity for exercising care where specimens vary in shape, such as for a nitrided fillet, where failure starts not at, but below, the surface.

Wider use of a method of analysis of stresses as they flow through the machine elements themselves, in proper mountings, as in service, was advocated by Mr. White. Here, he said, is a field for application of brittle lacquers of the "Stresscoat" type. He described tests made with a lacquer coating from three to eight thousandths of an inch thick on an aircraft-engine crankshaft, mounted in the same cylinder block in which it is designed to function, and set up in the Tinius Olsen Torsion Testing Machine. In these tests, torque was applied to the transmission end, the crank end being held rigid in a suitable fixture. Calibration strips are prepared so the experimenter can identify stresses in the crankshaft as low as 19.50 psi, within an accuracy range of 15%.

In slides showing results of such tests, Mr. White pointed out that, at times, conditions, apart from design, cause stress concentration. In one instance he displayed a strain pattern developed in and around a hole drilled for counterbalancing purpose in the crankshaft. He also discussed methods for recording the complete pattern, suggesting a free-hand sketch, with the lacquer crack patterns included on it for a permanent record.

40-Acre Engine Plant Constructed In 10 Months

■ Dayton

PROBLEMS of planning and building an aircraft-engine factory which progressed from the ground-breaking ceremonies to the first completed engine in less than ten months were described by Rudolph F. Gagg, Wright Aeronautical Corp., to the Dayton Section and guests in the corporation's new Lockland plant near Cincinnati, Ohio, Oct. 17.

Conceived in June, 1940, to double the capacity of existing factories and partially meet President Roosevelt's request for 50,000 planes a year, the completed plant covers forty acres, registers a floor area of 2,250,000 sq ft, Mr. Gagg said. It was so planned that it can be easily adapted to other duties at the conclusion of the present emergency.

Space has been used with utmost efficiency and all facilities such as the cafeterias and locker rooms have been located underground. This leaves the main floor free for unrestricted planning and routing of processes. Offices have been standardized with expensive materials and furnishings cut to a minimum. Wiring and piping to machines have also been standardized so that in most cases machines can be replaced or relocated with only slight delay.

One dock has been provided for incoming material whether it arrives by rail or truck. Inspection is accomplished at this dock and the material is handled with a minimum of reshuffling. Adequate facilities have been provided for the cleaning and reclaiming of the several varieties of cutting oils.

The electrical power consumed by this plant is enough for a small city. Four independent lines supply the plant and these lines feed twelve transformer stations totaling more than 20,000 KVA capacity. These stations have been so cross-connected that it is practically impossible to cause an accidental shutdown of any one section of the plant. A general level of illumination of 37 ft-c is provided throughout the plant by 12,000 fluorescent lamps, each having a rating of 200 w.

To produce such a vast structure in so short a time required expert planning and implementation. The first duty of Mr. Gagg and his group was to estimate the size and cost of the new plant and then to formulate a plan which included a schedule of construction and equipment, an outline of operating and management policies and a method of financing.

Two plans were considered. The first was to build a large central plant with approximately six subsidiary plants in a radius of 200 miles. The second, and the one selected, was to build one plant and to obtain the assistance of five or six cooperating companies. This plan distributes the responsibility for manufacturing among large and competent organizations with experienced personnel. The cooperating companies include: Otis Elevator Co., Ohio Crankshaft Co., The Eaton Corp., Graham and Hudson automobile companies. Besides the cooperating companies, there are about 200 subcontractors.

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TRUCK and BUS

PRODUCTION
★ TWO PLANT TRIPS

DIESEL ENGINE
★ IGNITION ACCELERATORS

AIRCRAFT
★ LIGHT PLANE DESIGN and DEVELOPMENT
★ FIRE CONTROL

FULL PROGRAM AVAILABLE SOON

AIRCRAFT ENGINE

FUELS and LUBRICANTS
★ ENGINE WEAR

PASSENGER CAR
★ ALTERNATE MATERIALS
★ ALUMINUM TO CAST-IRON PISTONS

PASSENGER CAR BODY
★ DEFENSE REVOLUTIONIZES BODY PRACTICES

TRANSPORTATION and MAINTENANCE

★ JUNIOR-STUDENT SESSION

(*Continued from page 28*)

Reports on the work done to select a location fill a 130-page book. Primary considerations involved in selecting a location included remoteness from either coast, availability of workers familiar with the metal working trades, nearness to a reasonably large center of population and adequate utilities. Having selected the general location, two more months were devoted to the choosing of a specific location. This required a detailed survey of transportation facilities, roads, housing, and flood hazards.

Operation has been based on line production methods rather than on job shop methods. This was done because it requires fewer skilled operators, it imposes a smaller load on the machine-tool industry and the cost of building as well as tooling is less.

Mr. Gagg has been with the Wright Aeronautical Corp. since 1930 and at the conception of an Inland plant in June, 1940, he was made construction supervisor of that plant. Exactly one year later the first engine was shipped from the new plant. As a result of this work Mr. Gagg has been loaned to the Federal Government for services in connection with constructing the NACA Engine Laboratory at Cleveland, Ohio.

Attendance Record Broken

The Oct. 17 Dayton meeting set an all-time high for attendance at that Section's get-togethers. Six hundred members, guests and Wright employees heard Mr. Gagg speak and toured the new Wright plant. A large contingent of military and engineering personnel from Wright Field and the Cincinnati Engineering Society were guests of the Dayton Section at this meeting.

Blind Flying Trainers Shown in Air College Tour

▪ St. Louis

A STUDENT who learns to fly by instruments can operate any plane regardless of size, because he does not depend on the "feel" of the plane for banks, turns, and similar maneuvers, 125 St. Louis Section members and guests were told during a conducted tour of the Parks Air College grounds, at East St. Louis, Oct. 14.

Demonstration and explanation of the "Link Radio Beam and Instrument Flight Trainer" were given – the device in which a prospective pilot can practice instrument flying and make a record of the course he would have flown if he were actually in a plane. The model goes through all the motions of a real plane when the controls are manipulated.

Equipped with a complete set of controls and instruments for blind flying, the cockpit completely cuts off the pilot's vision. Off to the side is a control station from which the radio beam signals originate and at which a graph of the theoretical flight of the plane is plotted by means of a remote-controlled "crab."

In operation, the pilot closes himself in the cockpit and must "find the station" from which the radio beam is coming and also bring the plane down to a level 300 ft above the air field near the station, at which time, in actual flight, he should be able to see the ground if he is to land.

Training planes were viewed in the hangars during the tour, and motor overhaul shops, instrument repair rooms, welding and sheet metal repair departments, and meteorology and radio laboratories were visited.

Prior to inspection of the grounds, St. Louis Section members were dinner guests of the college and heard an address by President Oliver L. Parks, entitled "Keep 'em Flying," and an address by Frederick H. Roever, superintendent of the Executive and Engineering Schools, outlining the college program of instruction and training.

President Parks described the growth of Parks Air College from its founding in 1927 to its present position among the top-ranking aeronautical schools in the country.

Fuel Conservation Methods Outlined for Fleet Owners

▪ Pittsburgh

THERE is no magic formula by which the fleet owner can obtain maximum fuel economy other than by careful attention to details of carburetor maintenance, ignition maintenance, and vehicle condition, together with a sensible approach to the question of driver training, Errol J. Gay, manager, Commercial Engines & Fleet Division, Ethyl Gasoline Corp., told 197 members and guests of the Pittsburgh Section, at the Oct. 28 meeting, held in the Mellon Institute. "Fuel Conservation" was the title of Mr. Gay's paper.

Stressing the fact that motor fuel is the life blood of the defense effort, Mr. Gay said, "If the United States becomes engaged in a full-fledged, all-out war, industries' demands for fuel and transportation, together with the needs of our armed forces may create a real need for fuel conservation." One division of the U. S. Armored Force estimates its fuel requirement at 120,000 gal per full day's march, Mr. Gay stated, adding, the recent Army maneuvers in Louisiana required about 25,000,000 gal of fuel. Conservation of fuel is a patriotic duty for all and a sound operating policy for the fleet owner, Mr. Gay said.

Pointing to the tremendous gasoline output increases in the past 20 years, Mr. Gay told the engineering audience it would take a canal 10 ft deep, 100 ft wide and 630 miles long to hold the 25 billion gal of gasoline produced this year to satisfy the demands of automobiles, trucks, and buses (not including aircraft and other demands). Of this nearly 6 billion gal are used by buses, trucks and passenger cars owned by commercial fleets. A 3% savings in fuel based on 1941 fleet requirements would amount to nearly 180,000,000 gal. This amount is well worth saving, Mr. Gay stated.

Practical methods by which fleet owners could conserve gasoline were given. They are as follows:

Carburetor Idle Systems – Any engine in city service idles a large share of its operating time. The idle system can save or waste an appreciable amount of fuel per month.

The best procedure begins with maintaining correct carburetor fuel levels. Each carburetor manufacturer has recommendations for correct fuel level setting. The adjustment should be slightly on the low side of manufacturer's recommendations. This will compensate for wear of the needle seat and the float pin during normal service operation.

Many fleet operators have placed a small plug in the side of the carburetor bowl to facilitate inspection of the fuel level. This practice is strongly recommended. The mechanic can tell within sixty seconds whether the level is high, low, or correct. In small carburetors, it is best to use an 8-32 round-head machine screw with a small gasket underneath. In larger carburetors, an eighth-inch brass pipe plug can be used. In either case, the bottom of the hole should be just flush with the recommended fuel level for the carburetor. The brass plug or machine screw should not project far enough into the float bowl to interfere with the float action.

Commercial vehicles should have the idle air-fuel ratio set as lean as possible and still maintain a satisfactory non-stalling idle. Some roughness of the engine may be disregarded, since this has no bearing on satisfactory operation of the engine. If a reliable exhaust-gas analyzer is available, it is most desirable to set the idle by using the analyzer. The best alternate procedure is to use a vacuum gage. In order to be sure the idle air-fuel ratio is set lean, lean-off the idle setting slightly more than maximum vacuum-gage reading.

A usual cause of richening of idle air-fuel ratios is the closing of the air bleed supply for the idle system due to carbon or dirt accumulation. These holes are not difficult to clean and can often be cleaned without removing the carburetor from the engine. It is desirable to check idle air fuel ratios at 2000 mile intervals or less since they tend to change quite rapidly.

Fuel Pump Maintenance – Little is gained in attempting to maintain correct carburetor fuel levels unless the fuel pump is kept up to specifications. It is easy and inexpensive to insert a "tee" between the fuel pump and carburetor with an eighth-inch brass plug that can be removed to attach a fuel pump pressure gage. This enables the mechanic to check fuel pump delivery while the engine is under actual operating condition. Within two or three minutes pump delivery pressure and capacity can be easily checked and the mechanic can decide whether pump repairs are required.

An adequate supply of fuel to the carburetor can help improve fuel economy. If the fuel supply is low so the carburetor tends to run too lean, the driver probably will run with a wider throttle opening and will work his vehicle harder in order to make schedule and thus use additional fuel.

Carburetor Inspection and Care – Fuel is often lost by leakage of gasoline past the vacuum piston which operates the power and accelerating jet systems. A cure for this trouble is the use of an oversize vacuum piston, being sure the piston seats properly at the top of the stroke. Some shops resleeve these vacuum cylinders, while others nickel or chrome plate the piston to the desired oversize.

Care of Chokes – Hand-operated chokes often are a source of wasted fuel because they do not work freely. Well-trained drivers usually return the choke button to its "off" position. However, if there is sticking of the cable between the dash and the carburetor the choke may be left in a partially open position. Each inspection of the engine should have the manual choke mechanism checked. It is considered good practice to have a return spring on the choke shaft so the choke will return to wide open posi-

tion. When starting, the driver will have to hold the choke button out until after the engine is started.

Automatic chokes tend to get out of adjustment as vehicle mileage is built up. Here again examination of the choke during routine inspection can soon prove whether the choke is operated properly. In commercial service it is desirable to set automatic chokes in the lean position so that they come to their wide-open setting as rapidly as possible after the engine is started.

Vapor Losses Due to High Fuel Temperatures – Operators should not overlook the fact that they may be losing fuel during warm weather due to vapor losses through the carburetor vent or balancing tube. This can also occur in very cold weather, when engines are protected by radiator shutters and thermostats. The refiner supplying the fuel should not be blamed for this since most refiners today devote considerable care to maintaining proper vapor pressure and distillation characteristics of their fuels. However, fuel line temperatures are in some cases high enough to heat the fuel to the point where a considerable amount of vapor is formed. With fuel pumps in good condition this fuel may reach the carburetor without causing vapor lock. As it passes the float needle seat the vapors formed by the high temperature of the fuel tend to escape through either the carburetor vent or balancing tube. These vapors are wasted fuel and result in loss of fuel economy.

A simple correction is to see that fuel pumps have an adequate flow of air past them when the vehicle is in motion and that fuel lines are shielded from hot surfaces such as exhaust manifolds. It is better to use metal shields between fuel lines and hot surfaces rather than protecting them with loom. Carburetors should be installed with an insulator gasket between the manifold and carburetor body and in some instances with a down-draft carburetor in stop-and-start city service it may be desirable to provide a metal shield between the carburetor and manifold so heat rising from the manifold is deflected from the carburetor bowl.

Care of the Ignition System – The best guarantee of maximum spark-plug performance is, of course, new spark plugs at short intervals. However, the commercial operator must also watch his spark plug cost and, therefore, should use the criteria of spark plug electrode wear as his indication for the need of new plugs. When either center or side electrodes show signs of serious loss of metal, the spark plugs should be replaced by new plugs.

Normal care of a distributor is always a good move towards best fuel economy. Breaker point gaps should be maintained within the limits as set by the distributor manufacturer and checked at each inspection. At reasonable intervals the distributor should be checked to see that the automatic advance mechanism is operating properly.

Correct ignition timing is perhaps the most important item next to carburetor idle air fuel ratios in regard to fuel economy. There is no reason why any mechanic should neglect a correct setting of the ignition timing provided easily accessible timing marks are available. Many modern commercial vehicles have engines so placed that timing marks are hard to reach. Engine designers first, and fleet operators second, should provide timing marks that are more easily seen so the mechanic does not pass over this item because it is difficult.

With accessible timing marks and a good timing light, the ignition should be checked with the engine at idling speeds. Timing should be set as far towards maximum power as the fuel being used by the fleet operator will permit. No engine should be allowed to operate with severe knock or detonation. However, at the same time care should be taken not to over-advance the spark, particularly on older equipment. A simple procedure to establish best timing is to check acceleration time on a level stretch of road or the time required to climb a given length of hill. Best spark timing is that which gives the best performance with no knock or best performance with slight knock, depending, of course, upon the compression ratio and knocking characteristics of any given make of engine.

Need for Determination of Best Road Settings – Good tune-up procedure for commercial vehicles must be based on road experience. It is necessary to make road observations on more than one vehicle of a given make and model to establish best spark setting and carburetor setting. After sufficient road experience has been built up with a given setting to prove its value, it can then be incorporated into routine tune-up procedure. Even when chassis dynamometers are available for tune-up, it is still desirable to have available the background of road experience. This will indicate the best engine speeds at which to set ignition timing and check carburetor air-fuel ratios.

Vehicle Operation for Best Economy – It is highly desirable for fleets to maintain a weekly gas and oil mileage record. Monthly records cause too great a time lag between the occurrence of trouble and its cure. These records should be given to the shop foreman and those men directly responsible for the care of carburetor and ignition systems. Vehicles below the fleet average should be red penciled, and given an immediate checkover.

Spillage of gasoline when filling vehicles, together with leakage of vehicle tanks due to expansion, can account for a considerable loss of fuel per year. There are devices now on the market that will shut off automatically when the gasoline reaches the filler nozzle. The amount of gasoline placed in the tank can be determined by the length of the nozzle. Large fleets might benefit by the use of such devices.

There is plenty of data to show that oil costs rarely exceed four per cent of total operating costs. In some cases this figure is as low as two per cent. Improper selection of the type of oil, or length of drain period, can often increase engine troubles which may directly reflect in increased fuel consumption.

Fleet operators should study the scheduling of vehicles to provide a minimum of "dead head" operating time, and to determine whether the operation is set up for best fuel economy. All drivers should be instructed to shut off their engines whenever possible. Severe cold weather may provide an exception to this rule. Engine-speed governors should be kept in condition. There is no one thing more destructive to fuel economy and general engine maintenance than overspeeding of the engine. Thought should be given to reducing vehicle speed, providing schedules are not affected. Other fuel losses generate from under-inflated tires, misaligned front wheels and dragging brakes.

Discussion under Chairman William A. Gruse got under way with a statement by R. J. S. Pigott, staff engineer of Gulf Research & Development Co., regarding the tendency of car owners to use oils of lighter viscosities in an effort to save gasoline. He warned against using oils of too light viscosity, adding that it was better to lean toward oils of high viscosity rather than toward oils of too low viscosity for safety's sake. There is also a tendency among owners of small fleets of trucks to "lean-up" the carburetor adjustment too much.

Stuart G. Page, Equitable Auto Co., said that for operation among Pittsburgh hills, he had been removing the car speed governors from his passenger-car type of vehicles and had noticed no particular increase in gasoline consumption; and that this made the drivers happier, as they had felt that governors implied criticism of their driving ability. Mr. Gay suggested "more education" for the drivers.

Engine speed governors are almost indispensable for buses and trucks, it was agreed, as otherwise the driver might have little conception of the speed at which the engines are operating in the lower gears.

David G. Proudfoot, Pennzoil Co., introduced the subject of lead in the crankcase, and ensuing discussion brought out the fact that the newer and clearly refined oils have more tendency to reveal any slight amount of lead that might be present, even though this is usually less than 0.5% by weight, and less than one-tenth of that amount by volume. Deceleration, piston ring wear, and cold weather operation tend to cause lead deposits, it was stated.

One discusser said it is often possible to start a bus that has been stalled by "vapor lock" by tossing a couple of glassfuls of water over the fuel pump on the engine. This condenses the bubbles in the pump and gives the fuel pump something to pull on. Mr. Pigott said that his tests with thermocouples had shown that the fuel pump, as a rule, adds only 5 to 10 F of heat to the fuel, but that the fuel lines (to and from the pump) often are too close to hot parts of the engine and so add a troublesome amount of heat to the fuel.

Joseph Beatty mentioned the possibilities of increasing gasoline mileage by using a generator control to reduce the power consumed by the generator when the battery was fully charged. Mr. Gay agreed that this might add a fraction of a mile per gallon.

Charles E. Chambliss, Jr., Chevrolet Motor Division, General Motors Corp., discussed the use of 30 and 40 SAE oils for heavy-duty use, and 20 and 10 SAE oils for passenger cars. This led Mr. Pigott to explain that oils had a dual duty to perform, that they had to have sufficient body to resist piston pressures and yet flow freely enough to cool the bearings. He said that load-carrying capacities were largely determined by the viscosity of the oil.

Mr. Gay concluded by saying that, owing to the needs of national defense, gasoline would probably have a tendency to be heavier rather than lighter during the next few years. This would make it more difficult to maintain the same octane numbers.

Defense Progress Discussed by Colwell

▪ *Colorado Club*

SAE President A. T. Colwell spoke before the SAE Club of Colorado on Oct. 24, at a meeting held in the Edelweiss Restaurant, Denver. John A. C. Warner, secretary and general manager, SAE, was also present at the dinner meeting.

"Behind the Scenes in National Defense Engineering" was the subject of Mr. Colwell's slide-illustrated presentation.

Fales Discusses Passenger Car Trends

■ *New England*

"REVIEWING 1941 and what is supposedly ahead in 1942 we find that the super-performance cars being turned out are not suited to New England roads," Dean A. Fales, professor of automotive engineering, Massachusetts Institute of Technology, told members of New England Section, at the Nov. 13 meeting held in the Engineers Club, Boston. Subject: "Latest Trends in Passenger Cars."

New cars, with a large proportion of their weight in the front, are hard to handle in that particular section of New England where sharp curves, hills, and high-crowned roads prevail, Mr. Fales said. Also the combination of hilly roads, low-seating positions, and bright headlights makes night driving hazardous. Steeply sloping windshields and rear windows increase blind spots, the latter gathering snow in winter.

Today's mudguards are spectacular, Mr. Fales said, but warned that when priorities become pressing there will be "headaches" trying to meet the need for alternates. Viewing the overall picture, Mr. Fales felt it was safe to say that alternates are as good, and some better, than the materials used in previous cars. However, they are more costly, he added. Parts will definitely be given priority to keep used cars on the road, even if it means no new cars.

There are no substitutes for the certain materials needed for exhaust valves, Mr. Fales said. There must be chromium and nickel, he believes. Cooling systems are facing trouble through lack of copper, but the speaker expressed faith that engineers will find some way to meet that problem. There has been trouble with water pumps, he said, and we may have to go back to the old packed types.

"Those who are anticipating plastic bodies expect too much too soon," Mr. Fales stated.

The public, becoming economy-minded through shortages and taxes, is looking for smaller cars, and is thinking in terms of miles per gallon rather than miles per hour, Mr. Fales concluded.

Lubrication Problems Increased by Engine Use

■ *No. California*

DEFENSE requirements will call for performance and operating conditions more severe than gasoline engines have ever before faced, stated Lloyd H. Mulit, Standard Oil Co. of Calif., in opening his paper on "Piston Gumming" at the Oct. 14 meeting of the Northern California Section. Lubrication problems heretofore connected only with diesel engines are also problems of gasoline engines, he added. The dictionary definition of lubrication as "making slippery" is no longer adequate for present-day lubricating oils, which must satisfactorily lick the problems of piston gumming, ring sticking, detergency and many others, he said.

Of one hundred cars in the San Francisco Bay area which had been sent to garages for checking or overhaul because of excessive lubricating oil consumption, it was found that 77% had clogged oil rings, and that piston skirts were coated or gummed in 62% of the cases, stated Mr. Mulit, adding that although the cause of gumming is primarily high temperatures, this trouble can be eliminated or reduced by use of proper additives. Many tests were made, however, before a compound was found which would (1) limit oxidation (2) prevent depositation of gum formed, and (3) still not interfere with other desirable properties of the oil.

Basing observations on test results obtained in actual engines in a laboratory, Mr. Mulit stated that as service or temperature conditions increase, more compounding is required to bring gum formation to a reasonable value. It was observed also that an increase in air-fuel ratio caused an increase in gum formation due to more air being present in the blowby gases. This would explain one reason why diesels experience more gum formation than gasoline engines, Mr. Mulit added. Faulty ignition is another cause of gum formation, he asserted, explaining that when a cylinder misses fire, the blowby contains a larger percentage of air than when firing normally. The remedy for excessive gum formation, concluded Mr. Mulit, is to maintain proper fuel mixture, proper ignition, and the use of properly compounded lubricating oil.

Discussion under the technical chairmanship of Salazar Onorato, Union Oil Co. of Calif., brought out two challenges as to the application of Mr. Mulit's findings. Hans Bohuslav, Enterprise Engine & Foundry Co., stated his belief that the quantity of air or oxygen in the blowby was of much less importance than temperature in causing gum formation. He pointed out that marine diesel engines have operated for years on non-compounded oils without showing excessive gum formation or lacquering, even though the blowby has been entirely air. Therefore, the question of air-fuel ratio and gasoline versus diesel, should be given little consideration when dealing with gum formations, Mr. Bohuslav added.

Sidney B. Shaw, Pacific Gas & Electric Co., questioned the relation between the piston skirt discoloration used in Mr. Mulit's tests and actual operating properties of the lubricating oil, such as ring sticking and oil-hole plugging. Further along this line, Edwin E. Charles, General Petroleum Corp. of Calif., asked if presence of fuel soot did not make piston skirt appearance a faulty guide as to suitability of a lubricating oil. Extensive laboratory tests, replied Mr. Mulit, indicate that piston gumming and deposits on piston sides are indications of a faulty condition of the oil, which would soon result in ring sticking and plugging, and that the selection of an additive which reduced lacquering is the remedy for the other conditions, also.

2000 hp-hr per gal is a good average lubricating oil consumption, stated Mr. Mulit in response to a question raised by Mr. Shaw. Extremes of 400 hp-hr per gal or 4000 hp-hr per gal are equally undesirable, he added.

Replying to a question by Charles A. Winslow, Winslow Engineering Co., Mr. Mulit pointed out that all oils contain oxidation inhibitors, and that when these are used up the rate of oxidation increases rapidly. Similarly, when the additive is used up in a compounded oil, its rate of gum formation, sludging and oil-hole plugging increases rapidly.

The trend toward supercharging truck diesels should cause an increase in piston gumming, said Mr. Mulit in response to a question by Peter Glade, Purity Stores. However, Mr. Bohuslav called attention to the fact that if truck operators would be satisfied with a reasonable increase of power, say 30 to 40%, and that if the engine was properly designed for supercharged operation, the piston temperatures would be no greater than ordinary and, in fact, might be less due to cooling effect of scavenging air. Therefore, under proper design and operating conditions there should be less gum formation when operating supercharged than when operating under un-supercharged conditions. Mr. Bohuslav again pointed out that it was the high temperature of overloaded engines rather than presence of air which was the cause of piston gumming.

"The Legacy of Dr. Diesel," a motion picture with sound, depicted many of the events in the life of Dr. Diesel, and continued with an account of several years development by the Standard Oil Co. of Calif. of a compounded lubricating oil meeting present-day requirements.

German Aircraft Engine Parts Displayed At R. W. Young Talk

■ *Milwaukee*

AT the Oct. 10 meeting of the Milwaukee Section, Raymond W. Young, chief engineer, Wright Aeronautical Corp., read his paper on the Mercedes-Benz DB-601A aircraft engine. (Mr. Young's paper was printed in full in the Transactions Section of the October SAE Journal p. 409-431). Over 300 members and guests inspected the engine parts on display at the meeting.

The Student Branch of the University of Wisconsin at Madison met on Oct. 14 for the first meeting of the season. It was in the nature of a membership pep rally. Short talks were given by P. C. Ritchie, advertising manager, Waukesha Motor Co.; and by Charles T. O'Harrow, student chairman of the Milwaukee Section, T. L. Swansen, and Walter Strehlow, the latter three being connected with Allis-Chalmers Mfg. Co. They cited the advantages, benefits and opportunities of SAE Student Membership.

Transmission Design Reviewed by Backus

■ *Northwest*

IN developing a commercially practical truck transmission three questions need to be considered: (1) Will it work under all conceivable operating conditions? (2) Can it be sold at a price that is reasonably competitive with units already available? (3) Is it simple enough to be easily serviced in the field and of such a design as not to require frequent adjustment? The above instructions came from "Transmission Design and Operation" a paper jointly prepared by E. L. Ludvigsen and Thomas Backus and delivered by Mr. Backus at the Oct. 23 meeting of the Northwest Section, held in Crawford's Seafood Grill, Seattle, Wash.

"For truck installations we believe that the conventional truck selective gear type of transmission most satisfactorily answers the requirements of a commercially practical unit." Mr. Backus said, "Its ability to function satisfactorily under all conceivable operating conditions is not a matter of conjecture but of actual experience. Adaptation of

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SAE T&M Engineers Gird For Defense Tasks at Cleveland NATIONAL MEETING

(*Continued from page 25*)

prospective truck maintenance work. He suggested the possibility of:

Centralizing replacement parts where they would be more available. Unnecessary duplication of parts stocks must be controlled, he said.

Dealers have already been offering training for apprentice mechanics, and will continue this offer in larger numbers, he believed.

"Closer cooperation between manufacturer, dealer, and the fleet operator, based upon a broad understanding of mutual problems, will do much to improve existing facilities to the end that the defense program will be better served," he concluded.

How can fleet passenger transportation be augmented?

A part of the answer is the use of employees' cars, according to Mr. Glynn. Two-thirds of the utilities fleets permit some use of employees' cars. Of this two-thirds many fleets hedge the use of employees' cars with so many restrictions that the effective percentage is far less.

There is a general use of drive-yourself cars for peak loads, occasional requirements and special uses. Therefore, with its wide acceptance, no recommendation is necessary except to say that if in the emergency drive-yourself companies can continue to operate, they may provide excellent help.

There seems no general enthusiasm for the substitution of buses, trolleys and other local transportation for company passenger cars among utility fleet operators. Most fleets report that on account of the nature of the work they cannot recommend this substitution.

Increasing hours in service for both passenger cars and trucks was a question which brought out the fact that the 8-hour day and 40-hour week are now rather generally observed by public utilities. Problems:

Office and home premises must be made available for utilities installation and service for sufficient hours in an emergency to permit two shifts of workmen.

If workmen work two shifts, supervision must also, thereby doubling up on the use of passsenger cars as well as trucks.

In dire emergency of national disaster there will be no "off hours" or week-ends and the problem of intensity of use of motor equipment may automatically solve itself.

Solutions:

Operate emergency service trucks on 3 shifts and 24 hours per day. Have crews take odd days off in lieu of Saturdays. This permits pooling trucks.

Use passenger cars (coupes) at night for emergency service by providing slip-in partitioned bins and drawers to carry necessary equipment.

Load all materials and tools on the work cars and trucks at night to minimize loss of time in leaving garage.

Utilities may be compelled in emergency to cooperate with the Army and Navy, Mr. Glynn pointed out, because the Signal Corps, Ordnance, Quartermasters and other branches of the service have purchased utilities'-designed trucks and equipment in large numbers.

There seemed to be an even division of opinion among the utilities fleet superintendents on pooling trucks within a company. With no particular reference to the type of utility, about half believe that a truck should be assigned to a man or crew and used exclusively by him or them while the other half has apparently and successfully standardized their truckload of tools, equipment, etc., as to make trucks available to any workman or gang. Of the utilities reporting pooling of trucks, only one reported unsuccessful experience.

However, workmen can be "pooled" by changing from one man to two men crews on ½ ton trucks and operate installation and service crews from a foreman driven truck. Further, we can operate gangs from tool and supply trailers with one truck to move 5 to 10 gangs from job to job if a truck for each gang becomes unattainable.

Some fleet superintendents went beyond the questionnaire and suggested that with similarity in design of trucks there would seem no valid reason why the gas company, the light company, the steam company, the water company and the telephone company, etc., could not in dire emergency, interchange many of their vehicles.

Just how far the utilities would wish to go in this cooperative effort in the pooling of vehicles would depend entirely upon the severity of the emergency. If sizeable cities were bombed and communities destroyed, it is probable that such pooling of effort would come about without dissenting voice, he thought.

Pooling, or rather cooperative effort as it might better be called, on a national scale between the various utilities was evidenced in its highest degree during the New England hurricane of September, 1938, for example.

Regional emergencies could probably be handled by utilities through the temporary assignment of vehicles from nearby unaffected areas.

It was gratifying to Mr. Glynn to find that the utilities practically unanimously made repairs on trucks at night, thereby preserving the usefulness of the trucks during productive hours.

Pooling repair work with other fleets is a relatively new thought, but all the reasons given why repairs should not be pooled with other fleets could be easily "knocked down" the author thought.

"I want to add my statement to those of cabinet members, congressmen, and other government spokesmen, that the American standard of living is being lowered and that higher costs will prevail for everything. Trucks will not be an exception. The deeper we get into this war 'they, the trucks, ain't goin' to be what they us'ta be!' Perhaps even the factory branches will be selling us salvaged parts (and we will be glad to get them) before this emergency is over!"

Fred B. Lautzenhiser, chief transportation engineer, International Harvester Co., discussed in detail pooling of repair work for national defense.

Lt.-Col. W. B. Johnson, chief, Engineering Section, Quartermaster Depot, Holabird, discussing Army motorization problems with Charles H. Miller, vice-chairman, Cleveland Section, who worked with Chairman Arthur Townhill, Secretary Harry Hooker, Treasurer Harry F. Gray, Richard Huxtable, Robert Gottron, vice chairman for Akron and Canton, R. F. Steeneck, Floyd Stewart, Roger Weider, Edward P. Kerruish, Fred Haushalter, Editor Mark Mulcahy, and their associates in making the National Meeting one of resounding success

T. L. Preble, SAE Vice President for T&M, and chairman of the Society's T&M Activity Committee

Major-Gen. Edmund B. Gregory, The Quartermaster General, honor guest at the National Meeting

F. K. Glynn, who told the meeting how to get more out of transportation equipment

LAUDING SAE cooperation with The Quartermaster Corps, Major-Gen. Edmund B. Gregory, The Quartermaster General of the United States Army, told the National Transportation & Maintenance Meeting, Nov. 13, in Cleveland that ". . . We are not going to be satisfied until we have motorized every division sufficiently to carry every man in it . . . the new motorized division has 16,129 men and officers . . . these men and their equipment weigh 3753 tons . . . its convoy consists of 2434 vehicles, and if placed in column formation, with 30-ft intervals between vehicles, the division would be at least 30 miles long . . . or, if moved by rail would take 1707 freight cars or 17 trains each a mile long." The general cited specific achievements made through QMC-SAE cooperation, warned that the "Army market is unlike civilian requirements," and that if the Army "does not know today what it may want tomorrow" it is only because no seer of this stripe exists – at least not on the side of the Allies.

"Already we have a slight indication of what the wrong thinking can bring to transportation in the condition that exists today. Already one truck manufacturer has been forced to 'freeze' a great number of repair items in his warehouse stocks in order to insure availability for vehicles that are actually out of commission. This 'freezing' consists of preventing the removal or sale of any of these parts unless the truck owner's name and engine number of the vehicle needing the part are furnished," he said.

Sane planning for emergency maintenance must therefore start now with a program of parts manufacturing that will not only insure adequate stocks of parts and replacement units in all sections of the country, but this planning must also provide for the maintenance of an adequate balanced stock of these parts at all times in all localities. Availability of adequate repair parts strategically located is the basic foundation around which the pooling of truck repair work must be built.

All factories will probably continue to school mechanics to take the place of those called to the colors and to other defense work, he believed.

"From a broad industry standpoint a pooled repairs program can be worked out on a national basis," he said.

Specialized engine rebuilders in metropolitan centers might be utilized to an advantage should the emergency be of sufficient size and duration to make the necessity of complete engine overhaul or replacement necessary, he believed.

"Each emergency locality should have its own individual supervisory committee. The committee should consist of the truck manufacturers' local service representatives together with the local fleet supervisors," he suggested.

Mr. Glynn reported that about half of the utilities fleet superintendents reported that they had experienced beneficial results through a careful rescheduling, rerouting and improved dispatching of their trucks. Perhaps in the extreme emergency the appointment method of handling such work will have to go by the boards for the sake of greater production of men and equipment. "Fleet superintendents should review periodically, scheduling, routing, and dispatching practices with all departments concerned and particularly now at the start of national emergency. The benefits are greater production, less mileage, lower costs, savings in gas, oil, tires, repairs, etc.," he said.

"A utilities work truck rolling down the road represents a total loss of production for utilities truck drivers are craftsmen. A utilities truck at the job can be 100% productive. Let's keep the rolling time to a minimum and the working time of truck and men at a maximum during the emergency," he concluded.

Two-thirds of the utilities fleets use "hours of productive service" as an index while the other third use miles. This latter index could be debated for, of course, miles operated by a winch truck unless the speedometer is attached to the engine is not indicative of the work of the truck. Mr. Glynn pointed out that the only exception to this is the use of "miles per customer vs. customer density." Perhaps other work factors and indices should be developed as this particular exception seems to be rather thought provoking.

"There is much to be done, many plans to be made, practices to be initiated with painstaking hours of labor on the part of the public utilities fleet superintendents if the intensity of use of their motor vehicle equipment is to be brought to the point where it will meet or preferably move a step in advance of the vital requirements of the national emergency.

"This task at best is enormously difficult. It becomes easier in direct proportion to the efficiency with which our men out in the field use, repair, and maintain the equipment you already have. Washington is far away from the points where such

arrangements can be designed and accomplished advantageously.

"In the last analysis the emergency conditions which confront us need not and must not throttle individual initiative. For those who recognize these conditions, they simply open up new avenues down which such initiative can travel."

Under the chairmanship of H. E. Simi, chief engineer, Twin Coach Mfg. Co., Fred B. Lautzenhiser, International Harvester Co., brought transportation engineers up to date in respect to state restrictions of motor vehicles.

He reported that the picture just ahead is brighter than at any time before, largely because public officials have been educated to the losses incurred by state barriers through the work of organized interests in motor transportation.

His paper, illustrated with exhaustive tables and diagrams of various types of vehicles, will stand with the most important studies, year by year, of the problem as an important contribution to the subject. He credited Louis Morony, Automobile Manufacturers Association, for much of the information he offered.

He pointed out that today highway officials are thinking along parallel lines with the SAE committee, and showed by a tabulation that the new SAE Code is more liberal than the recommendations of either the American Association of State Highway Officials or the Western Association of State Highway Officials. He pointed out that the SAE Committee considered the "bridge formula" as unnecessary, because of the view that all bridges and culverts should be designed to carry the load calculated into the highway. "Otherwise, we must assume that all highways are to be subjected to load limits of bridges," he pointed out in discussion.

Reciprocal statutes have been enacted by 17 states, he reported. The trend is accelerating. "In my opinion, trade barriers to truck transportation are on their way out," he concluded.

Forced by the need of getting more transportation out of every dollar spent during this national emergency, Errol J. Gay, Ethyl Gasoline Corp., stimulated his current study of "Engine Deposits – Their Prevention and Removal" by restating the problem. Eighty-three members of the Society agreed to work with him in reporting their fleet operation experience on this problem. He will make a report at an early meeting of T & M members of the Society, he said. The study is the assignment of the T & M Activity Subcommittee B-1, of which Mr. Gay is chairman. S. G. Page, Pittsburgh, served as session chairman.

Commissioner John L. Rogers of the Interstate Commerce Commission, and Chairman of the Central Motor Transportation Committee, announced a series of intensive studies now under way to fit the nation's motor transport industry into the requirements of all-out national defense.

"The person responsible for our troubles is Adolf Hitler, and we are dealing with the most powerful and best organized system of world aggression ever devised by mankind," he said.

He reported that T. L. Preble, T & M vice president for the Society's T & M Activity, was now working on a maintenance manual for operators. "The problem of intensifying use of vehicles is not so serious among fleets, but the huge number of trucks owned and operated by individuals. This, in terms of registration, is 80% of our problem."

H. E. Simi, Twin Coach Co., chairman of the Truck & Bus Session, with E. S. Pardoe, OPM, and Errol Gay, in charge of the SAE T&M Subcommittee on Engine Deposits – Their Prevention and Removal, who presented a progress report at the National Transportation & Maintenance Meeting, Nov. 13 and 14, in Cleveland

Drawing largely upon Britain's experience, Commissioner Rogers said that America was forewarned, and the Committee's studies of coordinating manufacturing and servicing of vehicles, intensifying use, and other major problems were each designed to turn a billion-dollar peacetime industry into an "arm of national defense."

"We hope the OPM priorities orders will fulfill their purpose, but there is some indication that a system of allocation has already been initiated to help achieve the purpose of supplying defense needs on the one hand and getting materials and parts to the indirect-defense fields on the other.

"Now is the time to think through these problems. Later, it may be too late to do any planning," he warned.

Commissioner Rogers is counting heavily on the ingenuity of the automotive manufacturing and operating executives. He called attention to the difference between "preparedness" and "hoarding" and urged operating executives to refrain from piling up abnormally large stocks of equipment and parts.

E. S. Pardoe, on leave from the Capitol Transit Co. to the Automotive, Transportation, and Farm Equipment Branch, OPM, urged all operators to try to solve their problems in so far as possible before resorting to OPM for relief.

Commissioner Rogers hoped operating executives would keep in mind:

Safety in operation,

Adequacy of transportation to and from new defense plants, cantonments, and enlarged plants,

Supplies of gasoline, and

Closer cooperation of transportation executives with regulatory officials and shippers.

"This is a challenge to the best thinking of you engineers in the SAE," he concluded.

Mr. Preble assured the Commissioner that the SAE had heard the challenge, and had accepted it. "We have begun a wide scope of studies, and already we have been able to report genuine progress," he said. Randolph Whitfield, Georgia Power Co., and Chairman of the session, opened the meeting to off-the-record questions and answers about a wide variety of defense problems in connection with motor transportation.

Aviation Program Under Full Steam, Fuels & Lubricants Meeting Is Told

(*Continued from page 24*)

while fuels for high-speed diesel engines of today sacrifice none of the advantages of safety or economy characterizing diesel fuels in general, the high-speed design requires a fuel which has been more carefully selected and more critically controlled if it is to meet all the requirements of the high-speed diesel engine.

In comparing the present CFR method using the mechanical coincident flash method and the Penn State magnetic pick-up for the coincident flash method with the Caterpillar cetane valve, they find that a wide variety of fuels when measured by the Caterpillar method shows exceptionally good correlation with the CFR engine method (throttle valve) will give slightly higher ratings than the CFR delay method on lower cetane value fuels, and that it will give slightly lower ratings on higher cetane value fuels.

They pointed out that while the Caterpillar method permits determination of cetane number of an unknown fuel in about one-third the time required for the CFR engine method, the latter method is likely to remain the standard for some time because most companies have the converted CFR engine, whereas only a relatively small number of companies now have the Caterpillar equipment.

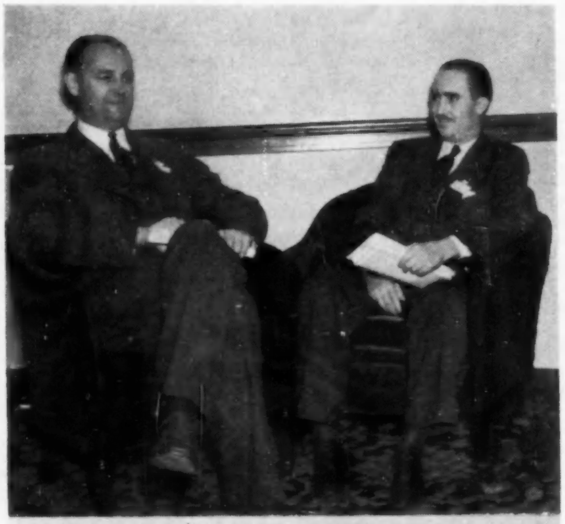
The authors recommend consideration of the ASTM diesel fuel classification grade 2-C for general distribution, especially to the Army, and they have included a cetane number of 45 minimum. The Army agrees fairly well with this but prefers a 47 minimum cetane number.

The best overall oil performance, so far as piston deposits indicate, is obtained from oils made from Coastal naphthenic crudes, A. E. Smith and J. P. Stewart, Socony-Vac-

uum Oil Co., reported in discussing "Diesel Lubricating Problems." The next best results are obtained from oils from Mid-Continent crudes, while the worst piston deposits result from using Pennsylvania oils, they have found. Pennsylvania oils give less lacquer than those from other crudes, and distillate products give better results in so far as deposits are concerned than do those containing residual products. The effect of refining treatment on lacquer deposits does not follow a distinct pattern and the broad assumption that a distillate product is superior to a residuum in lacquer formation does not appear to be justified. Blending of naphthenic and paraffinic stocks to obtain the best properties of each shows that these blends are in general superior to those obtained with the individual distillates.

Paraffinic residua and distillates appear to have substantially the same load-carrying capacity, and both have a higher capacity than do naphthenic oils, they found. Addition of paraffinic stocks to increase the load-carrying capacity of naphthenic oils does not follow a clear pattern, some stocks improving this capacity while others do not. More work is necessary to determine the effect of various refining methods on oil properties, they concluded.

Wide use of compounded lubricants has caused considerable difficulty in the use of oil filters, Dr. Ulric B. Bray emphasized in discussing this subject. Some filters remove addition agents, wholly or in part. However, a new type of filter is now in use which permits maintenance of clean oil and yet does not remove the addition agent, he said. "The use of an efficient filter," Dr. Bray concluded, "should not be taken as justification for extending the oil drain period beyond reason." He urged the regular replacing of oil at proper intervals, and changing the filter element at oil change periods as the best system for obtaining the greatest returns from good oil filters.



(Above) Prominent among authors and discussers at the Tulsa F&L Meeting were (seated, left to right) Dr. Ulric B. Bray, Los Angeles, who delivered a paper on Oil Filters and presided at the student debate; G. C. Richardson, chairman of the opening session, and Frank A. Suess, co-author of a paper on Changes in Oils and Engines. (Standing, left to right) C. S. Hansen was chairman of the Friday morning session; W. G. Ainsley, a co-author on Cetane Number Studies Friday morning; A. L. Heintze was chairman of the Thursday afternoon session; R. C. Alden presented a provocative paper on the Sligh apparatus, and A. E. Smith was the co-author of a paper on Diesel Lubrication

East met West (center) when G. H. Cloud, Linden, N. J., diesel fuel research engineer, Standard Oil Development, discussed fuels with Bert H. Lincoln, Okla., a co-author of the paper on Changes in Oil and Engines

L. Skelly, left, relaxes with Carl A. Tangner, president, GM Diesel Power & Machinery Co., Oklahoma City

The greatest development in oil cooler systems has been in aircraft-type coolers, H. F. Brinen, Young Radiator Co., said in a paper delivered by L. Skelly of the same company. These new types of equipment function to handle oil warm-up and temperature stabilization as well as cooling the oil. The newest coolers will cool adequately the oil in engines of 2000 hp and more, Mr. Brinen stated. A very comprehensive summary of different types of cooling systems was presented, including types from the smallest to the largest. Dr. U. B. Bray in discussing the paper pointed out that in more severe service this type of equipment is needed as an integral part of engine design, and that the best way to approach the problem of oil stability and longer life is to cool the oil well below the critical temperature.

John A. C. Warner, SAE secretary and general manager, brought home strikingly the value of SAE membership and what the Society does for its members. He showed how huge a job is being done for National Defense by the members of the Society and by the organization itself. In aeronautics alone 37 committees including 146 men have averaged a committee meeting every two-and-one-half days in coordinating and pushing that program forward.

Debate

A regular feature of the F & L Meetings in Tulsa is a student debate on some pertinent question. B. E. Sibley, father of the idea, arranged for teams from the College of Engineering, University of Oklahoma, and the Division of Engineering, School of Technical Training, Oklahoma Agricultural and Mechanical College. The former took the affirmative and the latter the negative side of the question: "Resolved—Passenger Car Crankcase Draining Every 1000 Miles Is To Be Preferred Over Draining at Greater Distances." Men making up these respective teams were: Vance Cameron, Don Malvern, Lynn C. Nordahl, and Harry Musser, under the direction of Prof. L. E. Haas, University of Oklahoma team, and the Oklahoma Agricultural and Mechanical College team was composed of E. C. Bright, W. C. Buck,

Raymond Dowell, and Howard Webb, under the direction of Prof. R. G. Hilligos. The negative side of the question won the decision of the judges although they admitted they were still in doubt as to the correct time for oil changes.

Notes:

A threat of pneumonia was responsible for keeping B. E. Sibley in bed at his home in Ponca City, Okla., and prevented his attending the meeting. Following the announcement of his illness at the open session, a motion to send him a message of cheer and regret at his absence was unanimously adopted by a rising vote.

A crowd of flying cadets from the Sparton School of Aeronautics in Tulsa was present at both evening sessions though prevented from more active participation by classroom and flying time requirements. Many of these men are graduates of the U. S. Military Academy, West Point.

WEST COAST T&M Sessions Stress Problems of the Immediate Future

(*Continued from page 25*)

Thursday afternoon session. Presiding at the opening session and serving also as toastmaster at the Thursday evening banquet was Northern California Section Chairman, Charles F. Becker. Howard Baxter and Sam Bogart of the Northern California Section presented native son A. T. Colwell with a remembrance of his early California upbringing.

Presiding at the opening session Wednesday morning Nov. 5, Northern California Chairman Charles F. Becker, Tide Water Associated Oil Co., introduced SAE President A. T. Colwell, SAE Secretary and General Manager John A. C. Warner, and other out-of-town members present for the T&M two-day sessions.

The object of the Transportation & Maintenance Activity was stated by Sidney B. Shaw, Pacific Gas & Electric Co., as follows: to "evolve for transportation and maintenance engineers a concise manual of new and/or usable information to help them to get more automotive transportation out of a dollar." A further objective is to collaborate with National Defense agencies and to assist them in every way possible with automotive transportation and maintenance engineering advice and information.

Peter Glade, Purity Stores, Ltd., who, together with Mr. Shaw, arranged the series of meetings, pointed out that the term "engineering" includes usage as well as design, and said: "That's where T&M comes into the picture."

▪ Bearings

Leading off the first day's morning session, Albert B. Willi, Federal-Mogul Corp., presented an illustrated paper on "Bearings for Heavy-Duty Automotive Engines." Increase of load requirements, especially at higher temperatures, necessitated the development of cadmium, copper and lead alloy bearings, stated Mr. Willi. However, he added, it is necessary to thoroughly understand the peculiarities and characteristics of each.

In 1932 a babbitt thinness of 0.005 in. was considered desirable but could not be obtained with manufacturing methods then developed. At the present time, steel-back babbitt-lined bearings with lining as thin as 0.0025 in. are being produced and used in several engine models, stated Mr. Willi. In presenting test results which showed an increase in bearing life for the thin linings, Mr. Willi pointed out also the danger of too thin a babbitt – inability to imbed and absorb a reasonable number of foreign particles (cast iron chips, road dirt, etc.), and said that when thin linings are "burned out" the extra clearance between shaft and back may cause insufficient noise to warn the operator that something is wrong. Adequate oil filters and air cleaners are therefore an absolute necessity for thin bearing jobs, he added.

Reground crankshafts and undersize bearings are probably used for many more miles per engine than standard shafts and bearings, said Mr. Willi, adding, however, that use of thinner linings is not possible for such replacement service due to necessity of holding steel-back thickness constant in the modern high production continuous steel strip bearing making machines.

Higher load capacity bearings should be used for replacement than for original equipment, continued Mr. Willi in discussing bearings from the repair man's point of view. Unless the engine is thoroughly rebuilt at time of bearing change, such items as worn and out-of-round crankpins and journals, a bowed crankshaft retained in line by force thus imposing a heavy and false load on the main bearings, a warped crankcase with the main bearing saddle bores out of line, out-of-round connecting-rod bores and main bearing saddle bores, excessive amount of dirt within the engine picked up due to carelessness, bearing caps misplaced sidewise, and improper oil clearance, all work to decrease the life of replacement bearings, he stated. Detonation due to incomplete removal of cylinder radiator and engine water jackets, low oil pressure due to wear and excessive clearance at bearings and in oil pump, are other items which cause decreased life of bearings, Mr. Willi added. Even in spite of the above handicaps a bearing life equal to or even greater than the original equipment can be obtained by use of cadmium silver copper bearings as replacements.

As the result of a recent cross country survey, Mr. Willi stated that copper-lead bearings have been marvelously successful in some operations, and in others very unsatisfactory. Their properties and characteristics must be better understood so that they may be applied within their particular limitations.

To aid maintenance men in handling copper-lead bearings, Mr. Willi suggested fifteen specific rules.

Answering questions from the audience, Mr. Willi offered the following information: Pressure distribution within a bearing drops at the point where the annular groove is cut, so the designer must take this into consideration when figuring bearing capacity.

The initial high rate of wear of copper-lead bearings which occurs before protecting lacquer is formed is not serious unless bearing clearance is too small.

Recent European use of annular oil grooves on back side of bearing, with oil transfer holes through to inside, is a matter of individual preference.

Detonation, if sufficient to be distinctly noticeable, approximately doubles the load on bearings.

The use of special additives or break-in oils by individual operators is an item to be checked into very carefully since certain additives under high temperature conditions cause very rapid corrosion failure of bearings.

▪ Selecting Lubricants

Dr. R. I. Stirton, Union Oil Co. of Calif., in a paper, "Factors Influencing the Choice of Heavy Duty Lubricants," prepared by himself and C. C. Moore, Union Oil Co. of Calif., said that, despite the great amount of work done by petroleum technologists, the only really conclusive method of selecting the best motor oil for a certain engine service is by comparison of the oils in the actual service, and pointed out that the field of possible oils can be narrowed by eliminating those known to cause trouble in any particular application. For instance, Dr. Stirton suggested, in selecting lubricating oil for use in a high-speed high-temperature, heavy-load trucking service, the procedure might be as follows:

Red engine oils are known to be unsatisfactory for this service and can be eliminated. If bearings are either copper-lead or cadmium silver, oils known to cause corrosive attack can be eliminated. Similarly if, in a particular engine, ring sticking or piston lacquering is a problem, oils known to improve these conditions should be selected. Thus, Dr. Stirton said, the field of possible oils is narrowed to perhaps two definitely superior oils and possibly a cheaper one almost as good.

The small operator, he continued, should choose between the two superior oils, trying each one out. On the other hand a large operator would no doubt put one of the superior oils in half his fleet, the other superior oil in most of the remaining half, and try out the cheaper oil in just a few units. Final choice, therefore, would be made after carefully conducted tests under actual conditions.

To secure the information on which to base his comparison of oils, Dr. Stirton advised operators to request data from the refiner's research department as to test results on the particular oil under actual conditions similar to those the operator is encountering. It may be necessary, he added, to recheck lubricating oil selection from time to time to keep up with latest developments.

In discussion of this paper, one operator suggested that, since physical properties of lubricating oil are not an indication of their suitability, especially when additives are present, the phrase "the oil must lubricate this equipment satisfactorily" be included in purchaser's specifications.

▪ Taxicabs

"We could not exist and operate profitably, were it not for a system of preventative maintenance," stated Lewis A. Schroyer, Yellow Cab Co. of California, in presenting data on the operation and maintenance of a fleet of 50 cabs. While maintenance costs are secondary to driving-personnel payroll, they are non-productive of revenue and therefore get considerable attention, he explained.

The old built-for-the-purpose type of taxicab cost approximately $3500, weighed nearly 6000 lb and with a gasoline mileage of 8 mpg was costly to operate, stated Mr. Schroyer, comparing them with the present type of light commercial vehicle which costs $1500, goes 12 mpg, and has an excellent turn-in value after three years and 125,000 miles of city driving. Rubber bushed rear spring shackles, and a governor set at 50 mph, are other features of present cabs.

Describing in considerable detail the various maintenance operations through which each cab is put every six weeks, Mr. Schroyer pointed out the following practices as of considerable importance:

Because of standardization of cab units, parts and supplies can be obtained from local dealers and a large stock room is unnecessary. Lubricating oil drained from crankcases at 3000-mile intervals passes through two settling tanks, an oil reclaimer, and two more settling tanks before being re-used. Hypoid lubricant in the differentials is changed at 10,000-mile intervals. Construction and use of special tools proved to be more rapid than the use of universal tools in servicing cabs. Heavy-duty wide rim six-ply tires are supplied and maintained by an outside company and have an average life of 30,000 miles. In making repairs, unit exchanges are made wherever possible to lessen cab time in shop.

Mr. Schroyer concluded his paper with the statement that maintenance costs are 0.006¢ per mile.

Questioned by Howard Baxter, Winslow Engineering Co., concerning use of tap water in batteries, Mr. Schroyer replied that, upon advice of the battery manufacturer, this can be done in this particular area. Sid Shaw, Pacific Gas & Electric Co., added that samples of water should be analyzed in each particular instance before use in batteries.

In response to a series of questions by Mr. Shaw, the following data were added by Mr. Schroyer:

Mileage between six-week checking periods runs 2000 to 3000. Fixed crews of one or two men each do the maintenance work with the cars moving along as on an assembly line. Battery life averages eighteen months. Clutch plate runs 25,000 miles average before replacement.

Although the Yellow Cab Co. hoists its units when doing maintenance work, Technical Chairman Fred C. Patton, Los Angeles Motor Coach Co., pointed out that on bus maintenance this may be too time-consuming.

The maintenance cost of 0.006¢ per mile is for parts and maintenance labor only, said Mr. Schroyer in answer to a question by Sal Onorato, Union Oil Co., adding that the total cost of cab operation including depreciation and drivers' time is 16¢ per mile.

▪ Chromium Plating

Chromium plating is a means for improving motor equipment service, said Ellis W. Templin, Department of Water and Power, City of Los Angeles. The pressure to "keep them rolling" under defense conditions will compel maintenance men to revise their procedures and in many cases force them to rebuild old parts because of impossibility or delay in getting replacements. The use of chromium plating is a method of salvaging worn parts which may actually extend their life beyond that of new parts, stated Mr. Templin.

Although many maintenance men do not know of the benefits possible with chrome plating, he said, a recent survey indicates that the following parts have been plated either as a renewal measure or for prevention of wear or corrosion: Engine cylinder walls, crankshafts, camshafts, piston rings, cylinder sleeves, water pump shafts, piston pins, rocker arms, eccentrics, chassis shackle pins, shaft splines, brake camshafts, universal joint journals, ball and roller bearing seats, differential spider journals, steering pivot arms, tractor track roller thrust washers, bearing seal washers, air tool equipment, drills, taps, reamers, gages, facing of forming tools, pump liners, pump plungers, plug valves, printing rolls and plates, capscrews subject to removal, and almost every moving part in hydraulic control system in aircraft.

The following are typical reports obtained by Mr. Templin as part of the survey mentioned above:

Sidney B. Shaw, Pacific Gas & Electric Co., reported a very great decrease in wear after chrome plating of cylinder liner in a truck in service since 1933. Wear on the chrome plated liner was only 0.001 in. as compared to 0.009 and 0.010 in. for other liners in same engine.

A. T. Stahl, International-Plainfield Motor Co., reported that plating thicker than 0.010 to 0.015 in. is not economical, and that best results had been obtained by plating chrome directly on steel rather than over an initial copper plate as had previously been done.

J. B. Fisher, Waukesha Motor Co., reported that under extremely dusty conditions chrome plated sleeves showed one-twentieth as much wear as good grade chrome-nickel iron sleeves. Ability of plating companies to apply chromium evenly and at a reasonable price has been the greatest drawback, according to Mr. Fisher's report.

F. P. Frankford, Fifth Avenue Coach Co., reported that his company has chrome plated the bores of 450 engine cylinder blocks with profitable results.

Mr. Templin, quoting from A. Mankowich's article in *Metal Finishing* for June, 1941, outlined and reviewed the technique and details of chromium plating. Most economical or practical thicknesses for various services, grinding of metal prior to plating, plating solutions – including effect of temperature, hardness of base metal, and a sequence of operations were the items covered in detail in this review.

▪ Chrome Hardening

For the second part of the presentation on chromium plating, Mr. Templin read a paper by H. van der Horst, van der Horst Corp., dealing with chrome-hardening of cylinder liners. Pointing out the various items which affect cylinder wear, Dr. van der Horst mentioned the following: sand or dirt in air intake, dirt in the fuel, water in either air or fuel, any amount of sulfur above 2% in the fuel, and use of extremely heavy fuel such as in motor ships.

Many experiments and many installations – well over a hundred thousand engines, chiefly diesel – show that chrome plating of cylinder bores is an excellent remedy for wear due to any of the above causes, he stated. Not only is cylinder wear decreased by chrome plating, but wear of cast iron piston rings is decreased to about one-fourth when running in chrome barrels. Reduction of lubricating oil supply, he continued, sudden change of cooling water temperature from 200 F to 50 F, and elimination of oil cooling to piston so as to cause piston seizure, were the conditions imposed on a 14-in. bore – 14-in. stroke test engine in checking performance of chrome plating. Reduction of wear in an 11 in. bore marine engine from 0.014 in. per 1000 hr, to 0.015 in. per 28,000 hr, was one of the many instances cited by Dr. van der Horst.

Concerning the application of chromium to cylinder liners, Dr. van der Horst had the following to say: The electrolytic coating must adhere perfectly; the thickness of the coating must, within limits, be equal all around and from top to bottom; there must be no tiny ridges for the piston or the rings to run against; the ordinary bright, dense coating of chromium is not suitable as it does not hold lubricating oil; and in order to hold oil, it is essential that the chromium be very porous. Dr. van der Horst sees important application of chrome-hardened engine parts within the next year.

In discussion, Joseph Geschelin, Detroit technical editor, Chilton Co., pointed out that in this country chromium plating is more than ten years old, and has been used not only for replacement parts but for original automotive equipment and for metal cutting tools. He emphasized the difference between ordinary chromium plating and the kind required for cylinder liners, adding that the technique is of extreme importance and is generally covered by patents.

Hans Bohuslav, Enterprise Engine Co., called attention to the undesirability of chromium plating for some services such as parts exposed to salt water and parts subject to high unit pressures.

A. T. Colwell, Thompson Products, Inc., stated that chromium plating of large gun barrels is showing good results.

To Mr. Bohuslav's question, "Do chromium plated rings work equally well on cast iron and steel liners?" Mr. Colwell stated that the chromium plated rings now available were developed for use on steel liners of aircraft engines.

Touching on the economics of chromium plating, Sid Shaw pointed out that after plating, increased time between overhauls pays for cost of plating, especially when the unit is in heavy service or the part inaccessible.

▪ Engine Manufacturer

A feature of the Wednesday afternoon session was a sound motion picture of Wright aircraft engine manufacture. Starting with recent plant expansions, the camera tour proceeded through engineering department, foundry, machine shop, assembly floor and test stands, giving many interesting shots of individual parts and sub-assemblies.

▪ Compulsory Inspection

"First and last, compulsory motor-vehicle inspection is a safety project," said J. Verne Savage, superintendent, Municipal Shop Motor Vehicle Inspection Station, City of Portland, in presenting a summary of the benefits resulting from compulsory motor-vehicle inspection.

Brakes, steering mechanism, and lighting equipment are the most important items, followed by regularly required accessories such as horn, windshield wiper, muffler and exhaust system, stated Mr. Savage, pointing out that regular check by state or city personnel of those items which vitally affect safety of vehicles follow right along with, and are comparable to requirements imposed upon railroads, steamships, and operators of boilers, elevators.

Mr. Savage called attention to the fact that it is momentum – product of weight times speed – that automotive engineers and operators are concerned with when considering proper braking effort. Braking time is di-

Some Speakers and Chairmen at West Coast T&M Meeting

1. Howard A. Reinhart, consulting engineer, Sacramento

2. Peter Glade, Purity Stores, Ltd.

3. Dr. R. I. Stirton, Union Oil Co. of Calif.

4. Lee Ketchum, Six Robblees, Inc.

5. E. W. Templin, Los Angeles Department of Water and Power; Joseph Geschelin, Detroit Technical Editor, Chilton Co.; and Fred C. Patton, Los Angeles Motor Coach Co.

6. Hamilton Migel, Magnaflux Corp.

7. Albert B. Willi, Federal-Mogul Corp.

8. J. Verne Savage, City of Portland, Municipal Shops and Motor-Vehicle Inspection Station

rectly proportional to speed; and braking distance is proportional to square of speed, pointed out Mr. Savage in presenting the following somewhat arbitrary formula which gives results close enough for most purposes: stopping distance in feet

$$= \frac{\text{Square of speed in mph}}{\text{Braking effort in ft-lb} \times 30}$$

Equalization of brakes is of great importance, he said, when it is noted that skidding is caused by braking effort being in excess of the coefficient of friction between wheels and road.

The large fleet operator has less need for and derives less benefit from compulsory inspection than any other class of vehicle owner, stated Mr. Savage, attributing this to the critical inspection and preventive maintenance program of each operator. However, even the best run fleets can and do benefit from regular check of their vehicles by keeping their mechanics more alert, he added.

Any movement which tends to make possible greater running speeds without impairing safety is certainly advantageous to the fleet operator and is reflected in more revenue per operation hour, stated Mr. Savage in conclusion, pointing out that economy of transportation should not be based upon a ton-

mile basis, but rather upon a ton-mile per hour basis.

Detailed operations of the Portland inspection station were shown in a series of colored slides.

In answer to questions by Sid Shaw, Mr. Savage stated that normal hours of inspection are 8:00 a. m. to 8:00 p. m. six days a week, but that other times can be arranged. Inspection itself requires 6½ to 7 min and waiting time is 10 to 15 min. Inspection is required each six months of all cars entering the city of Portland. If compulsory inspection were more widespread, Mr. Savage urged, insurance rates could be considerably reduced.

▪ Butane

Improper installations, poorly designed equipment and insufficient control of mixture temperature and spark settings are the handicaps under which butane is working, stated Howard A. Reinhart, consulting engineer of Sacramento, in opening his paper "Butane Fuel for Automotive Engines." Debate as to relative merits of butane, propane and mixtures thereof compared to gasoline or diesel fuels must wait until problems of performance and fundamental design are satisfactorily worked out, he said, adding that full knowledge of those problems may require years of development.

Using either butane or gasoline, a properly designed system will show 10 to 20% better power, economy and performance than a poorly designed one, stated Mr. Reinhart, adding that proper air-fuel mixture and correct spark advance are the most important items. Too lean a mixture results in a decrease in power, increase in temperature and rapid damage to pistons, valves and spark plugs, he stated, adding that truck installations of the future will need to control the temperature of the fuel vaporizer, and will need to interconnect spark setting and mixture ratio settings. Present mixers will need to be greatly improved, spark-plug voltages increased 3 to 5 times their present value, and special manifolds developed, stated Mr. Reinhart in closing his paper.

To a question by E. E. Charles, General Petroleum Corp. – "Are heat exchangers or mixers most responsible for erratic operation?" – Mr. Reinhart replied that lack of *control* of heat exchanger is the real issue.

▪ Supercharged Motors

Supercharged diesel engines have been the Pacific Coast truck operator's answer to increased competition, stated Peter Glade, Purity Stores, Ltd., in opening his paper on "Supercharged Motors for Highway Transport." Reduction of terminal to terminal time – with a resultant greater amount of working hours from any given equipment – has made this possible, he added.

Present supercharger units themselves are extremely satisfactory and there is every reason to believe that they will operate for long periods of time with a very minimum of service work being necessary, said Mr. Glade, adding that present problems have to do with application and effect of supercharger on the engine, rather than with the superchargers themselves – improper installations, improper servicing, some trouble with materials and parts, and with lubricating oils. The Roots type blower furnished as standard equipment on the Cummins truck engines operated by Mr. Glade is driven by three belts from the front of the crankshaft, produces a pressure boost of 3 lb per sq in. at operating speed of 1400 rpm, and weighs approximately 250 lb.

Physical size and weight of the supercharger presents a problem in making economical installations in commercial vehicles, said Mr. Glade, pointing out that the use of a larger and therefore heavier radiator is also necessary. Trouble with parts, materials and lubricating oils is tied up very closely with this matter of adequate radiators and oil coolers, he explained.

In regard to fuel consumption, the mileage per gallon is somewhat less. However, the additional performance and increased power enable transport trucks to move at a considerably faster average speed, particularly where many grades are encountered – and, since time is also involved in cost per mile, this more than offsets the difference in fuel consumption and places the supercharged engine well over the non-supercharged engine on an economical basis, stated Mr. Glade.

No broad rules can be stated regarding the application of superchargers to every highway transport truck use, but each installation must be treated individually – its requirements and operation problems must be carefully analyzed, said Mr. Glade, concluding with the statement that the present background knowledge of supercharged engine installations and operations gives assurance that this type of powerplant is here to stay.

Answering a question as to which engine parts can be expected to fail as a result of changing from standard to supercharged operation, Mr. Glade replied that pistons and exhaust valves appear to be the most likely casualties.

Asked about lead sweating of copper lead bearings in Diesel engines by Albert B. Willi, Federal-Mogul Corp., Mr. Glade listed three causes, as: insufficient oil clearance, corrosion due to lubricating oil, or too soft a shaft.

▪ Magnaflux

A lower factor of safety can be used today because of the improvement in our inspection technique, stated Hamilton Migel, Magnaflux Corp., in opening his paper, "Magnaflux Inspection in the Automotive Field." Better inspection is especially important with our present trend to increased loadings and decreased weight and cost, he continued.

The Magnaflux method is an inspection system designed for just this type of work, Mr. Migel pointed out, adding that it is simple, rapid, conclusive, non-destructive, and affords easy means of revealing cracks or any other defect which may be located at or near the surface of steel or its magnetic alloys. In describing the detailed procedure of testing parts, Mr. Migel described the inspection medium – fine powders in a choice of colors and used either dry or in oil; type of current – either direct or alternating; timing of current – depending upon magnetic characteristics of part tested; amount of current – depending upon size of part; direction of current – so as to flow across expected flaw; and demagnetization – where necessary such as for aircraft parts.

Rejection of defective material is one result of using Magnaflux methods in factory work, while another is the discovery of grinding and heat checks, stated Mr. Migel, adding that Magnaflux inspection in overhaul and repair work is increasing rapidly.

In discussion, it was pointed out that the greatest draw-back to the use of Magnaflux by maintenance men is its high first cost and license fee arrangement; also the Magnaflux company's unwillingness to allow installation of equipment in a commercial testing laboratory which could be used by anyone on a simple cost basis.

▪ Round Table

The Thursday afternoon session wound up with a round table discussion of considerable interest. E. W. Templin, Department of Water and Power, Los Angeles, warned that there is a coming need for road transport pools to handle traffic under war conditions ahead, and that transportation and maintenance men will need to prepare for such operations. Mr. Shaw reported that an SAE committee is working on this problem at the present time.

A question about experience with building up parts by metal spray brought responses from several of those present. One speaker suggested use of such spray to build up shafts to use standard-sized replacement bearings, but another argued that it would be cheaper to stock oversized bearings than to build up shafts.

Talk of compounded oils in gasoline engines brought the suggestion that use of such oils for heavy-duty diesels may be an economic waste; and that in the future there may be three oils – one for gasoline engines in light service, one for gasoline engines in heavy service, and one for heavy-duty diesels.

Reported also during the round-table discussion were increased use of wider-rim wheels; experience indicating that synthetic rubber tires show the same rate of wear as natural rubber after a year of service; and the fact that laws will soon be enacted to reduce smoke on highways.

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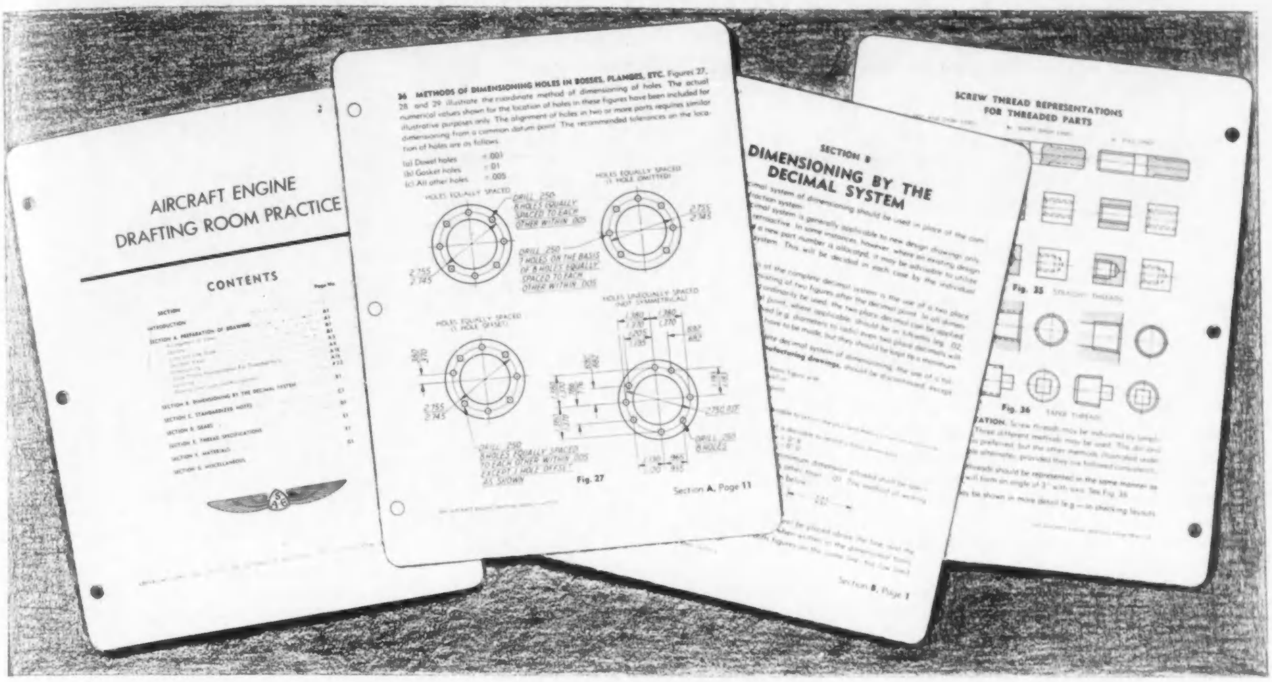
TYPICAL PAGES FROM HANDY NEW AIRCRAFT DRAFTSMAN'S MANUAL



New Aircraft-Engine Drafting Manual Standardizes Drawing Room Practices

FINAL version of the SAE Aircraft-Engine Drafting Room Practice manual has just come off the press, terminating thousands of hours of careful planning by Subcommittee E-8 of the Aircraft-Engine Subdivision, Aeronautics Division, SAE Standards Committee.

Rapid expansion of the aircraft industry has made imperative the development of a standardized procedure for the preparation of aircraft-engine mechanical drawings. The new SAE Aircraft-Engine Drafting Room manual is designed to establish a common drafting room "language" so that signs and symbols appearing on aircraft engine blueprints may be uniformly interpreted by manufacturers, subcontractors, and vendors. It embodies the best in common practice, as well as heretofore controversial practices on which agreement has been obtained.

In compiling the present edition, the committee consulted all available manuals, both published and unpublished, prepared by the Air Corps and various engine manufacturers to suit their individual needs.

Contents of the manual include, among other vital items, all-essential data on preparation of a mechanical drawing, such as arrangement of views, lines and line work, sectional views, dimensioning, screw thread representation for threaded parts, lettering, and drawing sheet sizes and arrangement. Dimensioning by the Decimal System is an important new method described. Amply covered also are: uniformity of drawing sizes, accordion folding of blueprints and arrangement of number blocks so that they may be filed without confusion, standardized symbols of cross-sections, a uniform system of symbolized general notes for concentricity, parallelism, circularity, case hardening, and similar definitions, and decimal designation for material instead of by gages, names and numbers (sizes will be determined from commercially procurable materials). Included in the manual is a handy sheet of abbreviations and symbols for use on drawings.

The 37 pages of illustrative drawings and text of which the manual is composed are produced in loose-leaf form. As Subcommittee E-8 will continue to function and develop additional material for inclusion in the manual it was felt the loose-leaf form will enable aircraft-engine draftsmen to incorporate extra printed pages as soon as the approved data are available, and thus avail themselves of a continuously up-to-date data book. Cut to fit any standard type of file binder, the pages are made of heavy paper for extensive, practical use. Copies of the complete manual are priced at $1.50.

On the introductory page of the manual are these explanatory words: "It is believed that this manual forms a sound basis for a uniform and standardized drafting (room) procedure which, followed by aircraft-engine draftsmen, will result in the preparation of drawings complete in their attention to the detail necessary if they are to serve their intended purpose – a common language for all who must use them."

The Men Behind the Manual

ARTHUR NUTT, Chairman Aeronautics Division	VAL CRONSTEDT, Chairman Aircraft-Engine Subdivision

Subcommittee E-8 Drafting Room Practices

J. G. PERRIN, Chairman Pratt & Whitney Aircraft	C. R. REYNOLDS Allison Division, GMC
J. H. CARPENTER Lycoming Division Aviation Mfg. Corp.	P. V. RICHARDS Wright Aeronautical Corp.
R. S. KELLOGG Packard Motor Car Co.	J. G. SCHWEIGER Ranger Aircraft Engines

About SAE Members

BRAYTON WILLIAM WESTCOTT was recently named vice president, Holley Carburetor Co., Detroit. Formerly he was sales manager of the company.

ROBERT C. WALLACE, who has been for several years assistant chief engineer of the Marmon-Herrington Co., Inc., Indianapolis, has been upped to director of engineering; while **GEORGE H. FREERS,** one of the Indiana Section's two-time chairmen, who joined the Marmon-Herrington engineering department last year, has been named chief engineer.

WILSON C. BRAY recently left the Tire Division, B. F. Goodrich Co., Akron, Ohio, where he was assistant sales manager, to become general sales manager, Ferguson-Sherman Mfg. Corp., Dearborn, Mich., designers and sales agents of Ferguson-Sherman tractors.

ROBERT J. HOWISON has left Whitney Chain & Mfg. Co., Hartford, Conn., where he was general sales manager, to become sales manager, Automotive Division, Morse Chain Co., Detroit.

The Young Radiator Co., Racine, Wis., recently announced that **ROBERT GRANT** has joined the company in a production and managerial capacity. He leaves the position

In Managerial Job

Robert Grant

of executive vice president, Fuller-Johnson Corp., Detroit. Mr. Grant was educated at Cornell University, and received his early production training as a line superintendent for the Nash Motors Division of Nash-Kelvinator Corp., at Kenosha and Racine, Wis.

GWILYM WILLIAMS has left James & Co. (Birmingham), Ltd., Spun Cast Works, Halesowen, England, where he was director, works manager, and secretary, and is now engaged as manager, Technical Department, Cylinder Components, Ltd., Birmingham.

H. D. BUBB, JR., has been advanced from chief engineer to director of engineering, Thompson Products, Inc., Cleveland.

J. W. OSTHEIMER, formerly managing director, Cie. Franco-Americaine des Jantes en Bois, France, has been in the United States since Aug. 7. He is staying at the Barbizon Plaza Hotel, New York.

The Vermilye Medal, awarded biennially by The Franklin Institute in Philadelphia in recognition of outstanding contribution within the field of industrial management, has been awarded this year to **WILLIAM S. KNUDSEN,** director general of OPM. The award to Mr. Knudsen is made "in recognition for long years of outstanding managerial ability in American industry, characterized by brilliant initiative, far-seeing vision and human understanding, culminating in invaluable service to his country in the administration of unprecedented production for national defense." Presentation of the medal will take place at a dinner given in Mr. Knudsen's honor at The Franklin Institute, Dec. 1.

H. A. ZELLER, sales engineer, Diesel Division, General Motors Overseas Operations, New York City, has been transferred to General Motors India, Ltd., Bombay.

Charles H. Chatfield
Elected Secretary

CHARLES H. CHATFIELD formerly director of research and executive assistant in the head office, United Aircraft Corp., East Hartford, Conn., recently was elected secretary of the company.

The Brake Lining Manufacturers' Association, Inc., at its annual meeting in New York City recently, elected two SAE members as officers for the coming year. **THOMAS L. GATKE,** president, Gatke Corp., was elected first vice president and member of the association's Executive Committee; and **PAUL B. HOFFMAN,** assistant vice president, sales, American Brakeblok Division, The American Brake Shoe & Foundry Co., was elected second vice president and a member of the association's Executive Committee.

FRED B. HUNT has been named sales engineer, Continental Oil Co., Ponca City, Okla. Previously he was a mechanical laboratory assistant with the same company.

NELS E. NYLIN recently joined the Timm Aircraft Corp., Van Nuys, Calif., as equipment and machinery designer. Formerly he was design engineer, Machinery Mfg. Co., Division of Machinery Sales Co., Vernon, Los Angeles.

GROVER N. CONLEY, JR., engine tester, Wright Aeronautical Corp., recently was transferred from the Paterson, N. J., plant to the company's new plant in Lockland, Ohio.

QMC Motor Division Chief

Col. Herbert J. Lawes

COL. HERBERT J. LAWES, commanding officer of Holabird Depot, Quartermaster Corps, has been appointed acting chief of the Quartermaster Corps' Motor Transportation Division. Col. Lawes for some 16 years has specialized in the operation and maintenance of motor transportation.

HARRY L. BILL has left United Aircraft Products, Inc., Dayton, Ohio, where he was president, to become vice president and general manager, Greenfield Tap & Die Corp., Greenfield, Mass. Mr. Bill has had experience in both the automotive and aircraft fields, having been connected with the Corbin Motor Vehicle Co., New Britain, Conn., as factory manager, with Chalmers Motor Co., and with the Bendix Aviation Corp. subsidiary, Pioneer Instrument Co., Brooklyn, as vice president and general manager.

C. R. ARMBRUST recently joined the engineering staff of the Willys-Overland Motors, Inc., Toledo. Formerly he was transmission engineer, New Process Gear Corp., Syracuse.

HAROLD H. McMAHON has resigned as chief project engineer, Menasco Mfg. Co., Burbank, Calif., to become a sales engineer, Aircraft Accessories Corp., Burbank.

EDWARD DUNNING has been named assistant manager, Products Application Department, Shell Oil Co., Inc., N. Y. Formerly he was engineer in charge of the department.

LAURENCE ALBERT MORGAN, former service engineer, Hydraulic Pumper, Inc., Tulsa, Okla., has taken a similar position with the Buda Co., Harvey, Ill.

N. E. WAHLBERG, vice president of engineering, will head the newly-formed Engineering Research Division of the Nash Motors Division, Nash-Kelvinator Corp.

FLOYD KISHLINE, formerly assistant engineer, will be the new chief engineer of the Nash Motors Division, replacing Meade Moore who will be associated with Mr. Wahlberg in the company's new research activities.

Recently **B. C. HEACOCK** was named chairman of the Executive Committee, Caterpillar Tractor Co. and Louis B. Neumiller became president of the company, replacing Mr. Heacock. Mr. Heacock will have headquarters in Peoria, Ill., although at present his time and attention are occupied in Washington, D. C., where he went last spring on a leave of absence to serve as special assistant to Under-Secretary of War Robert Patterson. He has been president of Caterpillar since 1930.

Executive Committee Chairman

B. C. Heacock

WILLIAM B. COLLINS, formerly assistant chief engineer, Kinslow Engineering Corp., Santa Ana, Calif., is now a partner, Exola Products, Los Angeles (lubrication specialties of every kind).

ROBERT E. DUNHAM was recently appointed chief research consultant of Harry R. Lewis Co., petroleum industry consultants, Warren, Pa. Mr. Dunham has gained widespread recognition in the petroleum and allied industries for his work as a member of the engineering staff of Sinclair Refining Co., and as chief lubrication engineer of Hyvis Oils, Inc.

J. W. MENHALL, formerly president, Highway Trailer Co., Edgerton, Wis., now heads the J. W. Menhall Drilling Co., Benton, Ill.

ALEXANDER SATIN is now with the Consolidated Aircraft Corp., San Diego, Calif., as design engineer. Before taking this position he was chief engineering instructor, School of Aeronautics, California Flyers, Inc., and chief engineer, Briegleb Sailplane Co.

Walter Alvin Parrish Named Executive Engineer

WALTER ALVIN PARRISH, former assistant chief engineer, Cummins Engine Co., Columbus, Ind., was recently named executive engineer, Superior Engine Division, National Supply Co., Springfield, Ohio.

HAROLD S. VANCE, chairman of the board of directors of The Studebaker Corp., has been elected to the board of trustees of the Illinois Institute of Technology.

A. P. FONTAINE, formerly chief engineer, Stinson Aircraft Division, Wayne, Mich., has been transferred to Vultee Aircraft, Inc., Downey, Calif. Vultee is the parent organization.

RICHARD H. JOHNSON recently joined the A. O. Smith Corp., Milwaukee, as assistant in charge of experimental procedure in the Aviation Division. Formerly he was development engineer, Briggs & Stratton Corp., Milwaukee.

RAYMOND D. KELLY was recently transferred from superintendent of engineering research to superintendent of development, United Air Lines Transport Corp., Chicago.

EDWARD WARNER, vice chairman, Civil Aeronautics Board, was one of the panel speakers at a recent session of the Foreign Policy Association Forum, held in the Waldorf-Astoria. The subject: "Our First Line of Defense."

HAROLD L. POPE recently went to work for the Glenn L. Martin Co., Baltimore.

Lee Oldfield Made Chief Engineer

LEE OLDFIELD has been made chief engineer of the Merz Engineering Co., Indianapolis. He was formerly consulting mechanical engineer, Schwitzer-Cummins Co., Indianapolis.

THOMAS O. GLOVER recently joined the Research Department, Lockheed Aircraft Corp., Burbank, Calif.

CURT SAURER, general manager, Curt Saurer Co. (sales representation and engineering of rubber parts), has been transferred from Detroit to Washington, D. C.

GARD D. GROCE was recently named West Coast manager, The Cleveland Tractor Co., San Francisco. Formerly he was general service manager, Buda Engine Co., Harvey, Ill.

JOHN GEORGE MYERS, formerly draftsman, Diesel Division, General Machinery Corp., Hamilton, Ohio, has taken a similar position with the Allison Division of General Motors Corp., Indianapolis.

HERBERT L. EGGLESTON, manager of the gas and refining departments, Gilmore Oil Co., Los Angeles, was recently elected to serve as manager on the ASME Council for a three-year term.

RAYMOND R. SNYDER, who has been in charge of the Indianapolis branch of the Curtiss Propeller Division, Curtiss-Wright Corp., has been put in charge of all propeller divisions of the company. He will continue his work at the Indianapolis branch.

RALPH W. LOHMAN is now an electrical engineer, Lake Washington Shipyards (steel ships – complete), Houghton, Wash. Formerly he was electrical research engineer, Lane-Wells Co., Los Angeles.

Director of Engine Factories

Harold Heath

HAROLD HEATH recently took over the directorship of the aircraft engine producing factories controlled by Rootes Securities, Ltd., Coventry, England. These factories are producing aircraft engines for the latest types of bomber and fighters and Rootes is controlling them on behalf of the British Government.

JOSEPH S. SELAN, who previously tested single-cylinder engines, Pratt & Whitney Aircraft Division, United Aircraft Corp., East Hartford, Conn., recently joined the Hartford-Empire Co., as an installation engineer.

FRED J. BOLL was recently advanced from supervisor, Production Process Research, Lockheed Aircraft Corp., Burbank, Calif., to manager, Research Laboratory.

HOLLISTER MOORE, manager of the SAE Membership Department, spoke before the New York Conference on Association Publicity, Oct. 24, as a member of a panel discussing "What Association Publicity Men Should Know About Membership Promotion."

ROBERT B. LEWIS has been advanced from junior to senior test engineer, Wright Aeronautical Corp., Paterson, N. J.

HAROLD W. KLAS has been called to Washington, D. C., as civilian expert technician in the Naval Ordnance Laboratory on Mechanical Devices. Mr. Klas was formerly eight years with General Motors Corp., over four years with Packard Motor Car Co., and briefly with Budd Heat Induction, Inc., Detroit.

ADELBERT ERNST KOLBE, formerly in charge of engine design, Drafting Room, Chevrolet Motor Division, General Motors Corp., Detroit, has been transferred to the Chevrolet Motor & Axle Division, Tonawanda, N. Y., where he will serve as quality engineer. The latter division is now engaged in making aviation engines.

R. E. KERR recently accepted a position with the Defiance Spark Plug Corp., Toledo, Ohio, as development engineer for aviation spark plugs. Previously he was assistant research engineer, Willys-Overland Motors, Inc., Toledo.

A. VANCE HOWE, who is now western manager of Manufacturers Sales, Bendix-Westinghouse Automotive Air Brake Co. Mr. Howe's headquarters are at Elyria, Ohio. Previously, he was Detroit district manager.

Appointed Sales Manager

Marvin W. Davis

MARVIN W. DAVIS was recently appointed sales manager, Suprex Gage Co., Detroit. Formerly he was sales representative, Sheffield Gage Corp., Dayton, Ohio.

In Military Services

SETH B. ROBINSON, JR., has been promoted from captain to major, U. S. Army, and is now connected with the Motor Transport Division, Office of Quartermaster General, Washington, D. C. His assignment is procurement control. Previously he was in the Plans and Training Section, 101st Field Artillery, Camp Edwards, Falmouth, Mass.

MAJOR P. H. ROBEY was recently promoted from a captain, U. S. Army Air Corps, Wright Field, Dayton, Ohio.

FRANK J. HIERHOLZER, has been promoted from captain to major, U. S. Army, is now instructor, Staff and Faculty, Department of Motor Transport, Fort Sill, Okla. He was previously in the Department of Materiel, Staff and Faculty, Field Artillery School, Fort Sill.

The status of **MAURICE A. BRISCOE**, senior procurement inspector of aircraft engines, Office of the Air Corps Representative, Aviation Division, The Studebaker Corp., South Bend, Ind., was changed recently to senior administrative procurement inspector.

STANLEY J. CZYZAK, has been transferred from the Ordnance Department, U. S. Army, Washington, D. C., to the U. S. Air Corps, and is at present assigned to Fairfield Air Depot, Patterson Field, Fairfield, Ohio, as engineering officer.

NICHOLAS POST is now a mechanical engineer, U. S. Army Air Corps, Experimental Engineering Section, Wright Field, Dayton, Ohio.

CLAYTON B. HOWE, formerly sales engineer, Steel & Tubes, Inc., Ferndale, Mich., has been called to active duty as a Naval Reserve officer.

MAJOR C. G. KUSTNER is on leave of absence from the Standard Oil Co. (Ind.) and is at present serving at the Engineer Training Section of the Engineers Replacement Training Center, Fort Belvoir, Va.

WILLIAM J. PATTISON now ranks as an aviation cadet, U. S. Naval Air Station, Jacksonville, Fla. Formerly he was experimental engineer, Charter Sleeve Valve Engine Co., Chicago.

GORDON E. TERPENNY is now a junior engineer (engine model test unit) Materiel Division, U. S. Army Air Corps, Wright Field, Dayton, Ohio. Formerly he was assistant procurement inspector, Aircraft Engines, at Wright Field.

ROBERT A. HERMANN, formerly junior aeronautical engineer, U. S. Army Air Corps, Materiel Division, Wright Field, Dayton, Ohio, recently was transferred to the Naval Aircraft Factory, Navy Yard, Philadelphia, in a similar capacity.

ROBERT A. RUSK has been called to active duty with the U. S. Army and is at present stationed at Fort Sill, Okla. Formerly he was in the Engine Research Laboratory, International Harvester Co., Fort Wayne, Ind.

CHARLES M. HANNUM is back at his job of chief machine tool designer, The National Tool Co., Cleveland, having spent six months as 2nd Lieutenant, motor officer, Troop D, 107 Cavalry, U. S. Army, Camp Forrest, Tullahoma, Tenn.

W. REX BRASHEAR recently left MacCarthy Motor Co., St. Louis, Mo., where he was a salesman, to enter the purchasing department, Emerson Electric Mfg. Co., St. Louis.

JOHAN HOEGH BOUMAN is now an industrial engineer, Sperry Gyroscope Co., Inc., Brooklyn. Formerly he held the same position with Carnegie-Illinois Steel Corp., Youngstown, Ohio.

Student Branch Chairman **ROBERT A. SFORZINI** has been selected as the SAE representative on the Engineering Council, the University of Michigan committee which governs the activities of the engineering students.

HARRY A. NOCERINO recently left Curtiss-Wright Corp., Columbus, Ohio, and is now employed as aeronautical engineering draftsman, Vought-Sikorsky Aircraft Division, United Aircraft Corp., Stratford, Conn.

University of California at Los Angeles, graduate **LEWIS T. WORKMAN**, is now employed as instructor in wood work, Downey Union High School, Downey, Calif.

ROBERT J. MEIER recently became production engineer, Defense Contract Service, OPM, Detroit. He is a Massachusetts Institute of Technology graduate.

JOHN M. ERDAHL has been named engineer, Transport Control Division, Allis-Chalmers Mfg. Co., Milwaukee. He was formerly laboratory assistant, Cadillac Motor Car Division, General Motors Corp., Detroit.

JOHN ALSTON CLARK who has been weight analyst, Pratt & Whitney Aircraft Division, United Aircraft Corp., East Hartford, Conn., now holds the position of designer with the same company.

HENRY GEORGE HELLIER is now associated with Trinidad Leaseholds, Ltd. (oilfield and refinery) B.W.I., as an internal combustion engineer. Formerly he was automotive engineer in charge of transportation, United British Oilfields of Trinidad, Ltd., B.W.I.

OSCAR W. SJOGREN has been appointed sales manager, Killefer Mfg. Corp., Los Angeles. He is also agricultural engineer.

WILLIAM E. HORENBURGER, formerly associate editor, *Motor Magazine*, New York City, was recently named technical assistant to the president, Excel Foundry and Machine Co., Inc., New York City. The company is engaged in manufacturing defense products, chemical equipment, and "Jeep" cars.

HAROLD P. MOON is now a test pilot of military aircraft, Curtiss Aeroplane Division, Curtiss-Wright Corp., stationed at the Buffalo Municipal Airport, Buffalo, N. Y. Formerly he was general manager, Summit Aeronautical Corp., Bendix, N. J.

IVAN H. NASH, formerly foreman, Experimental Engine Laboratory, Caterpillar Tractor Co., San Leandro, Calif., is now an instructor in diesel power (plant course for Naval Reserves), University of California, Berkeley.

THOMAS L. COWLES is now an engineering representative, Chicago plant, Aviation Division, Studebaker Corp. Formerly he was chassis engineer, Studebaker Corp., South Bend.

FRANK ARLEN recently joined the Kalamazoo Stove & Furnace Co., Kalamazoo, Mich., as plant engineer. Formerly he was connected with Sales, Century Metal Products Co., Kalamazoo.

WESLEY H. TEMMING recently left the Fisher Body Division, General Motors Corp., Flint, Mich., where he was metal checker, to become instructor, General Motors Institute, Flint.

RUSSELL HAROLD JOHNSON is now an automotive designer, General Motors Corp., Detroit. He was formerly a draftsman, Packard Motor Car Co.

Peter F. Rossmann
Military Engineering

PETER F. ROSSMANN has been named assistant to director of military engineering, Airplane Division, Curtiss-Wright Corp., and is stationed at the Airport Plant, Buffalo. He was formerly development and research engineer, Curtiss Aeroplane Division.

EINO HENRY NURME formerly structural designer, Douglas Aircraft Co., Inc., Santa Monica, Calif., is now doing full-scale layout (aircraft) work for The Murray Corp. of America, Detroit. The company is producing airplane sub-assemblies.

SYLVAN E. BURK recently left Bendix Products, Division of Bendix Aviation Corp., South Bend, where he was a junior designer, to become a tool designer with Wright Aeronautical Corp., Lockland, Ohio.

ALBERT CHARLES KELLY was recently named tool designer, Air Associates, Inc., Bendix, N. J. Formerly he was tool designer, Breeze Corporations, Inc., Newark.

RUSSELL MELVIL WHEELER was recently put in charge of the inspection department, Seneca Falls Machine Co., Seneca Falls, N. Y. He had previously been assistant chief inspector, Brewster Aeronautical Corp., L. I. City, N. Y.

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truck transmissions to scores of industrial applications indicates that the selling price is competitive even in fields for which this type of transmission was not primarly designed and last, the design is such that any good mechanic can service the transmission in the field, and no adjustments are necessary."

Owing to the differences in the function of passenger-car transmissions and truck transmissions an automatic or semi-automatic transmission which meets the requirements of passenger cars or perhaps even buses may be entirely unsuitable for trucks, simply because in varying types of service the necessity of employing any given ratio is not always the function of torque and speed requirements to which such devices are necessarily responsive, Mr. Backus said. A truck on which a rotary snow plow is mounted was used as an example. The speed at which the truck should be driven is dependent on the speed at which the plow can handle the snow. Designing a transmission which would automatically shift to the proper ratio for any conceivable truck requirement ranks in the class of perpetual motion inventions, Mr. Backus commented.

A transmission was defined as "nothing but a set of ratios." In a passenger car the major function of these ratios is to allow the desired acceleration, Mr. Backus said. In a heavy-duty truck the transmission ratios become, in effect, in combination with the axle, a series of axle ratios to provide the truck with the various ability factors required to meet varying conditions of operation.

The amount of maintenance work required and, in fact, the life of the transmission are influenced to a considerable extent by external factors beyond the control of the manufacturer. These were listed by Mr. Backus as: driver ability, roads, maintenance facilities.

Maintenance facilities are seldom given enough attention, he stated. When assembling a transmission at the factory every possible effort is made to keep dirt out of the unit. Each part is individually wrapped until actually assembled and the oil used in testing is run through a centrifugal cleaner. In many operations the first time the transmission cover is removed in the field, this effort toward cleanliness is nullified. We have observed maintenance set-ups close to large construction projects, Mr. Backus said, where special rooms have been built to handle motor repair work, and yet transmission repairs are taken care of in the open, often with clouds of dust blowing through an open repair shed and depositing dirt as a fine abrasive on gears and bearings.

A bearing which has been rendered unserviceable through operating in the presence of foreign matter is usually easy to recognize, Mr. Backus continued. The raceways and balls have a full gray lapped appearance, and the retainer is often worn. Both end play and radical clearances are excessive. A bearing which has failed through overload almost invariably shows some spalling on either the balls or raceways. A recent report from one truck manufacturer stated that service records indicated that bearings were consistently the first parts that need replacement in transmissions, and that such replacements were necessary when there was still plenty of wear life left in the gears.

On the subject of transmission lubrication, Mr. Backus said, the procedure that is often followed, when an operator considers that a transmission is running at too high a temperature, is to add more oil, or is to put in an oil of higher viscosity. Either or both of these procedures are wrong, he stated, as they increase the amount of agitation and the result is still higher temperatures. A reduction in the quantity of oil, or the use of an oil of lower viscosity, is a step in the right direction, but even such steps should not be taken without consulting the transmission manufacturer.

The first part of the paper was devoted to history of hydro-kinetic transmission development.

Aircraft and Diesel Engines Discussed At Student Group

ON Oct. 8, an SAE student group at the College of the City of New York, heard a talk on the development of aircraft engines by Ivan Harkleroad, Wright Aeronautical Corp. Mr. Harkleroad discussed the important controlling factors which exercise great influence in aircraft development, along with some of the features of aircraft testing in the various countries of the world. A lively question-answer session followed the talk.

Seven days later, at another weekly session, the Chapter was audience to W. J. Cumming, Surface Transportation Corp. He spoke on the problems confronting the use of diesel engines in buses and the economic justification of their use.

The student group is now in its fourth semester.

Load Distribution Stressed As Important in Fleet Economy

▪ Oregon

"THE important dimension in motor vehicle load distribution is the distance the load center or concentration point is located ahead of the truck or tractor rear axle. This may be either the vertical center of a body and its load on a straight truck or the vertical center of a fifth-wheel on a tractor truck. For a correct gross weight distribution on a given wheelbase the load center should be located at the same point whether straight truck or tractor."

This statement by Fred B. Lautzenhiser, chief transportation engineer, International Harvester Co., keynoted his talk "Motor Vehicle Load Distribution Factors," delivered before the Oregon Section, Oct. 21.

Placing the load center ahead of the rear axle distributes weight more evenly, increases the amount of payload which can be carried, saves tire wear, leads to better steering control, and makes for safer driving in mountainous regions, Mr. Lautzenhiser said.

Truly efficient transport operation, Mr. Lautzenhiser visualized as dependent upon five major factors: (a) maximum tonnage (b) best possible speed (c) continuous operation (d) safety (e) lowest cost per unit transported. To reach a correct analysis of a specific transport operation – one that will provide the basic information for properly specifying equipment – the following steps are recommended by Mr. Lautzenhiser. First, make a time study of the proposed operation, including nature of commodity, its weight and bulk, quantity to be moved per day, per hour, and frequency of trips. Next, check on nature of roads to be traveled, hills to be climbed, the traffic conditions, and number of stops and starts, with estimated time lost in loading and unloading. Then check-up and summarize restrictions imposed upon dimensions and weights of various types of equipment by the laws of the states through which the operation must proceed. And last, investigate the income that will accrue from the operation. This should provide an all-around picture for judging the type of body required, its general dimensions and its weight, Mr. Lautzenhiser said.

The speaker gave specific mathematical calculations for determining load center and fifth-wheel positions.

Thirty Men in Plants For Every Man at Front

▪ U. of Michigan

IF we had to go into the war today, there would be thirty men in the factories for every man on the front, Clyde R. Paton said in a talk before a joint meeting of the student branch of the American Society of Mechanical Engineers and the student branch of the Society of Automotive Engineers at the University of Michigan, Oct. 23. In World War 1, seven workers were required in the factories for every man in the trenches, he stated.

Mr. Paton, chief engineer, Packard Motor Car Co., gave approximately 120 members and guests an intimate picture of the problems encountered by engineers in the change-over from peacetime to defense production, in a paper entitled, "Production in Defense."

The talk was followed by a movie, with narrations by Lowell Thomas, showing the production of the Curtiss-Wright Cyclone engine, stressing the foundry and machine-shop operations.

Students View Pictures Of Indianapolis Races

▪ U. of Michigan

THE University of Michigan Student Branch opened its program for 1941-42 on Oct. 9, when 90 members and students witnessed movies of the Indianapolis Races, the John Cobb Run, and other racing subjects. Chester S. Ricker, director of timing and scoring at the 500-mile classic since 1911, related many interesting stories concerning racing activities. Richard B. Sneed, Chairman of Junior Activity in the Detroit Section, was also present to open the membership drive.

On Oct. 13, 27 members were the guests of the General Motors Corp. in an inspection trip through the General Motors Proving Ground.

(*Continued from page 23*)

ther puts the lead industry on an all-out defense basis. This first joint request to an industry by the two top OPM officials and the price administrator is seen as the pattern of things to come.

Truck Output Orders Extended to Jan. 31

In line with the basic provisions of OPM's motor-truck production program, limitation order L-1-a (Oct. 15) was extended to Jan. 31, as was P-54 (Oct. 15) to aid manufacturers to get required material. This is the second extension of the original L-1 (Aug. 30) restricting production of certain types of trucks and buses.

The order means:

- Production of light trucks (under 1½ tons) is limited to 24,169 units, or a reduction of 35.9% below that of January, 1941, except for defense orders.
- Manufacturers producing both light trucks and passenger cars may exceed their respective light-truck output if they correspondingly reduce the number of passenger cars built.
- Production of medium trucks (1½ tons or more) is restricted to 5/6 the number of trucks of this category produced during the first half of 1941, except defense orders.
- Production of heavy trucks (3 tons or more) is not restricted.
- Bright work, requiring critical materials, is banned after Dec. 15.

Extension of orders L-4-a (Nov. 14) and P-57 on replacement parts (Sept. 18 until Jan. 31) limits production to 1/3 the number of parts sold for replacement during the Jan. 1 to March 31, 1941, period.

Donald M. Nelson, director of priorities, announced that these extensions were made to permit manufacturers to place orders for steel to meet their production requirements.

Further extension of this order past Jan. 31 is expected to include a change from tonnage to gross-vehicle ratings as a basis for defining light, medium, and heavy vehicles – with the possibilities of some form of specific allotments in each of the three groups.

New ASTM-SAE Group Will Reduce Rubber Specs

A project intended to conserve rubber through a reduction in the number of specifications for automotive mechanical rubber parts, was launched at a meeting held in Detroit on Oct. 23. The meeting was called under the auspices of ASTM-SAE Technical Committee A on Automotive Rubber, and resulted in the naming of Subcommittee IV to carry on the work.

Compilation and classification of existing automotive rubber specifications have been undertaken, and will form the bases for discussion at a meeting to be held in the Book-Cadillac Hotel at 9:30 a.m. Dec. 3. At this meeting an effort will be made to reduce the present list of specifications to the minimum practical number for use by the industry for the duration.

Manufacturers interested are invited to send representatives to this meeting.

Arms Cost Ratio to Income Rising

Defense impact on everyone in the country is graphically shown in the steady rise cash outlay for equipping our Army and Navy, and satisfying Lend-Lease requirements during the past 15 months.

The tabulation below shows the ratio of these expenditures to income tax payments, i.e., salaries, wages, dividends, interest, entrepreneurial income, rents, royalties, and relief and unemployment insurance payments as calculated by the Department of Commerce. Defense payments were supplied to the SAE Journal by the OPM Bureau of Research & Statistics. Third quarter, 1941, defense expenditures estimated:

Period	Income Payments	U. S. Defense Expenditures	% U. S. Defense Cost of Income Payments
3rd Quar., 1940	18,695	682	4
4th Quar., 1940	20,708	1227	6
1st Quar., 1941	20,045	2313	12
2nd Quar., 1941	21,437	2835	13
3rd Quar., 1941	22,832	3593	16

Industry's Advice Gathers Weight

FURTHER indications of successful cooperative effort between the automotive industry and OPM were seen in announcements following the meeting of the OPM passenger-car subcommittee, Nov. 19, with Andrew Stevenson, acting chief, Automotive, Transportation, and Farm Equipment Branch, OPM, when recommendations made by the automotive advisory committee and its technical subcommittee during recent meetings were approved. They included:

- The reduction of 56.1% of the February, 1941, production for February, 1942, is in line with the August proposals.
- Bright work will be continued through December, 1941, in order to use up stocks on hand, instead of being eliminated Dec. 15 as required in the original bright-work order.
- Decorative parts already finished or in process on Oct. 27 which contain critical materials prohibited by the bright-work order may be used if painted over to eliminate their identity as bright work.
- Chromium and other scarce materials plating on external locks and their caps, windshield wiper arms and blades, interior screws, and window ventilator latches, may be exempted. OPM's AT&FE Branch has agreed with the industry that such plating of these parts is functional rather than decorative.

Off the record discussions with OPM chiefs by the SAE Journal indicate that the trend toward more use of the advisory committee in keying the whole industry to defense needs is rapidly becoming a fixed policy. (See "Transition," page 19).

This industry advisory committee structure will play an important part in the new SPAB program ("Materials Allocations throughout Industry Puts U. S. on War Footing," page 20), Leon Henderson, director, Civilian Supply Division, OPM, said, in warning that increasing scarcity of materials may require a further curtailment of automotive production.

The AT&FE Branch announced that it plans to ask the government to freeze the 1942 automobile designs for 1943, thus saving tooling costs and engineering manhours which could be diverted to the defense effort. At the same time, the Consumers' Division, OPA, has suggested reduction in the number of automobile models, apparently in the belief that such reduction would save critical materials. Presumably, these and other suggestions will be presented to the industry's advisory committee to work out the details and to report its recommendations to OPM.

Michigan Low-Ore Copper to be Tapped

Since the first time OPM mentioned the word "shortage" in connection with copper, Michiganders in general and automobile men in particular have been reiterating: "How can there be a real shortage when we aren't even tapping the 'high cost' mines in Michigan?" That argument finally is settled – for the time being at least. The U. S. Treasury Department has signed contracts to purchase the entire output of three "high cost" Michigan copper producers for six months – at production cost plus 1c per lb.

There is no indication, however, that this production will relieve current shortages in any important respect, OPM officials insist. If all the "high cost" copper practically available had been taken out this year it would have increased our supply something like 8% – and estimated shortages for civilian use are much more than that.

Allocation Order Puts Russia First

The clear trend toward all-out allocation (see "Materials Allocations Throughout Industry," page 20, and "OPM Materials Orders," page 22) took on added firmness when OPM issued Special Allocation Order No. 1 on Nov. 11.

This order, served on 35 machine tool builders, directed each to accept specified purchase orders placed by Amtorg Trading Corp., the Russian government's purchasing agency. Other OPM preference ratings and orders took a back seat.

Terms:

- Amtorg must meet established prices and terms of sale,
- Tool manufacturers must make deliveries on dates specified.

In effect, the U.S.S.R. comes ahead of U. S., British, and other Lend-Lease machine tool requirements for the next 12 months. OPM expects to allocate from $10 to $15 million of tools to Russia.

Machine tool production in the U. S. will amount to about $800 million this year, as compared with $423 million in 1940.

• A Round-Up On Materials Situations

(*Continued from page 19*)

steel in 1942 than in 1941. Indications now are that there will be available[2] for civilian use only about 35 to 40% of the 1939 civilian consumption.

Strong emphasis was placed, however, on the fact that general tonnage figures don't mean much until interpreted into demand and supply data for specific types of steel. (One speaker even stated that: "It looks as if nobody but the Army and the Navy were going to get any alloy steel next year.")

Immediate need for segregating various types of steel scrap – as already is being done with aluminum widely – was emphasized by several government spokesmen. (One stated that an OPM order is now being written to prohibit use – except by special permission – of iron or steel scrap out of which machine tools may be made.)

The situations outlined on other metals were no more optimistic – in some cases much less so.

Copper

The widely publicized copper stringency was further emphasized. Detailing a November supply of about 128,000 tons and a demand of about 227,000 tons – including nearly 152,000 tons for defense-rating and Lend-lease projects, it was brought out that production increases of 150,000 tons a year are arranged for but cannot become effective until late in 1942. The biggest quick hope for easing the copper situation, it was stressed, lies in conservation and return of scrap. Ordinarily, about 40% of the copper used returns to the smelters as scrap; now only 7% is returning. If scrap return could be brought back to normal, the available supply would be increased more than if all possible additional production were immediately available.

Possibilities of major copper saving through development of a satisfactory steel cartridge shell casing were stressed by EDB and NAS representatives, but it was recognized that testing, development and quantity production of such a substitute would take so long as to have little effect on the immediate shortage. "Moreover," one industry representative stated, "we might be shooting shells so fast by that time that we would need all we could get of both types."

Additional expansion of nickel mining and refining facilities are in process, the technicians were told, but these new facilities will not be in production until late in 1942. OPM is working with military as well as civilian agencies to eliminate "extravagant" uses of nickel.

Zinc

The zinc situation apparently remains relatively good. The government allocation pool now comprises 31% of the total output . . . and 40% of that 31% is going to Lend-lease projects. An industry representative stated that the 1942 supply of zinc would run about 978,000 tons, which – after subtracting currently-estimated military requirements – would leave about 429,000 tons in 1942 for civilian consumption. (In 1939 civilian consumption – excluding zinc used in brass – was about 425,000 tons.)

Tin and Lead

Tin, it would appear from OPM statements, is still tight, but not so acutely short as most other metals. Increased Lend-lease demands, however, have used up at least all of the savings accomplished by reduction in tin use in can coatings. Savings of tin in solder at the expense of lead, it was stated, is undesirable because defense demands for lead thus far have been greatly underestimated. About 40% of our lead is imported, imports coming from Canada, Mexico and Australia. However, a tin order is in the offing.

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Chrome

Increasing demands for chrome will make it hard to hold our own in supplies of this metal, OPM spokesmen stated. Chrome used in stainless steel can be further reduced, they said. The ferro-chrome supply, it was said, will limit the types of steel available for defense orders.

The increased bomber program is taking up all slack in stainless steel supply brought about by elimination of bright work from passenger automobiles, one speaker announced. Chrome production is now at 230% of its previous peak, but defense demands are mounting just as fast – and our ultimate supply depends largely on shipping.

One speaker thought that the proposed reduction of chrome in steel alloys would not apply to steels used in defense work, but another indicated his belief that demand for this metal is growing so fast that new non-defense specifications may have to go over into defense projects as well.

Tungsten

Tungsten supply and demand are running neck and neck this year, OPM estimates indicate, and a somewhat similar condition may be expected to continue next year – provided shipping for importations remains unimpeded. In fact, the supply may even exceed the demand slightly.

Molybdenum

The molybdenum situation, while not bad, does not indicate as favorable a relationship between demand and supply as has generally been accepted. This year, supply and demand are breaking about even, it is said, while estimates for 1942 point to a production of 48 million lb to meet a demand for 54 million lb.

The OPM high-speed steel order, requiring pound for pound use of molybdenum and tungsten, has accomplished wonders in tungsten conservation, it was stated. About ¾ of high-speed tool steel output now is high moly and low tungsten. There has, however, been a vastly increased use of tungsten in other-than-tool-steel uses. Studies are now under way to see just what these new uses are and to try to find some substitute for tungsten in these new uses. Government representatives say: "We hope the substitute won't be molybdenum."

Vanadium

The vanadium picture is dark, OPM experts say. Although production and consumption this year both will be around 4,200,000 lb, estimates for next year point to a possible production of 5,300,000 lb and a demand for 7,150,000 lb. "It's got us licked," one speaker groaned.

C. L. Warwick, ASTM general manager who is now working for OPM conservation division and heading up the development of emergency steel specifications, outlined the progress made in this project, after sketching its genesis and purpose. He concluded by expressing the hope that the job would be completed in such a manner as to make the reduced specifications desirable for continued use after the emergency had ended. The committees working on the project are using defense needs as the primary requirement, however, he emphasized, saying that modification of certain specifications themselves – in addition to elimination and simplification – probably will be necessary.

G. T. Weymouth, OPM Conservation Division, in charge of elimination of waste and stimulation of salvage activities in industry, outlined the objectives of his division, saying that if OPM can tell all industry (a) why scrap and salvage are necessary and (b) what it wants done and how . . . then industry will do it – as many companies already are. Widespread application of improved conservation and salvage methods, he said, should have an important favorable effect in meeting low-labor-cost competition after the war.

Requests EDB Aid

Mr. Weymouth specifically requested help from the Engineers' Defense Board, asking EDB to: (1) nominate a subcommittee to work with him on salvage; (b) to give him the names of factories doing a particularly good salvage job, from which he could get exhibits to help educate others; (c) help in getting, putting together and editing specific instructions to industries about how to do the salvage job; (d) provide a speakers bureau to help in a salvage education campaign; (e) appoint a consultant in each large city area to whom small companies can go for advice on better salvage methods; and (f) give aid in bringing up to date the "Waste Materials Dictionary" published in 1932 by the American Society of Mechanical Engineers.

APPLICATIONS Received

The applications for membership received between Oct. 15, 1941, and Nov. 15, 1941, are listed below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

Baltimore Section

Robinson, James D., Jr., draftsman, Glenn L. Martin Co., Middle River, Md.

Canadian Section

Barber, Frank F., president and managing director, F. F. Barber Machinery Co., Ltd., Toronto, Ont., Canada.

Danby, Charles B., teacher, Toronto Board of Education, Toronto, Ont., Canada.

Kilmer, Norman E., manager, Weatherhead Co. of Canada, Ltd., St. Thomas, Ont., Canada.

Chicago Section

Bennett, Dale L., chief engineer, United Specialties Co., Chicago.

Blaul, Walter L., truck equipment, Axle & Equipment Sales Co., Chicago.

Carson, Knight S., Lt., Adjutant, Elwood Ordnance Plant, U. S. Army, Joliet, Ill.

Galandak, August, mechanical research engineer, Victor Mfg. & Gasket Co., Chicago.

Hess, Everett J., charge of radial engine test equipment, Buda Co., Harvey, Ill.

Mains, Raymond D., sales manager, Fruehauf Trailer Co., Chicago.

Morey, Albert A., assistant vice president, Marsh & McLennan, Inc., Chicago.

Owen, Jack, purchasing agent, Western-Austin Co., Aurora, Ill.

Steder, Marshall, assistant general manager, L. J. Miley Co., Inc., Chicago.

Triplett, Harlow A., engine tester, Electro-Motive Corp., La Grange, Ill.

Cleveland Section

Doering, Clayton M., plant layout engineer, Thompson Products, Inc., Cleveland.

Kanavel, Charles H., sales engineer, B. F. Goodrich Co., Akron, Ohio.

Kinney, L. W., Jr., sales engineer, White Motor Co., Cleveland.

Mueller, Ralph J., sales and patent engineer, Lake City Malleable Co., Cleveland.

Parks, John T., engineer, Thompson Products, Inc., Cleveland.

Schaaf, Oliver H., vice president and general manager, Air-Maze Corp., Cleveland.

Dayton Section

Geddes, W. Hayward, research engineer, United Aircraft Products, Inc., Dayton, Ohio.

Detroit Section

Abbott, Robert L., engineer, General Motors Overseas Operations, Detroit.

Baxley, Robert Van Namee, experimental engineer, Detroit Diesel Engine Division, General Motors Corp., Detroit.

Braendel, Helmuth Gunther, experimental engineer, Chrysler Corp., Highland Park, Mich.

Brandau, Marvin W., chief engineer, Aeroquip Corp., Jackson, Mich.

Carson, Alexander E., sales engineer, Delco-Remy Division, General Motors Corp., Detroit.

Chambers, Allan C., sales manager, Bendix Products Division, Bendix Aviation Corp., Detroit.

Declerck, Oscar G., aircraft supervisory training, Chrysler Corp., Detroit.

DeWitt, Frederick James, Jr., assistant sales manager, Parker Rust Proof Co., Detroit.

Eipper, Eugene B., junior engineer, Detroit Diesel Engine Division, General Motors Corp., Detroit.

Fairbairn, Donald, salesman, B. F. Goodrich Co., Detroit.

Fox, Harold E., technical engineer, General Motors Truck & Coach Division, Yellow Truck & Coach Mfg. Co., Pontiac, Mich.

Gastren, Stuart M., processing engineer, General Motors Truck & Coach Division, Yellow Truck & Coach Mfg. Co., Pontiac, Mich.

Griffith, Bain, gear designer, Chevrolet Motor Division, General Motors Corp., Detroit.

Guernsey, Robert W., junior engineer, Detroit Diesel Engine Division, General Motors Corp., Detroit.

Jepson, Karl H., technical test engineer, Chevrolet Motor Division, General Motors Corp., General Motors Proving Ground, Milford, Mich.

Jones, Henry Warren, manager, Detroit division, Swan-Finch Oil Corp., Detroit.

Kelly, Eugene K., special assignment and project engineer, Yellow Truck & Coach Mfg. Co., Pontiac, Mich.

Ketchman, Rodger A., group leader, Ford Motor Co., Dearborn, Mich.

Kilgour, Thomas R., draftsman, Chrysler Corp., Detroit.

Mann, Neil W., engineer, McCord Radiator & Mfg. Co., Detroit.

Mattson, Raymond Lionel, junior research engineer, Research Laboratory Division, General Motors Corp., Detroit.

Miller, J. C., Jr., project engineer, Detroit Diesel Engine Division, General Motors Corp., Detroit.

Mitchell, John B., detail, layout, Detroit Diesel Engine Division, General Motors Corp., Detroit.

Oxley, Herbert, research engineer, Research Laboratory Division, General Motors Corp., Detroit.

Rothrock, George L., engineer, chassis units, Cadillac Motor Car Division, General Motors Corp., Detroit.

Rudnicki, William, draftsman, Chevrolet Motor Division, General Motors Corp., Detroit.

Smith, Charles J., manager, proving grounds, Packard Motor Car Co., Detroit.

Stamy, David R., sales engineer, Standard Products Co., Detroit.

Stefan, John Paul, research engineer, Ethyl Gasoline Corp., Detroit.

Stinson, Forrest A., sales engineer, Delco-Remy Division, General Motors Corp., Detroit.

Strickland, Paul D., engineer, Bower Roller Bearing Co., Detroit.

Sumner, Herbert Clark, research engineer, Ethyl Gasoline Corp., Detroit.

van der Steel, William, engineer, Styling Section, General Motors Corp., Detroit.

Vorhees, Roy W., Jr., experimental engineer, Chrysler Corp., Detroit.

Weidman, Arthur A., superintendent, Cadillac Motor Car Division, General Motors Corp., Detroit.

Welch, Harold L., experimental engineer, Chrysler Corp., Detroit.

Indiana Section

Hockert, Chester Edgar, designer and calculator, Allison Division, General Motors Corp., Indianapolis.

McCoy, Joseph E., chief draftsman, Cummins Engine Co., Columbus, Ind.

Meek, Eugene B., co-ordinator, Allison Division, General Motors Corp., Indianapolis.

Metropolitan Section

Albright, Ralph, automotive engineer, Socony-Vacuum Oil Co., Inc., Brooklyn, N. Y.

Bettoney, Wilfred Esty, assistant mechanical engineer, War Department, Air Corps, Materiel Division, Wright Field, Dayton, Ohio. Mail: 86 Diamond Bridge Ave., Hawthorne, N. J.

Clark, Ralph M., inspection engineer, Eclipse Aviation, division Bendix Aviation Corp., Bendix, N. J.

Engle, Gilbert B., assistant project engineer, Wright Aeronautical Corp., Paterson, N. J.

Goodwin, Ralph T., manager, aviation department, Shell Oil Co., Inc., New York City.

Hardy, Norman S., night superintendent, Belmet Products, Inc., Brooklyn, N. Y.

Hartley, Robert Lynd, automotive engineer, Tide Water Associated Oil Co., Bayonne, N. J.

Holms, Arthur G., student engineer, Ranger Aircraft Engines, division Fairchild Engine & Airplane Corp., Farmingdale, L. I., N. Y.

Lee, Nathan S., junior stress analyst, Wright Aeronautical Corp., Paterson, N. J.

Massiello, Manfred J., superintendent of transportation, Gottfried Baking Co., Inc., New York City.

Moore, Boardman W., engineering trainee, Wright Aeronautical Corp., Paterson, N. J.

Myers, G. Taylor, New York manager, White Motor Co., New York City.

Pipines, James, test equipment design, Wright Aeronautical Corp., Paterson, N. J.

Posner, David L., examining assistant, New York City Civil Service Commission, Transit Division, New York City.

Rich, G. Earl, junior engineer, Wright Aeronautical Corp., Paterson, N. J.

Scordato, Emil, mechanical engineer, Combustion Engineering Co., Inc., New York City.

Smith, Floyd Townsend, designer and checker, Breeze Corporations, Inc., Newark, N. J.

Smith, Frank M., detail draftsman, Wright Aeronautical Corp., Paterson, N. J.

Snider, Mike, assistant project engineer, Wright Aeronautical Corp., Paterson, N. J.

Sonn, Maynard P., fleet sales and engineering, Wadhams Division, Socony-Vacuum Oil Co., Inc., New York City.

Warneke, Walter H., garage superintendent, United Dressed Beef Co., New York City.

Milwaukee Section

Hansen, E. T., instructor, mechanical engineering, University of Wisconsin, Madison, Wis.

New England Section

Biron, Edward G., motor machinist, Everett Ave. Auto Parts Co., Brookline, Mass.

O'Neil, Gerald L., president, C-O Cape Co., Hyannis, Mass.

Northwest Section

Egbert, Fred P., shop foreman, Grey Line Tours, Seattle, Wash.

Oregon Section

Irving, G. Eldon, factory manager, Columbia Aircraft Industries, Portland, Ore.

Philadelphia Section

Hurleman, Walter A., assistant design engineer, Jacobs Aircraft Engine Co., Pottstown, Pa.

Lunda, Ernest Milo, bus engineer, Mack Mfg. Corp., Allentown, Pa.

Pearson, Walter H., assistant chief engineer, Szekely Co., Inc., Philadelphia.

Shaw, Harry E., president, Service Supply Corp., Philadelphia.

Stone, Ashton K., Ensign, U. S. Naval Reserve, Inspector of Naval Material, U. S. Navy, Philadelphia.

Pittsburgh Section

Bayster, N. G., lubrication engineer, American Oil Co., Pittsburgh.

Gallagher, J. A., owner, J. A. Gallagher, Pittsburgh.

Price, Edward R., salesman, American Oil Co., Pittsburgh.

Wallace, Gerald W., chief chemist, Freedom Oil Co., Freedom, Pa.

Wright, Sherwin H., industry engineer, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

St. Louis Section

Alco Valve Co., St. Louis, Mo.

Creveling, A. Ben, Jr., test engineer, American Car & Foundry Co., St. Charles, Mo.

Frederick, Arthur I., junior designer, Busch-Sulzer Bros. Diesel Engine Co., St. Louis.

Minges, Harold A., junior engineer, Busch-Sulzer Bros. Diesel Engine Co., St. Louis.

Southern California Section

Benham, Harry T., production control, Thompson Products, Inc., Bell, Calif.

Gish, Rollin E., Jr., research engineer, Ethyl Gasoline Corp., San Bernardino, Calif.

Gregg, Donald, industrial relations, North American Aviation, Inc., Inglewood Calif.

Grosch, Joseph G., Lt., 405th Signal Company, Aviation, March Field, Calif.

Klein, Arthur L., consulting engineer, Douglas Aircraft Co., Inc., Santa Monica, Calif., associate professor, California Institute of Technology, Pasadena, Calif.

Puglisi, Anthony G., designer, Douglas Aircraft Co., Inc., Santa Monica, Calif.

Smith, Paul Louis, engineer, Douglas Aircraft Co., Inc., Santa Monica, Calif.

Spitz, Samuel, president, Spitz Laboratories, Burbank, Calif.

Weishapl, Cyril S., engineering department, Six Wheels, Inc., Los Angeles.

Southern New England Section

Furay, Carl N., design supervisor, Pratt & Whitney Aircraft, division of United Aircraft Corp., East Hartford, Conn.

Lonsdale, James B., service engineer, Pratt & Whitney Aircraft, division of United Aircraft Corp., East Hartford, Conn.

Smith, Lyman J., assistant general foreman, Pratt & Whitney Aircraft, division of United Aircraft Corp., East Hartford, Conn.

Wilkerson, Mary Lee, junior mechanical engineer, United Aircraft Corp., East Hartford, Conn.

Wittmann, Franz J., inspector, U. S. Navy, Bureau of Aeronautics, Washington, D. C. Mail: R.F.D. No. 2, Rockville, Conn.

Syracuse Section

Rutishauser, Hans, designer, Diesel Division, American Locomotive Co., Auburn, N. Y.

Tulsa Group

Beckwith, E. Q., special representative, Phillips Petroleum Co., Bartlesville, Okla.

Hilligoss, Raymond G., assistant professor, Oklahoma Agricultural & Mechanical College, Stillwater, Okla.

Trapp, Henry L., service manager, Downtown Chevrolet Co., Tulsa, Okla.

Washington Section

Cawood, James M., supervisor, Safety Inspection Division, Department of Vehicles & Traffic, Washington, D. C.

Mac'Kie, John G., executive assistant to director, Civilian Conservation Corps, Washington, D. C.

McEntee, J. J., director, Civilian Conservation Corps, Washington, D. C.

Outside of Section Territory

Clanton, Vernon A., salesman, Cities Service Oil Co., New York City. Mail: 2740 Darien St., Shreveport, La.

Delamar, Carl D., assistant chemical engineer, Panama Canal Department Engineer, U. S. Army, Service Section, Balboa, Canal Zone.

Engstrom, W. A., experimental engineer, Continental Motors Corp., Muskegon, Mich.

Epler, Horace W., group leader, engineering department, Lycoming Division, Aviation Mfg. Corp., Williamsport, Pa.

Graham, Howard E., service engineer, Ingersoll-Rand Co., New York City. Mail: 1617 – 22 Marietta St. Bldg., Atlanta, Ga.

Poole, John Ward, research manager, Lion Oil Refining Co., El Dorado, Ark.

Symons, John Edward, flow bench operator, Bendix Products, Division Bendix Aviation Corp., South Bend, Ind. Mail: 129 E. Pearl St., Coldwater, Mich.

Foreign

Hunt, Alfred Stuart, transport manager, United British Oilfields of Trinidad, Ltd., Point Fortin, Trinidad, B.W.I.

Knight, Philip, Lt., A.F.V. Training Centre, South African Tank & Armoured Car Corps, Transvaal, S. A.

NEW MEMBERS Qualified

These applicants who have qualified for admission to the Society have been welcomed into membership between Oct. 15, 1941, and Nov. 15, 1941.

The various grades of membership are indicated by: (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate Member; (SM) Service Member; (FM) Foreign Member.

Baltimore Section

Koci, Ferdinand Joseph (A) internal combustion engineman, U. S. Naval Engineering Experiment Station, Annapolis, Md. (mail) 75 Conduit St.

Canadian Section

Collins, John Ralph (M) assistant general manager, Provincial Transport Co., 1188 Dorchester St. West, Montreal, Quebec.

Chicago Section

Wallace, Francis Lloyd, Capt. (J) United Air Lines Transport Corp., Chicago (mail) 7208 Oak Ave., River Forest, Ill.

Cleveland Section

Eberly, Walter C. (M) special machine designer, Thompson Products, Inc., 2196 Clarkwood Rd., Cleveland (mail) 6401 Euclid Ave.

Kramer, Karl (A) chief draftsman, Leece-Neville Co., 5363 Hamilton Ave., Cleveland.

Schmelzer, J. F. (M) tool engineer, Thompson Products, Inc., 6402 Cedar Ave., Cleveland.

Tanker, George E. (A) apprentice director, Weatherhead Co., 300 E. 131st St., Cleveland.

Dayton Section

Reinhardt, Thomas Friant (J) junior mechanical engineer, War Department, Air Corps, Materiel Division, Wright Field, Dayton, O. (mail) 12 Constance Ave., Southern Hills.

Detroit Section

Bayles, Allison L. (M) works manager, Hill Diesel Engine Co., Lansing, Mich. (mail) 238 Mill St.

Greaves, Robert K. (A) president, Sevaerg Metals Corp., 409 Fisher Bldg., Detroit.

Hatch, James W. (M) sales engineer, Ex-Cell-O Corp., 1200 Oakland Blvd., Detroit.

Hoxie, Ralph A. (J) research engineer, Continental Aviation & Engine Corp., Detroit (mail) 2229 Eastlawn, Apt. 2.

Livingston, William B. (A) assistant to president, General Motors Truck & Coach Division, Yellow Truck & Coach Mfg. Co., Pontiac, Mich.

Marquis, D. P. (M) metallurgist, Thompson Products, Inc., 7881 Conant, Detroit.

McQuaid, John Grant (J) junior engineer, Chrysler Corp., Highland Park, Mich. (mail) 14809 Strathmoor Ave., Detroit.

Vigmostad, Trygve (M) body engineer, Briggs Mfg. Co., Mack Ave., Detroit (mail) 505 W. Greendale.

Indiana Section

Armor, W. G. (J) laboratory engineer, International Harvester Co., Dept. 21, Engineering Research, Pontiac & Bueter Rd., Fort Wayne, Ind.

Metropolitan Section

da Silva, J. Mendes, Capt. (F M) Brazilian Military Aeronautical Commission, Brazilian Air Force, Room 2636, 17 Battery Place, New York City.

Jacobs, Joseph J. (J) research fellow, Polytechnic Institute of Brooklyn, 99 Livingston St., Brooklyn, N. Y.

Luttrell, John C. (J) assistant project engineer, American Airlines, Inc., Jackson Heights, L. I., N. Y. (mail) 22-25 77th St.

Walter, Gustave (A) owner, G. Walter Machine Co., 75 South St., Jersey City, N. J.

Northern California Section

Chapman, W. R. (M) shop foreman, Modesto Irrigation District, Modesto, Calif. (mail) Route 5, Box 1028.

Muchmore, Richard Wayne (J) plant engineer, Federal-Mogul Service, 210 S. Van Ness St., San Francisco (mail) R.F.D. 2, Box 3213, Redwood City, Calif.

Thorne, Elrick F. (A) truck mechanic, Air Reduction Sales Co., Park Ave. & Halleck St., Emeryville, Calif. (mail) 1141 Virginia St., Berkeley, Calif.

Northwest Section

Plumb, Robert N. (A) fleet supervisor, Whatcom County Dairymen's Association, Bellingham, Wash. (mail) 925 13th St.

Shults, Louis Ralph (A) Nelson Equipment Co., 1239 S. E. 12th Ave., Portland, Ore. (mail) 2604 N. E. 27th Ave.

Southern California Section

Cole, Francis Melvin (M) head, Engine Department, Curtiss-Wright Technical Institute, Glendale, Calif. (mail) 4424 Gainsborough Ave., Los Angeles.

Dannan, John H. (M) designer, hydraulic equipment, Consolidated Aircraft Corp., San Diego, Calif. (mail) 2330 Howard Ave.

Hackney, Lyle Raymond (M) weight control staff engineer, Lockheed Aircraft Corp., Burbank, Calif. (mail) 4659 Van Noord Ave., Van Nuys, Calif.

Sanders, Elmore J. (M) engineer, lubricating oil department, Gilmore Oil Co., 2423 E. 28th St., Los Angeles (mail) 2700 W. 82nd St., Inglewood, Calif.

Southern New England Section

Blakely, Carl Frederick (J) test inspector, Pratt & Whitney Aircraft Division, United Aircraft Corp., East Hartford, Conn. (mail) 315 Pearl St., Hartford, Conn.

Washington Section

Hill, E. Govan (M) president, Automatic Shifters, Inc., Richmond, Va. (mail) 3110 N. Boulevard.

Hughes, H. Herbert (A) economist, Automobile Manufacturers Association, 830 Transportation Bldg., Washington.

Outside of Section Territory

Kearns, Earl E. (M) head, urban transportation section, General Electric Co., Erie, Pa. (mail) 2901 E. Lake Rd.

Reeves, Thomas (J) development engineer, Continental Motors Corp., Muskegon, Mich. (mail) 1676 Smith St.

Foreign

Hutchison, Robert (F M) works manager, chief engineer, Port Elizabeth Electric Tramway Co., Ltd., Port Elizabeth, South Africa (mail) P.O. Box 225.

SAE Coming Events

Jan. 12-16, 1942 SAE Annual Meeting (and Engineering Display) Book-Cadillac Hotel – Detroit, Mich.

May 31 – June 5 SAE Semi-Annual Meeting The Greenbrier – White Sulphur Springs, West Va.

Baltimore – Dec. 11

Engineers Club; dinner 6:00 p.m. Aircraft Radio – W. L. Webb, chief engineer of Radio Receivers, Bendix Radio Corp.

Buffalo – Dec. 16

Joint meeting with American Society of Mechanical Engineers. Subject: Die Casting.

Chicago – Dec. 2

Hotel Knickerbocker; dinner 6:45 p.m. Magnetic Couplings and Electric Marine Transmissions – Martin P. Winther, vice president and chief designing engineer, Dynamatic Corp.

Detroit – Dec. 8

Hotel Statler; dinner 7:00 p.m. The Effect of the War on Engineering Progress – C. F. Kettering, general manager, Research Laboratories Division, vice president, General Motors Corp.

Indiana – Dec. 11

Plastics in National Defense – H. A. Frommelt, director of industrial research, Kearney & Trecker Corp.

Metropolitan – Dec. 11

Park Central Hotel; dinner 6:30 p.m. Student Debate – Resolved: That the air-cooled type of aircraft engine is to be preferred to the liquid-cooled type for military aviation. Affirmative – New York University. Negative – College of the City of New York.

Milwaukee – Dec. 5

Plant Inspection Trip through McCulloch Engineering Co., Milwaukee, 3:00 to 5:00 p.m. Milwaukee Athletic Club; dinner 7:30 p.m. Development of Mechanical Superchargers in Present Applications – John Ryde, chief engineer, McCulloch Engineering Co.

Northern California – Dec. 9

Hotel Leamington, Oakland; dinner 6:30 p.m. Some Results of Valve Gear Research – Carl Voorhies, research engineer, Wilcox-Rich Division, Eaton Mfg. Co. Precision Governing of Diesel Engines – G. F. Drake, chief engineer, Woodward Governor Co.

Northwest – Dec. 11

Crawford's Seafood Grill, Seattle; dinner 7:00 p.m. Fuels for Internal Combustion Motors: Diesel Fuel – Roy Severin, manager, Gas Tank Service Corp. Gasoline – John Stein, manager, Dependable Tank Corp. Butane – William Miller, master mechanic, Pacific Highway Transport.

Philadelphia – Dec. 10

Engineers Club; dinner 6:30 p.m. Engine Bearings – A. B. Willi, Federal Mogul Corp.

Pittsburgh – No meeting

St. Louis – No meeting

Southern California – Dec. 12

Elks Club, Los Angeles; dinner 6:30 p.m. Round Table Discussion of Fuels and Lubricants.

Washington – Dec. 9

Lee Sheraton Hotel; dinner 6:30 p.m. Anti-Tank Defense – Capt. G. W. White, U. S. Army, Office of Chief of Ordnance, Tank and Combat Vehicle Division.

Tulsa and Wichita Groups – Dec. 6

Allis Hotel, Wichita, Kan.; dinner 6:30 p.m. Simplifying Fuels and Lubricants for Mechanized Fighting Forces – C. M. Larson, chief consulting engineer, Sinclair Refining Co. Inspection Trip through Cessna Aircraft Co. plant to be held in the afternoon, leaving Allis Hotel at 3:00 p.m.

National Aircraft Production Meeting

(*Continued from page 18*)

SAE registration desk at 1941 National Aircraft Production Meeting was a busy place

A large crowd gathered to inspect the new Stout multipiece crankshaft

Fuel Feed at High Altitude – W. H. CURTIS and R. R. CURTIS, Thompson Products, Inc. (Paper presented by W. H. Curtis)

Presentation of this paper, stressing the laboratory work which has been done in investigating problems of fuel feed up to altitudes of 40,000 ft, rather dramatized the great intensification which has taken place recently in laboratory research of all kinds. We talk quite casually of fuel flow problems at 40,000 ft altitude, the authors pointed out, although practically no work dealing with problems at such altitudes had been done until recently. This paper presented a detailed report on a specific investigation of the efficiency of a centrifugal pump located at the fuel tank in retarding vapor lock in the fuel line. Effect of dissolved air in the fuel was treated, as well as fuel vapor pressures. The centrifugal pump at the tank works as a booster pump with the function of supplying fuel to the engine pump. It is cut in and out as required by conditions of climb and altitude operation. Action of the booster pump is not yet fully understood and the investigation is continuing. One feature of the pump is that it releases air and vapor from the fuel. Sufficient capacity must be provided in the feed to the centrifugal pump to permit escape of the vapors formed without retarding too greatly the flow of liquid fuel into the pump chamber. As altitude increases the formation of vapor increases at the pump until it becomes vapor locked even though submerged in liquid fuel. A fascinating feature of this paper was the remarkable series of slides showing high-speed photographs taken of vapor formation in an experimental tank under various conditions of temperature, altitude, and pump speed.

DISCUSSION

Walter Wise, Douglas Aircraft Co., inquired why the booster pump would fail at less than its capacity. Mr. Curtis replied that failures occur from various causes or combinations of causes, such as high rate of climb, high temperature of the fuel, improper venting, etc. Mr. Wise asked if the pump could be operated in the tank with the shaft located horizontally, instead of vertically, as shown in the laboratory tests. The reply was: "Yes, but that the capacity would be limited somewhat." Mr. Curtis further said that tests with centrifugal pumps had revealed "sinking spells" which were completely unaccountable and further investigation was required.

Donald Douglas, Jr., Douglas Aircraft Co., asked if tests had been run with the fuel heated in the lines to simulate a nacelle installation. Mr. Curtis said this had not been done, since such temperature rises had not proved at all troublesome in practice except for temperatures at the carburetor itself. This problem is under investigation, he added.

Lee G. Snyder, Vega Airplane Co., inquired about the relationship between the fuel temperature and pump speed. This is almost directly a factor of the vapor handling capacity of the pump installation, replied Mr. Curtis.

Mr. Douglas asked if the 1½ lb internal pressure in self-sealing tanks was of any advantage in suppressing pump failure and Mr. Curtis said definitely yes.

Have tests been made on fuels of greater than 100 octane rating? asked P. G. Smith, of Douglas Aircraft. Not yet, said Mr. Curtis, but they are planned.

Edward E. Gould, Lockheed Aircraft Corp., wondered about the limitations of the pump-tank combination. Would it not be of advantage to have the pump mounted as low as possible with respect to the tank in order to keep a full head of liquid on the pump as the fuel level dropped in the tank? And would high-speed operation require a larger feed opening between tank and pump chamber? Mr. Curtis replied that the greater the head that could be provided the more efficiency could be obtained from the pump, but this is controlled by airplane structure, tank location, and other practical considerations.

Quoting Edward Warner as saying that the side in this war which had superiority in high-altitude flying would have the equivalent of 2,000,000 troops on its side, Fred C. Crawford, president, Thompson Products, Inc., gave as his belief that the side which first solves the problems of high altitude airplane operation will win the war. He stressed the great value of the new $21,000,000 NACA engineering laboratory now nearing completion near Cleveland.

A. E. Smith, Lockheed Aircraft Corp., asked if tests had been run on recirculated fuel. Mr. Curtis replied that circulating the fuel had little effect on it but that the altitude effect made it impracticable to repeat a test with fuel which had been subjected to high-altitude conditions, due to removal of vapors.

Winford J. Lane, also of Lockheed, suggested the importance of work on controlling fuel tank temperatures. Work is under way along this line, replied Mr. Curtis, but it is not believed to be a vital factor.

Dr. O. C. Bridgeman commented favorably on the paper, but asked if there is positive evidence that vapor lock occurred in some cases at 3 to 10,000 ft lower altitude than predicted on the basis of fuel characteristics, since his own experience had been that vapor lock normally occurs at about 5000 ft higher than predicted. Mr. Curtis said that there were many cases to prove that vapor lock does occur at times at the lower levels.

Following this session there was an interesting demonstration of a new type of recorded sound lecture illustrated with slides. This particular lecture was titled "The Engineer's Relation to Assembly" and was presented by Trade Films, Inc., which has developed a whole series of such educational illustrated lectures for use in training of aircraft personnel.

ENGINES & AIRPLANES SESSION
Henry J. E. Reid, Chairman

In the absence of Dr. G. W. Lewis, National Advisory Committee for Aeronautics, Henry J. E. Reid, NACA, served as chairman. A capacity crowd jammed the auditorium for one of the most interesting sessions in the history of this series of meetings. Two papers were presented, one dealing with the development of two-speed supercharger drives by the designers of various countries, the other outlining the remarkable work by

W. B. Stout in the development of a new light airplane and engine.

Two-Speed Supercharger Drives – F. M. KINCAID, JR., Wright Aeronautical Corp.

Perhaps the most impressive feature of this illustrated paper was the exposition of the extremely precise manufacturing work required to produce gear assemblies which are of extremely light weight and compact arrangement for the high power transmitted. The American Wright two-speed drive, British Bristol, and Rolls-Royce "Merlin," German Junkers Jumo, and Mercedes-Benz units were all described in detail, together with diagrams and cross-section drawings of the various units done in two and three colors to simplify explanation of their operation.

DISCUSSION

A. T. Gregory, Ranger Aircraft Engines, asked concerning the relative merits of the British Bristol three-clutch unit and the American Wright two-clutch transmission. The reply by Mr. Kincaid stressed the devices used by the British to protect against overloading, and the fact that the British units were relatively heavy. Our system transmits equivalent power for little more than half the weight, he said. European supercharger drives generally are somewhat heavier than American practice.

The Small Airplane and Its Power Plant – W. B. STOUT, Stout Engineering Laboratories, Inc.

The inimitable "Bill" Stout is always entertaining and on this occasion was at his best. At the close of the talk he was swamped by the audience as they rushed the platform for a look at his new multi-piece crankshaft, centrifugally cast cylinder, and other parts he had on display. Devoting most of his paper to the new engine, Mr. Stout opened with a discussion of his work in developing a light airplane of stainless steel construction. Basic purpose, he said, is to solve the problems of stainless steel engineering and fabrication on very small planes with thin material because these problems become simpler and production is easier as the size increases. The Stout plane is now nearly ready to fly. Basic work has been done in developing a ductile alloy of stainless steel and in finding methods of forming, spot welding, and assembling various parts. Spot welding is now being done with less than one cycle (60-cycle current), on sheet material where 0.005 in. thicknesses are welded to 0.040. The thin sheet material on the new Stout plane is kept from wrinkling and deforming by the introduction of air pressure inside the wing when in flight. Strength of the wing is materially increased by this trick.

The new Stout engine, which is being developed in an effort to achieve the ultimate goal of a 100-hp engine to weigh 100 lb and cost $100, is designed around the multipiece crankshaft introduced some years ago by Jules Dusevoir. This shaft greatly simplifies manufacture of the entire engine since both case and connecting rods can be of one piece, bearings can be standard roller bearings. The shaft is cheaper and easier to build and it is possible to use high strength materials, and to vary the materials in different elements of the shaft. Claimed to be stronger and more precise in finished tolerances than a one-piece shaft, the Stout-Dusevoir shaft is assembled from crankpins and throws which are joined by means of semi-circular serrations for positioning. Bolts through each crank pin hold the parts together. Such shafts have proved rugged in extensive tests and it is believed they will have wide application. But the chief advantage is the simplification of the entire engine structure which is made possible. Other novel features are incorporated in the engine, such as centrifugally cast steel cylinders and a two-speed propeller gearing. The present engine weighs 120 lb, develops 120 hp, and is relatively cheap to manufacture. However, Mr. Stout presented all this as an interesting experiment and not as an accomplished fact at this time. Much operational testing remains to be done before the many novel features will be brought to the point of wide acceptance.

DISCUSSION

Chairman Reid immediately opened the discussion period with the most embarrassing question possible, "What is being done to reduce the cost of the many accessories that must be mounted on an engine?" Mr. Stout admitted that he was temporarily stumped on that point but hoped accessory costs could be somewhat reduced to match the engine achievement.

Further information on the reduction gearing feature was requested by Arthur Nutt. The gear is automatic, replied Mr. Stout, permitting the engine to turn up 3600 rpm for take-off with a propeller speed of 2500 rpm, and then shifting into high position as the 2500 rpm is reached.

Richard Mock, Lear Avia, Inc., inquired about the form of serrations. Mr. Stout said they are concentric circles, easily and cheaply formed with great precision.

"How about using hydrogen copper welding to weld fins on a cylinder?" inquired C. K. Greene, Triplett and Barton. Mr. Stout thinks the copper-hydrogen welding process is now made obsolete for this purpose by later developments, such as centrifugal steel casting.

Jack F. Cox, Vega Airplane Co., wanted to know if the 0.006 in. thick stainless steel wing skin "oil-canned" while the plane was taxiing along the ground without benefit of internal air pressure. Yes, Mr. Stout replied, there would be some trouble with this and all suggestions were welcome. He is applying an internal absorbent coating to the wing skin to reduce this effect.

R. F. Stoessel, Lockheed Aircraft Corp., asked how the wing skin could be spot welded to the structure. A portable welding gun has been developed, Mr. Stout said.

Dr. A. L. Klein thought he had Mr. Stout cornered when he asked how much 0.005-in. stainless steel sheet cost per lb. But "Bill" replied he didn't know how much it cost, but he could get it, which was not true of

Important standardization progress was made when the SAE Aircraft Materials and Processes Committee met at the Hollywood-Roosevelt Hotel, shortly after the close of the meeting

Standing, left to right: L. D. Bonham, Lockheed Aircraft Corp.; J. B. Johnson, Wright Field; B. Clements, Wright Aeronautical Corp.; J. D. Redding, SAE Staff Representative, Aircraft and Aircraft-Engine Activities

Seated, left to right: A. F. Barnard, Vultee Aircraft, Inc.; Eric Dudley, Curtiss Aeroplane Division; J. N. Huff, Curtiss-Wright Corp., Propeller Division; W. H. Graves, Packard Motor Car Co.; R. L. Heath, Allison Division, General Motors Corp.; H. J. Noble, Jacobs Aircraft Engine Co.; G. D. Wait, Consolidated Aircraft Corp.; Jack F. Cox, Vega Airplane Co.; C. E. Stryker, OPM; M. P. Crews, Civil Aeronautics Administration; A. W. F. Green, Pratt & Whitney Aircraft; C. E. Carrigan, Ranger Aircraft Engines; F. P. Gilligan, Henry Souther Engineering Co.; D. H. Brown, Bendix Aviation Corp.; Donald Haley, Douglas Aircraft Co.; R. E. Pelzell, Douglas Aircraft Co.; F. S. Klock, Hamilton Standard Propellers

aluminum. This brought down the house.

"Have any applications of the Dusevoir joint been made to long crankshafts where torsional vibration problems are serious?" asked Joseph Johnson, of Lockheed. Not yet, said Mr. Stout, but he believes the joint is practical in any size shaft, including those used on ocean liners.

At the close of the discussion the audience was given an opportunity to examine the crankshaft closely, assembling and disassembling it.

AIRCRAFT SESSION

John K. Northrop, Chairman

Three papers were presented, two of them strictly production subjects. The third dealt with the annual report of the CFR investigation of vapor lock. One of the production papers dealt in detail with the experiences of the Briggs Mfg. Co. in getting into production on aircraft assemblies. The other was a description and demonstration of the new du Pont explosive rivet.

Off the Ground – Into the Air – H. H. BUDDS, F. W. HOFMANN, E. E. LUNDBERG, H. T. PLATZ, and H. J. ROESCH, Briggs Mfg. Co.

This paper, prepared by five authors, was presented by a sixth, Joseph Geschelin, Detroit technical editor, Chilton Co. Contrary to the old adage that "Too many cooks spoil the broth" this was a most interesting, informative, and encouraging paper. Written in a spirit of cooperation between the automobile and aircraft industries, the paper started with the admission that mass production methods are not directly applicable to most aircraft manufacturing work and that the automobile people still have much to learn from plane builders. Considerable stress was laid on the development of efficient assembly jigs and on the training of personnel. Specific training is given in employee attitude, in an effort to instil workers with a feeling of importance in the whole production scheme so that they may take more pride in the operation, no matter how small, which each one performs.

DISCUSSION

Conrad O. Rogne, Vega Airplane Co., suggested that there is still much to be done in the way of breaking down production into smaller individual assemblies. Mr. Geschelin agreed to this suggestion and remarked that the authors had had little opportunity to investigate this phase prior to preparation of the paper, as their work to date had dealt with manufacture of parts to detailed specifications. Some of the points suggested in the paper are found already to be in practice in some of the Southern California plants, which are believed to be far advanced in development of efficient production techniques.

Chairman Northrop commented that one benefit of the present emergency had been the disclosure by various companies of advanced production methods which had formerly been kept under cover. This trend was helping to speed all production work through collaboration among various aircraft and automobile firms.

Can riveting be done outside the jig without deforming the structure? asked Dr. A. L. Klein, California Institute of Technology. Mr. Geschelin replied that most of the riveting was done on positioning jigs after removal from the assembly jig.

Chairman Northrop asked if the troubles recounted by the Briggs people could be considered as typical of those among other auto makers turning to aircraft work. Mr. Geschelin opined such was the case and that the auto people have much to learn. Eventually, he believes, improved methods would be contributed by the auto people just as they are doing in armament and munitions manufacture. For example, the auto industry is now doing horizontal internal broaching of rifle barrels, a process never accomplished before.

John W. Dunn, Curtiss-Wright Corp., commented that many auto makers are freely admitting their ignorance of aircraft production methods and are complaining that they cannot get the aircraft people to cooperate. Mr. Northrop suggested that the aircraft people have so many troubles of their own, and executive personnel has been spread so thin, that there simply isn't time for much outside consultation.

The du Pont Rivet – A New Blind Fastener for the Aircraft Industry – L. A. BURROWS and D. L. LEWIS, JR., E. I. du Pont de Nemours & Co. (Presented by D. L. Lewis, Jr.)

This paper was particularly interesting for its on-the-stage demonstration of explosive riveting. Chief present application of the explosive rivet, Mr. Lewis said, is in riveting blind sections where it would be difficult or impossible to drive standard rivets. Eventually, Mr. Lewis speculated the explosive rivet may be as cheap to drive as standard rivets and so may find wide application throughout all riveting. Problems of corrosion, he declared, have been solved completely through use of noncorrosive chemicals. Rivets are manufactured and loaded with explosive to extremely precise limits. Tests that show the explosive rivet joint to compare favorably with driven rivets were discussed.

Clarence Johnson Awarded Wright Brothers Medal

Presentation of Wright Brothers Medal: Clarence L. Johnson, medal recipient, and Henry J. E. Reid, who made the presentation

Clarence L. Johnson, chief research engineer, Lockheed Aircraft Corp., was awarded the 1940 Wright Brothers Medal for a paper entitled, "Rudder Control Problems on Four-Engined Airplanes." Presentation of the award was made by Henry J. E. Reid, chairman of the Medal Board, during the three-day National Aircraft Production Meeting at Los Angeles.

The Wright Brothers Medal is awarded annually to the author of the best paper on aerodynamics or structural theory or research, or airplane design, or construction, presented at a meeting of the Society or any of its Sections during the calendar year.

Mr. Johnson has been active in flight testing, stress analysis and design work ever since he joined Lockheed in 1933. He went to England in connection with design and later testing of the Hudson bomber. In his present position he has charge of airplane design, tunnel and flight testing.

Born in Ishpeming, Mich., on February 27, 1910, Mr. Johnson obtained his B.S.E. degree in aeronautical engineering at the University of Michigan in 1932. He was awarded the Sheehan Fellowship for Aeronautics in 1933, and worked on boundary layer control work for an M.S. degree. Graduating the same year, he went with Lockheed as aerodynamicist and flight test engineer. In 1937, he was given the Sperry Award by the Institute of Aeronautical Sciences for "Important Improvements in the Design of High-Speed Transport Airplanes." He became chief engineer of Lockheed in 1938.

The medal-winning paper was delivered at the 1940 SAE Annual Meeting and later published in full in the June 1940 SAE Journal (pp. 262-270).

(Left to right) Ellis W. Templin, past chairman of Southern California Section, and Robert E. Roy, U. S. Engineering Corp., shown with revolving SAE globe and planes designed and built by Mr. Roy and used in connection with the membership booth

DISCUSSION

Robert Nolan, Frank Wiggins Trade School, asked if du Pont is following the AN code on rivet designations. Mr. Lewis replied that this procedure was not practicable due to differences in shank size and manufacturing tolerances, but a simplified color code system has been developed, making it very difficult to mix rivets.

How can the inspector tell if the rivet on a blind joint has been exploded? inquired Wayne H. Allen, of General Electric. By pressure on the sheet adjacent to the rivet, replied Mr. Lewis.

Dr. A. L. Klein, Cal-Tech, asked about the reaction if the rivets were gun-driven by mistake, or got tossed into a salt-bath heat-treat. Mr. Lewis said either condition would probably cause the rivets to explode, but without serious trouble.

On the matter of costs, Mr. Lewis said that standard-driven rivets cost an average of about 7¢ overall. Explosive rivets installed average about 7¢. But some tough riveting jobs, done at the 7¢ rate with explosive rivets, cost from 25¢ to as high as $3 by other methods. In Germany, he reported, explosive rivets are replacing driven rivets to a high degree. Marked saving in labor required, he revealed, results from elimination of the bucker-up.

Aviation Vapor-Lock Investigation – Cooperative Fuel Research Committee Report – Dr. O. C. BRIDGEMAN, Leader, Survey Group, CFR Aviation Vapor-Lock Projects.

Illustrated with slides, this paper brought the audience up to date on progress of the continuing work of the CFR committee. It brought out that test procedures have been further developed and investigation has been carried into the hydraulic liquid field, in addition to fuel research.

DISCUSSION

A. L. Stanley, Shell Development Co., asked if investigation had been made of fuels with same vapor tendencies but different physical composition. Quite a bit has been done along this line, Dr. Bridgeman said.

W. C. Lawrence, OPM, inquired why there was such a wide divergence in data obtained with a drop of ½ psi as against a drop of 1 psi in the fuel line. Dr. Bridgeman replied that it has been found that, contrary to what one would suppose, the action of fuel vaporization is not directly variable with pressure.

Which is the more important, asked Fred Hosterman, Lockheed Aircraft Corp., to maintain pressure in the fuel tank, or to decrease temperature of the fuel? Maintenance of pressure is far more vital than considerations of temperature, replied Dr. Bridgeman.

Isn't it possible to perfect a method for extraction of the air from the fuel, inquired Weldon B. Allbaugh, Lockheed Aircraft Corp. Dr. Bridgeman replied that the only practical solution is to increase the vapor-handling capacity of the system.

D. P. Barnard, Standard Oil Co. (Ind.), reported that gasolines will dissolve about 17-18% of air by liquid volume and that venting systems must take this large volume of air into account. It had been found, he said, that vents should be able to handle at least five times the volume of the liquid flow. Dr. Bridgeman agreed that this was a correct statement.

Willson H. Hunter, B. F. Goodrich Co., asked if restricting the vent size might not help. In some cases yes, answered Dr. Bridgeman.

AIRCRAFT SESSION
Mac Short, Chairman

Production was the theme of this session One paper, presented by C. E. Stryker, stressed the important relationship between standardization and production. The other, by George Tharratt, showed how production can be speeded through substitution of perspective assembly drawings in place of blueprints on the final assembly line.

Standards for Defense – C. E. STRYKER, Office of Production Management.

Since our first line of defense is the "production line," it is unfortunate that the tremendous expansion of production came before important standardization work already under way could be concluded, Mr. Stryker emphasized. As important as standardization is we must now take as our standards keynote that, if it is not absolutely necessary to set up a new standard then it is absolutely necessary not to set up that standard, he brought out. Standards come under two major classifications: A. production standards, and B. development standards. Forms of standardization can be classified as drawings, charts, diagrams, curves, etc., or specifications. Under specifications are materials, parts, performance specifications, processes.

Two forms of tangible results are developing from present standards work, Mr. Stryker concluded.

1. An increased number of Government standards are being issued.

Space in the Engineering Display was sold out weeks in advance of the meeting, indicating the continuing popularity of this feature. Featuring extensive presentations of materials, products, and accessories, used in aircraft production, exhibitors were as follows: Aeroquip Corp., hose lines and fittings; American Bosch Corp., ignition equipment; Bendix Aviation, Ltd., hydraulic parts and accessories; E. G. Budd Mfg. Co., welded stainless steel assemblies; The Cleveland Pneumatic Tool Co., landing gear assemblies; Ducommun Metals & Supply Co., tools, gages and metals; International Nickel Co., Inc.; Interstate Aircraft & Engineering Corp., hydraulic and other component parts; The Lamson & Sessions Co., screw products; Magnaflux Corp., working demonstration of magnaflux testing; Menasco Mfg. Co., hydraulic struts and parts; The National Screw and Mfg. Co., screw products; The Parker Appliance Co., aircraft plumbing; Purolator Products, Inc., special oil cleaners; Rohm & Haas, Plexiglas plastic parts; Ryan Aeronautical Co., the patented Ryan ball and socket manifold system; Simmonds Aerocessories, aircraft accessories; Solar Aircraft Co., stainless steel exhaust manifolds; Swedlow Aeroplastics Corp., plastic aircraft parts; Tinnerman Products, Inc., speed nut fastening devices; Turco Products, Inc., special aviation cleaning compounds; and Wright Aeronautical Corp., a Wright Cyclone engine.

A Few of the Booths at the Engineering Display

2. Many industrial standards are being established, representing agreements between or among competing industrial organizations.

DISCUSSION

Chairman Mac Short commended the speaker on his concise presentation of an involved and difficult subject.

J. B. Johnson, Wright Field, reported that 150 standards relating to materials and processes have been established from Wright Field.

Col. D. G. Lingle, U. S. Army Air Service, emphasized the tremendous amount of work remaining to be done in standardization.

Lt. Harry J. Marx, U. S. Navy, Bureau of Aeronautics, reported that the standardization problem is a deeper one than many people realize. There is real danger of sectionalism developing, he brought out, of certain groups favoring products with which they are familiar. Much depends on studies of service reports.

H. J. Alter, Vega Airplane Co., commented that servicing military aircraft calls for maximum possible standardization in order to simplify the problem of training green ground personnel.

Lt. Marx contributed the comment that our Army Air Service is using more than 80 different sizes of airplane tires, whereas the Germans use but 25 sizes. It is difficult to design landing gears, he said, both tires and brakes, because designs "grow." Armament and other loads are added to increase greatly the gross load of the plane. One model was increased 50% above the prototype. This practice overloads tires. Brakes have been failing in England, he reported, because planes are dispersed far from operational centers. Sometimes the plane is taxied as much as ten miles to reach the take-off point. Brakes not designed for such work are failing.

Illustrated Production Breakdown – GEORGE THARRATT, Douglas Aircraft Co. (Now appointed chief engineer, Adel Precision Products Corp.)

This was one of the most entertaining papers of the meeting, not only because it was very liberally illustrated with unusually interesting drawings, but because of the off-the-cuff delivery of the author, a former Scotchman who still retains his thick brogue. The gist of the Tharratt method of production illustration, now being widely used by a number of aircraft companies, is the replacement of all blueprints on the final assembly line with perspective drawings showing the assembly as it should look when completed. Such drawings are much more easily followed by the workmen and can carry a great deal more information in usable form than can the ordinary working drawings, the author contended. Big problem, he said, is to get engineers who have talent for such illustrating. Such men, when trained for production illustration, are usually so valuable that they are promoted to more important work. To solve this problem, he reported, it has been necessary to develop illustration groups, using an engineer-illustrator as lead man and with specially trained commercial artists for the more routine work. Especially in view of the inexperience of new men now being drafted into aircraft production work, he emphasized that it is necessary to simplify instructions. This is best demonstrated by the Chinese proverb: "One picture is worth ten thousand words." A good production breakdown illustrator, Mr. Tharratt contended,

will save hundreds of man-hours of time on the production line. Many of the men now assembling airplanes would normally be soda jerkers, beer truck drivers or "prune pickers."

DISCUSSION

Such drawings probably have wide application in engineering as well as in production, suggested Lee G. Snyder, of Vega Airplane Co. Mr. Tharratt confirmed this point. For example, a pictorial copy of the mock-up is made available in perspective drawing form to all engineers working on a new design. This procedure saves many hours of time spent in referring to the mock-up.

Chairman Short said that such production drawings had proved to be the solution of the problem of transporting Boeing bomber production more than 1200 miles from Seattle to Burbank and Long Beach as is being done at this time under the B.V.D. plan (Boeing, Vega, Douglas). Without liberal use of such drawings, he said, it would have required the transfer of complete trained crews to the new plants now going into production on "Flying Fortresses."

Have such drawings reduced the size of the engineering liaison group? asked H. J. Alter, of Vega. Mr. Tharratt replied in the negative – the new illustrating system has simply helped speed their work.

Lt. Harry J. Marx suggested that such drawings be incorporated in field service and maintenance manuals for the use of mechanics. Mr. Tharratt replied that this was already being done on a wide scale.

J. E. Shea, of Vega, asked how such drawings are kept up to date with engineering and production changes. Mr. Tharratt replied that, once the drawing is delivered to the production floor, it is the responsibility of the crew there to see that changes are added. When such changes come to the production men, they bring them to the attention of the illustration department.

This session also was again followed by a Trade Films demonstration of a slide illustrated transcribed lecture, this one being on the subject of mechanic training. A second lecture was also shown under the title: "The Engineer's Relation to Production."

GENERAL SESSION
Carl B. Squier, Chairman

This Friday evening general session has developed into a tradition at the National Production Meetings, and again the hall was packed to the rafters. Among the distinguished guests was the Hon. Robert Hinckley, Assistant Secretary of Commerce for Aeronautics.

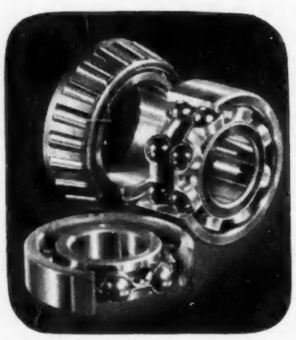
Blitzkrieg and How to Stop It – A. T. COLWELL, Thompson Products, Inc., and President, SAE.

Mr. Colwell opened with the announcement that a Western Office of the SAE is being opened at once in Los Angeles, under the direction of E. F. Lowe, SAE assistant general manager.

In speaking of the blitzkrieg, Mr. Colwell treated it from the technical rather than the military standpoint. Having studied the recent blitzkrieg tactics of the Germans as a personal hobby, Mr. Colwell had drawn deeply on established military authorities for information and opinions concerning its methods. He first remarked that it was much over-publicized, being represented as a combination of Buck Rogers and Superman in the German army uniform. The truth is that it has been a battle of the future against the past. The Germans have simply thrown organized mechanization against old weapons and old ideas of warfare. The science of war remains unchanged but the art of applying the principles of war has been revolutionized by development of the internal-combustion engine. The German army simply took all the new ideas developed throughout the whole world and put them all together in one place. The basic idea of the panzer forces is to develop a sustained break-through, with unity of action by all elements. Experiences in Poland taught the Germans new lessons about massing armored forces and operating them in close cooperation with air forces. The whole German war concept was based on offensive action rather than defensive inaction. Out of a study of German methods and German successes comes the conviction that it is simply a question of developing superior industrialization, greater manufacture of armored units, thorough preliminary training, and the blitzkrieg of the panzer forces can be stopped short by the opposition of an equal or greater force of like composition. Key to mechanized action is maintenance of equipment. And the key to successful action is air superiority. Never again will there be an important victory on either land or sea without first gaining control of the air, Mr. Colwell emphasized. America's tremendous capacity for industrial activity and her superior design developments in the fields of high-altitude flying, bombsights, supercharging, high octane fuels, and in many other ways, makes it clear that we can develop the power to stop the blitzkrieg. SAE President Colwell closed with a plea for all-around cooperation in industry to get the job done. He urged that the New Deal be replaced by the "Square Deal." No discussion was permitted due to the lateness of the hour.

Morale – An Account of Our Production Position – T. P. WRIGHT, Office of Production Management.

T. P. Wright, as chief consultant to the OPM on aircraft production problems, is



probably the world's outstanding authority on aircraft production matters. His report was most heartening for he not only made a plea for improved morale, but gave us specific reasons why our morale should be better now than a year ago. Although there was considerable justification for criticism of our production progress as of a year ago, he said, we have now accomplished production miracles and we should show proper confidence in the leaders who have made this possible. Our rate of production increase during the past year has been faster than it ever was in either Germany or England. While we still are faced with many production problems, the most troublesome are already behind us. By next summer our entire program should be rolling smoothly and at a rate surpassing anything that would have been thought possible as recently as last summer. Time is still the chief of all bottlenecks but, with every month that we keep England from defeat, the chances for final victory grow brighter.

Of all our specific troubles the most vital is labor, Mr. Wright declared. We must find a way to eliminate strikes and their attendant upsets to planned schedules. Wage stabilization must be sought and labor must be made to feel some sacrifice in our defense effort, perhaps through reinstatement of the 48-hr work week. Strong action against strikes would have a salutary effect on all labor troubles. We can draw up a general blueprint for victory although the specific steps are not easy to define. Our plan should be – bombs – blockade – and rebellion. By winning dominant air superiority we eventually will break the German hold on conquered territories. For the future we must look to some form of collective security which will prevent a repetition of the present holocaust. His talk closed with a quotation from a recent speech by President Roosevelt: "We must produce planes, tanks, guns, in a stream that will grow to a river, a torrent that will engulf the tyranny which seeks to dominate the world."

Prior to the close of the meeting Carl Squier introduced Assistant-Secretary of Commerce for Aeronautics Robert Hinckley. Mr. Hinckley praised the accomplishments of California in the field of aviation, pointing out that 10% of all planes and pilots of the entire United States are located in that state. He commented on the fact that this one area of the country was producing almost half the airplanes now rolling out of the factories and expressed the hope that other areas might more nearly match this accomplishment in the future.

High point of this session was the presentation of the Wright Brothers Medal to Clarence L. Johnson, Lockheed Aircraft Corp., for the best SAE airplane paper of 1940. Mr. Johnson's paper was presented in Detroit at the SAE Annual Meeting, Jan. 15, 1940. Presentation of the Wright Brothers Medal was made by H. J. E. Reid, of the NACA.

AIRCRAFT SESSION
Stanley A. Bell, Chairman

Production and design considerations balanced the session. The first paper outlined the importance of quality control in achieving a smooth production flow. The second took under consideration the critical weight control problem, with hard-won performance gains all too often canceled out by installation of heavy armament or accessories.

Airplane Quality Control – J. W. DUNN, Curtiss-Wright Corp.

This paper was liberally illustrated with slides to show tables of organization suggested in setting up quality control, and also with photos of special tools, gages, and fixtures used for checking dimensions, especially of odd and irregular parts. Suggested major points of quality control in the order of their importance were given as: 1. Safety, 2. Performance, and 3. Interchangeability. To illustrate the magnitude of the quality control problem the author referred to a standard model pursuit plane that contains about 16,000 detail parts, 100 major and minor assemblies, and 72,000 rivets and bolts. A total of about 152,000 inspection operations are required for each plane. Subcontractor, or vendor inspection is a new factor which has increased tremendously in the past year. Best method of handling this, he said, is to set up resident inspectors in major subcontract plants. An inspection manual serves as a guide for all inspectors. A salvage department is vital in preventing wasteful rejection of parts which can be saved.

Will Accessories Impede Our Payload? – L. R. HACKNEY, Lockheed Aircraft Corp.

While this paper might have been interpreted as a complaint against the accessories manufacturers, any such thought was immediately dispelled by the author himself in complimenting those organizations for the active cooperation already rendered in developing weight-control principles. Addition to military plane designs of heavier armor and armament, heavier self-sealing fuel tanks, high altitude supercharging and pressuring equipment, and so on, all have made it more important than ever that every possible ounce of weight be saved in the plane

itself, or in its accessories and standard equipment. This objective is particularly difficult to attain because the average weight of such items as communication equipment, furnishings, anti-icing equipment, electrical items, has been rising steadily. A comparison of a 1941 transport design shows the total weight of such items amounts to 221 lb per passenger as compared with 149 lb per passenger on a standard transport of 1939 design. Aircraft manufacturers are spending a total of more than $1,500,000 per year on weight-control personnel, the author revealed. But even the expenditure of twice this amount, he said, would accomplish negligible results without the cooperation of the equipment suppliers. He proposed that an approved-weight stamp be issued for all equipment items which meet certain minimum weight standards, together with other factors dealing with efficiency and cost. He believes that purchasing departments should be made more weight-conscious so that consideration is given to weight as a more vital factor in buying competitive items.

DISCUSSION

Prepared discussion was presented by Walter A. Semion, Hughes Aircraft Corp. Many weight savings were suggested by Mr. Semion, such as greater use of magnesium alloys, light plastics, simplified structures, higher heat-treat on steels. A novel suggestion is that designers in the future overestimate the weight of the tail assembly. It is better, he believes, that the completed design should be nose-heavy than tail-heavy and the gross weight can be controlled easier under such a condition. Strength factors may be reduced somewhat, he believes, on such items as castings, welded joints, and so on, producing further weight savings.

A further suggestion was that removable equipment be incorporated in air liners so that they carry only such items at all times as are absolutely necessary. Chairs, buffet, berths and bedding, and various such items might be left at way stations or picked up as needed.

Dr. A. L. Klein remarked that the accessory people are to blame for many features of design in which improvement could be achieved through increased cooperation. Accessory flanges are frequently not sufficiently sturdy to carry units under heavy operational load and additional heavy brackets have to be supplied. As for the removable-equipment idea, he doubts if that would work because there are always more people going into cities than coming out of them, based on a lengthy statistical compilation. This would mean that all the equipment would pile up in the cities.

G. H. Hill, Airesearch Mfg. Co., remarked that performance is frequently as important to the overall weight as is the specific weight of the accessory. For example, two intercoolers may be designed for the same job but the heavier one may be more efficient and so be the proper installation.

Robert G. Hoof, Bendix Aviation, reminded the audience that weight savings could never be allowed at the expense of performance.

Dr. A. L. Klein brought up a point rarely ever mentioned: that the allowable tolerances on aluminum sheet material are so great as to result in a possible overweight of 25% on that one item if loads are figured to the minimum tolerance and all sheets run the maximum. There is at present no way to guard against this discrepancy, he brought out, and closer tolerances are to be hoped for.

Richard M. Mock, Lear Avia, pointed out that, like airplanes, accessories grow in weight as experimental designs are put into production. It is important, he emphasized, that the purchasing department know whether they are purchasing from specifications covering an experimental or a production design.

Increased efficiency of electric motors frequently pays back more than the required increase in weight which may result, according to C. W. Morris, General Electric Co.

W. C. Lawrence, OPM, expressed surprise that reliability and service life had not been mentioned in connection with weight. These factors must be evaluated in determining allowable weight, he emphasized.

T. P. Wright, OPM, related the story of an early torpedo plane which was found to be overweight exactly the amount of the weight of the torpedo it was supposed to carry.

It is discouraging to equipment people, said E. M. Greer, of Simmonds Aerocessories, to find out what is done in the way of installing an item of equipment on which hard-won weight savings had been made. For example, an hydraulic accumulator was so installed in a certain plane as to require 65 lb more of tubing and liquid weight than was necessary if the installation had been made properly.

Mr. Hackney agreed that consulting on proper installation was one of the proper functions of the weight-control engineer.

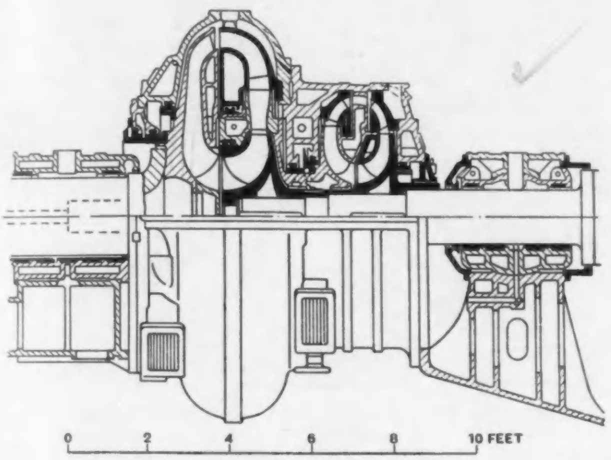
BANQUET AND GRAND BALL

After taking time out to attend the afternoon football game, the entire group of local and visiting engineers got together for an evening devoted purely to social activity at a banquet in the main ballroom of the Biltmore Hotel.

FLUID TRANSMISSION OF POWER

by N. L. ALISON

General Manager, Hydraulic Coupling Division, American Blower Corp.



■ Fig. 1 – Föttinger transformer, with ahead and astern circuits for marine propulsion – Capacity, 16,000 hp at 1000/180 rpm

AS the title suggests, this paper deals with the transmission of power by means of fluid couplings and torque converters of the hydrokinetic type. The descriptions and illustrations apply particularly to the Föttinger type of torque and speed transformer and a more recent descendant of this device, the Vulcan and Vulcan-Sinclair hydraulic couplings.

Various applications are touched on briefly, but particular emphasis is given to the use of the hydraulic coupling in combination with change-speed transmissions as applied to automotive and heavy-duty industrial applications. Due to the characteristics of the hydraulic coupling, it is applied most successfully with constant-mesh transmissions, either of the planetary or balked engagement type, and a number of such are mentioned.

The torque converter is shown in its simplest form, and the combination of torque converter and hydraulic coupling for traction service is described briefly.

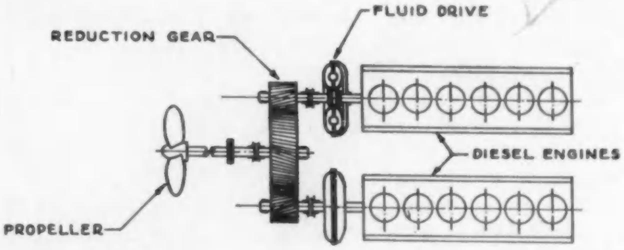
■ ■ ■

FLUID power transmission dates back to the "Föttinger speed transformer" which was invented by Dr. Föttinger in the early part of this century. This development was of the torque-converter type, having the conventional primary and secondary vaned members, and interposed between these, a stationary vaned reaction member to reverse the direction of the fluid, thereby making possible torque and speed transformation. Fig. 1 shows a typical forward and reverse transformer of this type. About this time the steam turbine was introduced for marine propulsion but, as the helical reduction gear had not yet come into use as a speed reducer for this class of service, the Föttinger transformer enjoyed considerable popularity for a time as a means of direct-connecting a high-speed driver to a low-speed propeller.

The Föttinger speed transformer gave a speed reduction on the output shaft of about 5:1, and a torque conversion, with the secondary member stalled, of about the same amount; but, due to the hydraulic losses inherent in such a device, the efficiency did not exceed about 85% in practical operation. It was possible, however, to regain a portion of this 15% power loss by recirculating the boiler feed water through the transformer and thus increase the overall efficiency to somewhere near 90%.

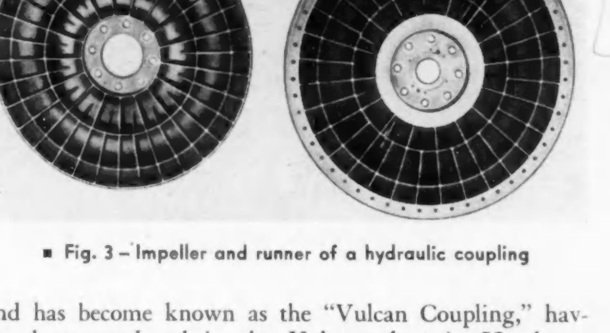
■ Marine Couplings

The advent of the high-speed geared diesel engine for marine propulsion made necessary some sort of vibration damper which could be introduced between the engine crankshaft and the pinion gear to protect the gears from the severe torsionals inherent in the early designs of diesel engines. This, however, was not a question of speed or torque conversion and, as a result, the stationary reaction member of the Föttinger transformer was eliminated and a simple coupling, having a 1:1 torque ratio resulted. Such a coupling was developed by Dr. Bauer,



■ Fig. 2 – Two medium-speed diesels connected through hydraulic couplings and helical reduction gears to a propeller shaft

[This paper was presented at the National Tractor Meeting of the Society, Milwaukee, Wis., Sept. 24, 1940.]



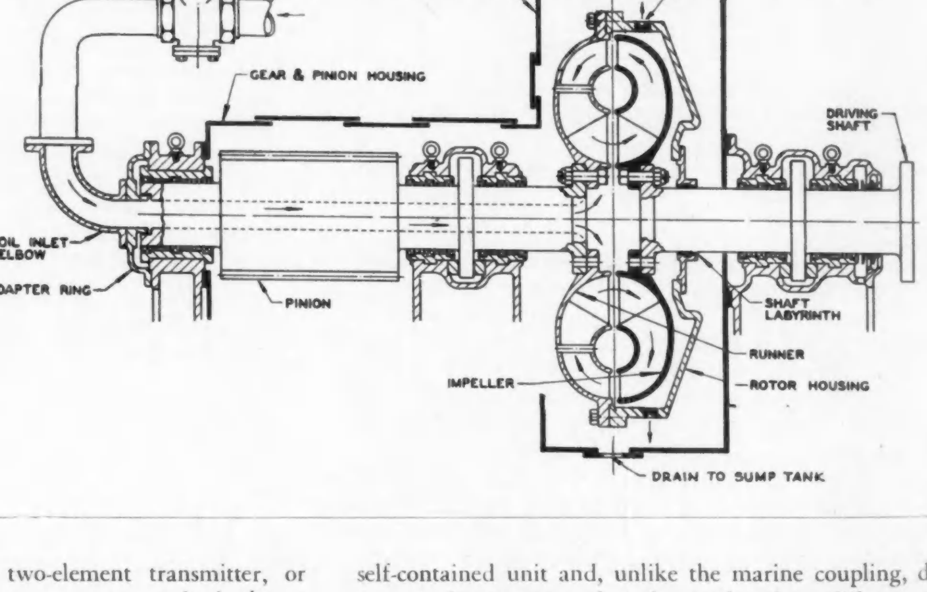
■ Fig. 3 – Impeller and runner of a hydraulic coupling

and has become known as the "Vulcan Coupling," having been produced in the Vulcan plant in Hamburg. Since that time, more than 2,000,000 hp of them have been used on commercial and naval vessels.

coupling serves the further purpose of preventing the operating characteristics of one engine from interfering with the others, and also permits easy clutching and declutching, while under way, by simply filling or emptying the couplings.

■ Variable-Speed Couplings

By partially filling the marine coupling, it was found that wide-range stepless speed regulation of the propeller shaft could be secured with the engine running at constant speed, and this led to the development of the Type VS, or "Scoop-Tube" variable-speed fluid coupling shown in Fig. 6. This coupling has been used extensively for the variable-speed drive of centrifugal fans, pumps, turbo-blowers, and so on, in units up to 2000 hp in capacity. It has been applied principally to electric motor drives, but has also been used in connection with certain diesel engine applications. The Type VS coupling is a

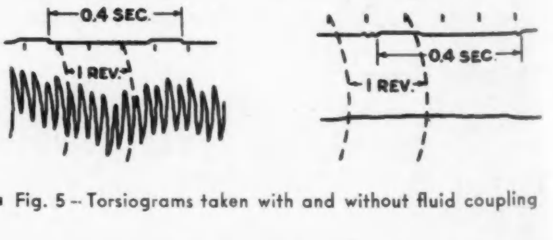


■ Fig. 4 – Hydraulic coupling and pinion gear of a typical marine installation

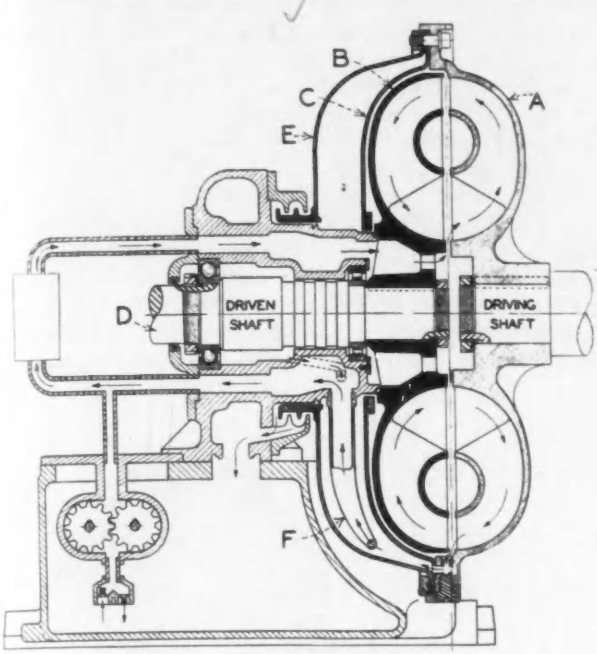
A typical layout of the two-element transmitter, or Vulcan fluid coupling, having a 1:1 torque ratio, is shown in Fig. 2, while Fig. 3 shows the construction of the rotating members of the coupling. A cross-section of the fluid coupling and pinion gear is shown in Fig. 4, from which it will be noted that oil is fed into the coupling through the hollow pinion shaft to keep the coupling filled, and that there is no mechanical connection between the driving and driven members, the power of the engine being transmitted to the gear by the kinetic energy of the driving fluid, which circulates between the radial passages formed in the two rotating elements. A rotor housing enclosing the back of the impeller to retain the working fluid, is bolted to, and rotates with the driven member – or runner.

Because of its ability to prevent the transmission of torsional vibrations, as shown in Fig. 5, thus completely divorcing the vibrating system of the engine from that of the gears, shafting, and propeller, the fluid coupling has been an important factor in the development of multiple-engine marine drive. In this application, the

self-contained unit and, unlike the marine coupling, does not require an external stationary housing. Oil removed from the coupling circuit is stored in the tank shown and speed regulation is obtained by operating the motor-driven pump in one direction or the other, as it is desired,



■ Fig. 5 – Torsiograms taken with and without fluid coupling



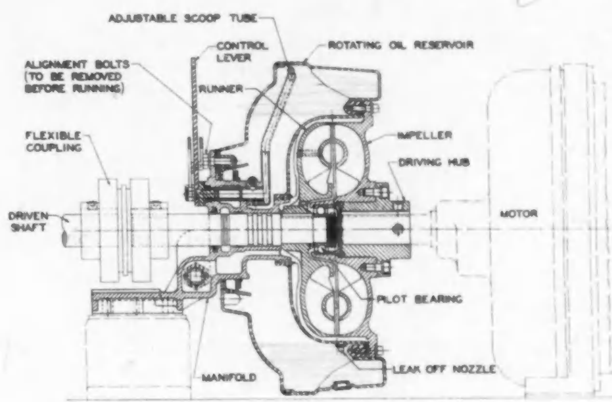
■ Fig. 6 – "Scoop-tube" variable-speed hydraulic coupling

to increase or decrease the output speed. The pump can be controlled from a remote point by means of pushbuttons, or various types of automatic control can be applied. The pump only operates when a change of speed is called for and at all other times remains idle, circulation through the oil cooler being maintained by the action of the scoop-tube.

■ Scoop-Control Coupling

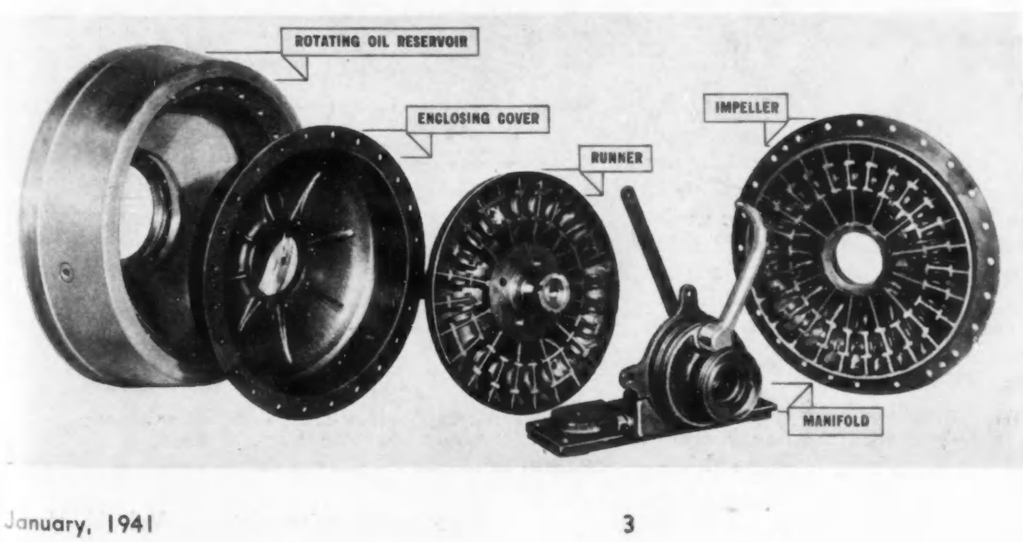
A more recent development of the variable-speed coupling is the scoop control type shown in Fig. 7. The component parts of the coupling are shown in Fig. 8, and it will be seen that, in addition to the impeller, runner, and inner casing of the Type VS coupling, the scoop control coupling has a rotating reservoir of sufficient capacity to hold the content of the working circuit. It also will be seen that an adjustable scoop tube, extending into the reservoir and controlled by an external handle, is mounted on the coupling manifold. Oil passes from the working circuit through nozzles provided in the inner casing and collects in the rotating reservoir, from where it is picked up by the scoop tube and returned to the coupling circuit. The scoop tube is mounted off center so that, in its fully extended position, it handles all of the oil in the rotating reservoir and maximum output speed is obtained. By moving the control handle through an arc of about 60 deg, the scoop is brought into a fully retracted position, so that all of the oil drains into the reservoir, and the coupling is completely disconnected. By placing the scoop control lever in any intermediate position, varying quantities of oil are delivered to the working circuit and any desired output speed can be obtained.

When used with diesel engines driving oil-well drilling machinery such as slush pumps, rotary tables, and draw works, the scoop control coupling supplies a convenient means of declutching; provides variable speed for "fishing" operations; and, when equipped with an oil cooler, permits the slush pumps to remain stalled for extended periods, maintaining pressure on the hole, with the engine continuing to run at reduced speed.

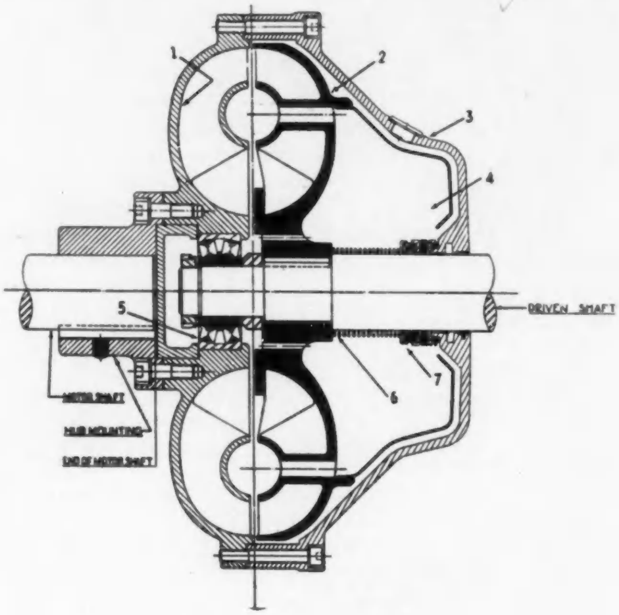


■ Fig. 7 – Section through scoop control coupling

The characteristics of the scoop control coupling with



■ Fig. 8 – Component parts of a scoop control coupling

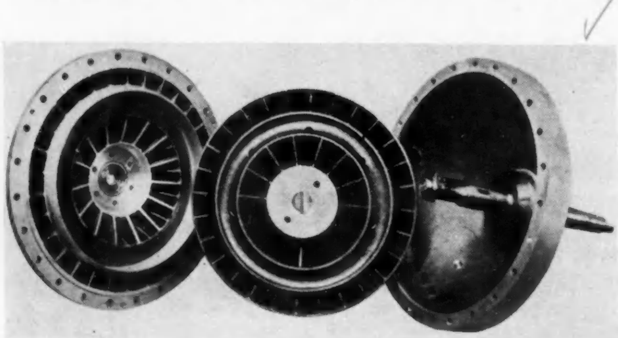


■ Fig. 9 – Section through a traction coupling

respect to slip, and so on, are the same as the traction coupling to be described in the following paragraphs.

■ Traction Coupling

The simplest form of the fluid coupling, and the most widely used, is what is commonly known as the traction-



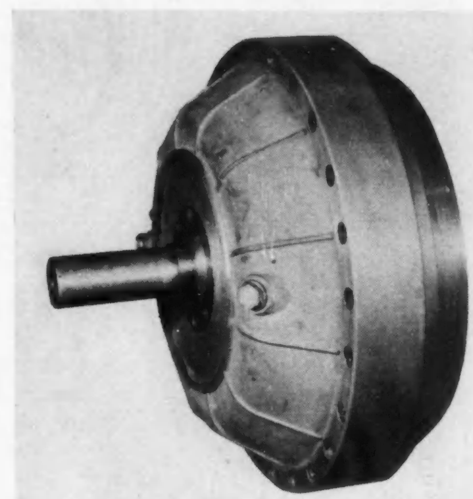
■ Fig. 10 – Disassembled view of traction coupling

type coupling. From Fig. 9, which is a section along the shaft, it will be noted that this is a self-contained unit, with a filling plug in the casing at the proper level for the correct filling, and a seal between the driven shaft and the casing to prevent the escape of oil from the working circuit. The only metallic contact between the driving and driven members is this seal and the internal thrust bearing, the outer race of which is mounted in the center of the impeller, and the inner race attached to the runner shaft to keep the two members spaced in proper relation with each other.

The traction coupling differs from the marine and variable-speed types in that, in it, no provision is made for completely disconnecting the driving from the driven members. It can be bolted directly onto the flywheel of the engine, or by means of a hub can be mounted on a shaft as shown in Fig. 9. It consists of the four principal parts shown in Fig. 10, namely: impeller, runner, enclosing cover, and driven side stub shaft. Fig. 11 shows the complete assembly of a traction coupling ready for either hub or flywheel mounting.

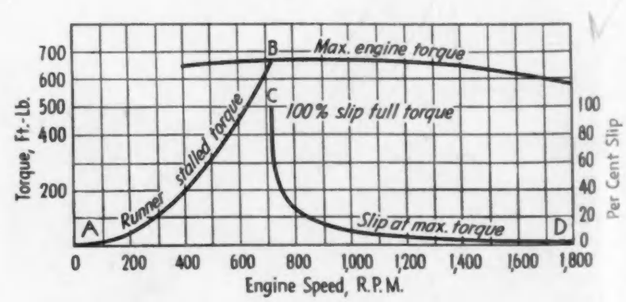
There is a certain amount of axial thrust set up by the fluid circuit in a hydraulic coupling, and this thrust is taken care of by the pilot bearing between the runner shaft and the impeller. This coupling operates with a fixed quantity of oil and does not require an external tank or pump. Heat generated within the coupling, amounting to about 3% of the power transmitted at

■ Fig. 11 – Traction coupling assembly

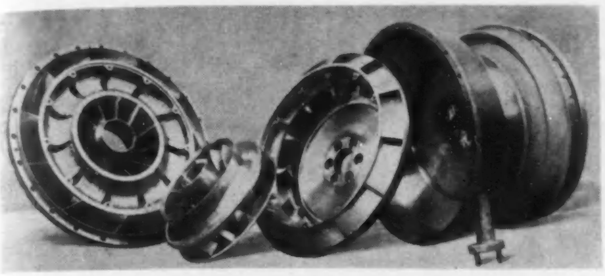


normal speed, is dissipated by radiation. When starting up, the coupling is filled with light lubricating oil to the level of the filling plug in the enclosing cover and, barring leaks, no further filling is required.

The slip and torque characteristics of the traction coupling are shown in Fig. 12. Curve *C-D* shows the slip of the coupling from 100% to 40% engine speed when delivering a maximum engine torque to the driven shaft; it will be seen that this slip increases from about 2½% at point *D* to 100% at point *C*. This represents a full-throttle condition. Curve *A-B* indicates the torque output from the coupling with the driven shaft stalled and the engine operating at reduced speed. For example, in starting up, the drag torque at point *A* is zero and, as



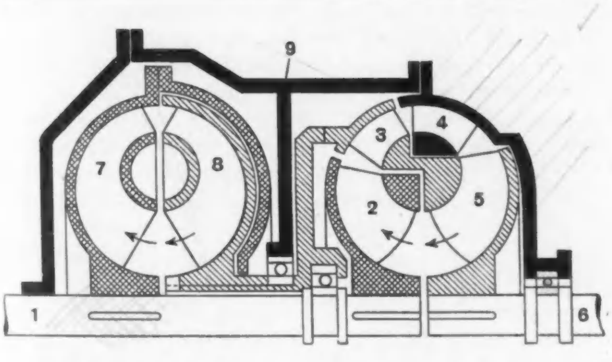
■ Fig. 12 – Traction coupling performance curve



■ Fig. 13 – Parts of Voith turbo transmission – Impeller and runner are mounted in a common casing to which the guide wheel is connected

the engine increases in speed, the torque of the coupling immediately builds up toward point *B*, setting the driven shaft in motion, after which the slip of the coupling rapidly decreases and falls to about 3% in the range of normal speed and load.

The speed at which the fluid travels around the vortex of the working circuit is influenced by the relative speeds of the driving and driven elements; hence, with the runner stalled and the coupling turning at engine speed, the working fluid possesses a maximum of kinetic energy. This results in a very low stalling speed and a correspondingly high stalled torque exerted by the coupling. To



■ Fig. 14 – Diagrammatic section through a Voith turbo transmission

1. Input shaft
2. Turbo converter impeller (primary)
3. First stage
4. Stationary guide vanes
5. Second stage
6. Output shaft
7. Turbo coupling impeller
8. Turbo coupling runner
9. Fixed casing

alter this, a ring-shaped baffle (8 in Fig. 9) is attached to the inner profile diameter of the driven member, which acts as a dam across the path of the oil to impede circulation when working at high slips. This baffle has no effect on the slip at normal speed and load as the runner speed throws the oil out beyond the baffle and, by varying its outside diameter, the point of coupling stall when delivering full torque can be changed best to suit the characteristics of the engine.

The reservoir on the back of the runner serves as an expansion chamber; consequently, there is no possibility of building up excessive pressure due to overheating of the oil in case the coupling is allowed to remain stalled for long periods with the engine developing full torque.

■ Hydraulic Torque Converters

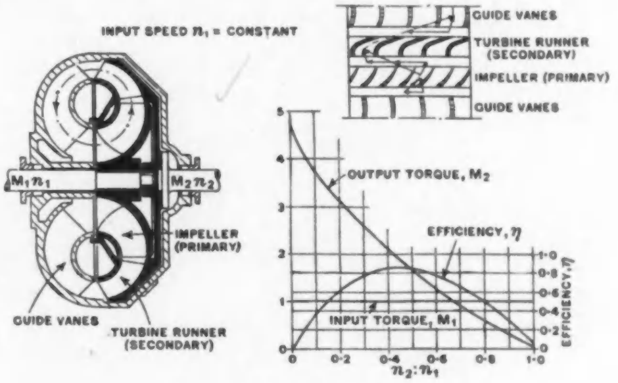
Hydrauic torque converters have been used extensively for diesel railcar, shunting locomotive, and bus drive, and a discussion of the "Fluid Transmission of Power" would be incomplete without a description of the converter.

Fig. 13 shows the component parts of a Voith torque converter, designed to transmit 200 hp at an engine speed of 1500 rpm, while Fig. 14 is a section through the working circuit. It will be seen that the converter consists of an impeller connected to the driving shaft, a runner connected to the driven shaft, a set of stationary guide vanes, and an enclosing casing. Torque and efficiency curves are also shown in Fig. 15.

Maximum efficiency of about 85% is obtained with the output shaft rotating at approximately 50% of input shaft speed and, at this point, the torque output is in the neighborhood of 175% of input torque. Maximum torque output is about 475% with the driven shaft stalled and the input shaft operating at approximately full speed. With the driven shaft running at 20% of engine speed, the output torque is about 320% and the efficiency 62%.

To secure the high starting torque of the converter for accelerating rail vehicles, and at the same time to benefit by the high efficiency of the fluid coupling for normal operation, a combination of fluid converter and coupling has been developed and a section through a unit of this type is shown in Fig. 15. The action of this unit is entirely automatic and in starting up it is simply necessary to open the engine throttle, the change-over from converter to coupling and coupling to converter taking place at about 60% vehicle speed through the action of a governor connected to the transmission output shaft.

The Voith turbo transmission also is used often with two hydraulic couplings following the torque converter, arranged to give a further step-up in torque. The second coupling is connected through a gear of about 1.5:1 gear ratio to the driven shaft, making an overall torque ratio under stalled condition of approximately 7:1. This arrangement further increases the overall efficiency of the transmission since the converter is not in use so much of the time. Fig. 16 shows torque, efficiency, and an input



■ Fig. 15 – Flow in Föttinger torque converter

speed plotted against vehicle speed, using a converter and two couplings. Such an arrangement can be used to advantage on switching locomotives, where high torque conversion is desirable.

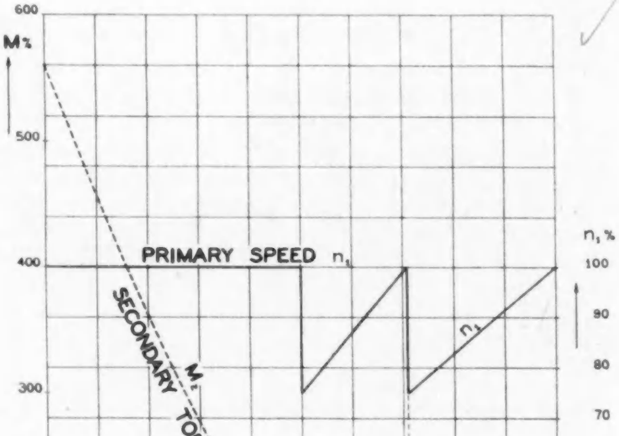
One definite advantage of the converter-coupling combination is that, in shifting from the converter speed to the coupling speed, full engine torque is maintained on the driving axle, and no slowing up of the vehicle is observed.

■ Applications of Fluid Couplings

To obtain the benefits of "fluid drive" in trucks, buses, passenger cars, tractors, and rail vehicles, fluid couplings are used in combination with a variety of mechanical transmissions, a few of which are listed below:

(1) Wilson Epicyclic Transmission.
(2) Sinclair Traction Transmission.
(3) Sinclair Type SSS Transmission.
(4) Fuller Transmission.
(5) Spicer Transmission.
(6) Chrysler Transmission.
(7) Oldsmobile Transmission.
(8) Banker Transmission.

Most of the transmissions listed are of the constant-mesh type utilizing either the planetary principle or the lay-shaft design with balking rings to facilitate engagement of dog clutches. When the conventional sliding-gear type of transmission is used, such as the Fuller gear box shown in Fig. 17, a friction clutch is located between



■ Fig. 16 – Curves of the characteristics of a three-stage Voith turbo transmission

the fluid coupling and the transmission to permit complete disengagement during shifting. With all types of transmissions, the fluid coupling has the effect of reducing the amount of gear shifting required and permits the vehicle to be started in a higher gear than is possible with the same transmission not equipped with fluid coupling. This is due to the smoothness with which the load is applied, and also to the fact that the engine operates at all times at or near its maximum torque, regardless of vehicle speed. A further decided advantage is gained from the slip of the coupling, allowing the engine to come immediately up to full speed and exerting the momentum of the flywheel and coupling to accelerate the vehicle. This characteristic of the fluid coupling is shown in the traction coupling performance curve.

It is not possible in this paper to include a description of each of the transmissions listed previously, but most of them have been covered in some detail in papers and magazine articles, and a reference list is given at the end of this paper.

The most recent development is the new Chrysler transmission, which employs about a 3½:1 axle ratio in high gear with about a 1½:1 ratio between high and intermediate. The shift is made going up by a release of the throttle at about 15 mph, and a governor drops back to intermediate at about 12 mph. A kickdown also is provided to shift from high into intermediate at any time that conditions make this desirable, such as when passing another car. Manual shift is provided to go into reverse and also into low gear in an emergency. With fluid drive, however, low gear is required only when starting on a grade or to pull out of a ditch.

The combination of fluid coupling and Fuller sliding-gear transmission shown in Fig. 17 has been used successfully in a number of heavy-duty ore trucks of a gross load capacity of about 65,000 lb. These units are operating in the Minnesota iron mines, and all reports indicate that they are giving a good account of themselves. The first of these trucks was built with an eight-speed transmission, but operating experience demonstrated that a four-speed box was sufficient for starting this load even under the most severe condition. All later trucks were equipped with a four-speed box and, at that, the low gear is seldom required. A heavy ore truck equipped with fluid coupling is shown in Fig. 18.

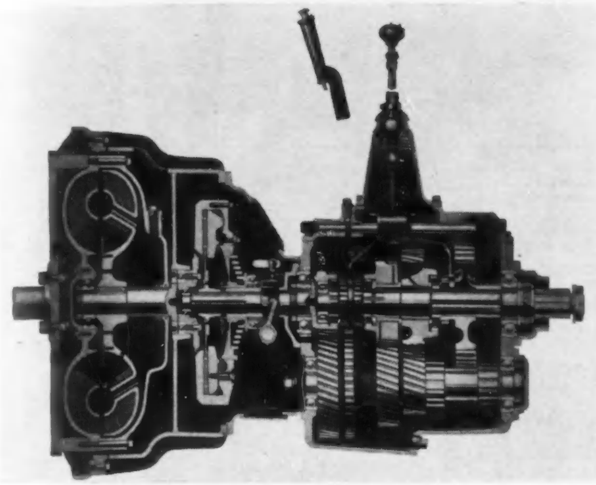
The Spicer Fluidgear Transmission is designed to operate behind a hydraulic coupling without the need of a friction clutch in the line. Conventional helical gears are used in the transmission, the shifting being done by jaw clutches. The jaw clutches are prevented from engaging by means of balking rings until the speeds of the two members to be engaged are synchronized and cross each other.

The shift control consists of an air cylinder head which does the actual shifting, the operator moving a small air lever to select the desired speed. By closing the throttle the torque on the jaws is relieved, the air head disengages the jaws and attempts to engage the ratio the driver has selected. Actual engagement is prevented by the balking rings until the speeds of the two parts to be engaged are equalized and cross, when the balking rings unlock and engagement takes place.

Before stopping the vehicle, first gear is selected and, as the vehicle comes to rest, the speeds cross and first will be engaged. If the vehicle is stopped with gears in neutral, then speeds will not cross unless the engine is stopped

and rotated backwards enough to unlock. To engage under this condition a rocking brake is provided which stops the driven member of the coupling and rotates it backwards through an arc sufficient to unlock the balk.

Thus, the unit gives the smooth start obtained by use of a coupling; the drive is cushioned hydraulically at all



■ Fig. 17 – Fuller transmission with fluid coupling

times; the operator is relieved of the effort required for declutching and shifting gears; and no friction clutch is required in the drive line.

As an indication of the extent to which fluid drive has advanced abroad, it is interesting to note that approximately 5000 buses with fluid drives were in service more than a year ago in London alone. These installations were divided between torque converters, and hydraulic couplings equipped with Wilson gearboxes.

While it has been impossible to learn definitely what has been used in Army transports in Europe during the last year, indications are that fluid power transmission has played an important part in both the German and English armies.

■ Oil-Well Drilling Equipment

For oil-well drilling, the fluid coupling increases the usefulness of the diesel engine by giving it the smoothness and flexibility of the steam engine which it is rapidly replacing in many parts of the world. For this service either the "scoop-control" or "traction" coupling can be used, the type selected being influenced by the horsepower involved and the nature of the drilling.

Either type of coupling protects the machinery from shocks and vibrations, facilitates compounding of engines, prevents engine stalling, and provides a completely smooth pickup of the load. The "scoop-control" coupling offers the further advantage of wide-range speed regulation and acts as a disconnecting clutch, thus making it unnecessary to equip the engines with friction clutches in the case of multiple engine installations.

Speed regulation obtained with the scoop control coupling is especially useful for rotary-table drive during "fishing" operations, and where lower drilling speeds are desired than can be obtained by regulating the speed of the engine alone. Speed regulation also can be used to advantage in deep drilling for slush pump drive, where it is desired to operate the pump at speeds below the stable operating range of the engine. Fig. 19 shows a scoop-control coupling, diesel engine, and V-belt sheave assembly.

One important advantage of the steam engine over the direct-connected diesel for slush-pump drive is that it permits the pump to stall completely while continuing to maintain pressure on the hole in case of a plugged bit or when testing casing. However, if the diesel is connected to the pump through a fluid coupling, the performance of the steam pump can be duplicated, inasmuch as the pump can remain stalled with the diesel continuing to run at reduced speed and delivering full torque to the pump shaft. At such times the coupling, of course, operates with 100% slip, and an oil cooler is required if the pump remains stalled for extended periods. A hydraulic torque converter is not suitable for this application because of the damaging pressure that would be developed by the stalled pump due to the high torque output of the converter.

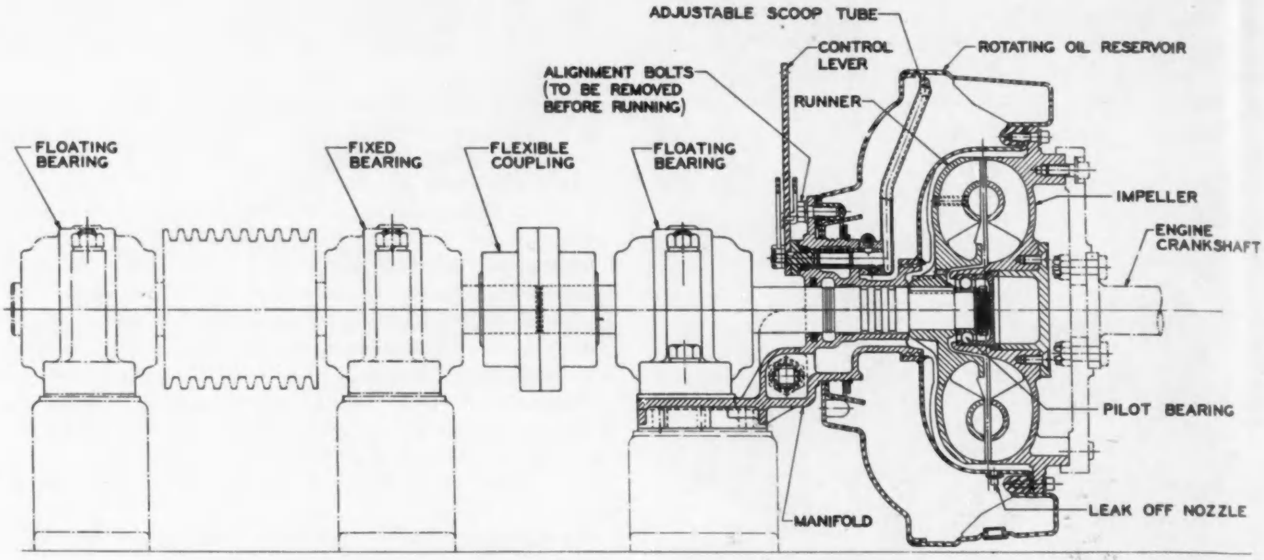
As suggested above under the heading "Applications of Fluid Couplings," the fluid coupling permits starting a load with a diesel engine in a higher gear than is possible with straight mechanical drive, and this advantage is of particular importance in the case of draw works installations from the standpoint of speed of hoisting. For example, if a 2:1 gear ratio can be used instead of a 3:1 ratio, the speed of hoisting is increased by 50%.

Many drill rig operators utilize the stored energy in the engine and flywheel for breaking loose stuck drill pipe by operating the engine at top speed and suddenly engaging a friction clutch connected to the hoisting drum.

■ Fig. 18 – Dart truck equipped with a fluid drive

This method puts excessive strains on the machinery and causes high clutch maintenance, but it produces results. By locating a fluid coupling between the engine and the friction clutch, similar results can be secured without maintenance and without shock to the equipment due to the initial cushioning effect of the coupling which allows the "slip" to be taken in the fluid coupling instead of in the clutch.

Fig. 20 shows the torque-slip curve of a fluid coupling based on the driving member or impeller operating at constant speed, and the driven member or runner operating with varying amounts of slip up to 100%. From this curve it will be seen that approximately six times the normal running torque corresponding to 3% slip is



■ Fig. 19 – Scoop control coupling mounted on diesel engine with V-belt power take-off

required to stall the driven shaft, assuming that the driving shaft continues to run at constant speed. Now, if we take the example of the draw works drive in the preceding paragraph with the engine operating up to speed and with the friction clutch disengaged, the slip of the fluid coupling is zero, and the output torque is zero. If, then, the friction clutch is suddenly engaged, the slip of the fluid coupling immediately increases to 100% and the stored energy in the engine and flywheel, and hydraulic coupling impeller amounting to several times normal engine torque, is instantly delivered to the output shaft of the coupling, and is available to start the driven load in motion. This means that the driller can operate the draw works in the gear that will provide the maximum hoisting speed so as to utilize the full power of the engine with an average transmission efficiency in the neighborhood of 95%.

The torque-slip curve in Fig. 20 shows that the stalled torque is approximately six times the normal running torque at 3% slip. It should be explained, however, that this figure is subject to a wide variation depending upon the design, and this value can be as low as approximately three times or as high as eight times the normal torque.

■ Other Applications

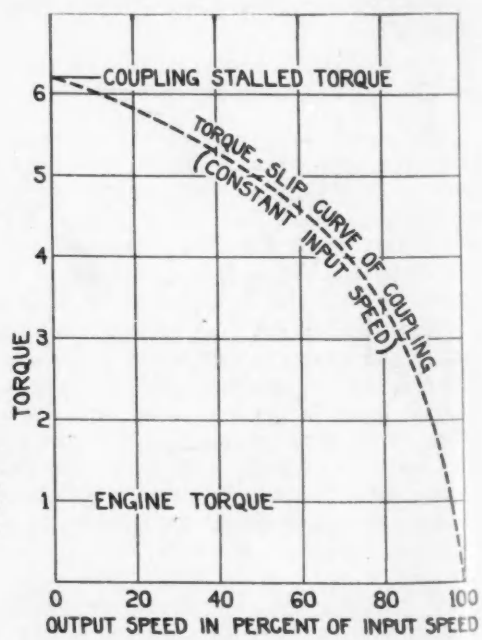
The "traction coupling" also can be used to advantage for connecting diesel engines to air and refrigerating compressors, especially where automatic control is provided for starting and stopping the engines. Here the low drag of the coupling permits an electric starter to start the engine at reduced load, and a smooth acceleration of the compressor is provided. The coupling also prevents the transmission of shocks and vibrations from the engine to the compressor and from the compressor to the engine, as well as providing a certain degree of flexibility, which is of particular value in the case of portable units.

For use in power shovels, drag lines and ditchers, the traction coupling used as a "power take-off" protects the engine as well as the gears, chains, and ropes from shock loads. The fluid coupling in this application also prevents engine stalling in the event the operator takes too big a "bite," or if the bucket runs into some obstruction such as a heavy boulder or rock. The effect of the overload is to pull down the engine speed, but in no case is it possible to stall the engine. A power shovel equipped with diesel engine and fluid coupling is shown in Fig. 21.

■ Summary

Internal-combustion engine applications of the fluid coupling can be placed in three general divisions: (1) marine, (2) automotive, including rail traction, and (3),

■ Fig. 20 – Torque - slip curve of fluid coupling, driving member operating at constant speed



industrial. Up to January, 1940, marine installations amounted to over 2,000,000 hp, while automotive and industrial drives accounted for more than 1,000,000 hp. In addition to the applications described, fluid couplings are used extensively in connection with constant-speed electric motors for variable speed, to limit output torque, and to provide a smooth take-up of the load.

Theory of Fluid Drives

It is not possible within the short space of this paper to discuss the theory of torque converters and hydraulic couplings to any extent, but anyone interested in really going into the subject will find interesting data on it in the Proceedings of the Institution of Mechanical Engineers for 1935. There are two articles on torque converters, one by Messrs. Haworth and Lysholm of Leyland Motors Ltd., England, on both the fixed and adjustable-vane types, and another by Dr. Hahn of the J. M. Voith Co. on the fixed-vane converter and the combination converter and coupling, comprising the Voith turbo transmission.

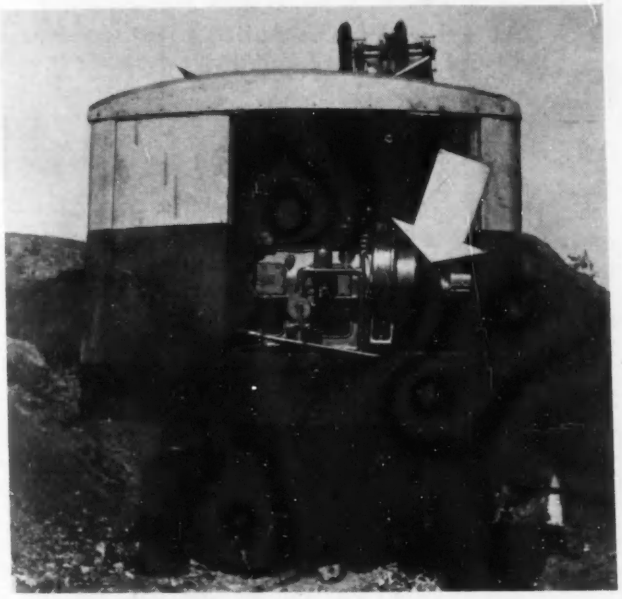


Fig. 21 – Rear view of Marion power shovel

The torque-converter design is considerably more involved than that of the hydraulic coupling, as blade angles and curvature play such an important part in hydraulic torque conversion while, in the hydraulic coupling, the blades are all perfectly flat, and are set radially with the blade surface parallel to the shaft. No attempt will be made to discuss converter design other than to state that being a hydro-kinetic type of power transmitter, it follows hydraulic laws governing both centrifugal pumps and hydraulic turbines.

The hydraulic coupling also follows these same laws, but its problem is simplified in that the blades are radial, their surfaces flat, and the diameters of the driving and driven members are the same, which is to say – no attempt is made to convert torque. The output torque is always equal to the input torque and, for any given slip, it varies directly as the square of the speed or $(rpm)^2$ .... (1)

As a consequence of the torque varying as the square of the speed, it follows that, at a given slip, the hp varies as the $(rpm)^3$ .... (2).

Assuming the revolution (rpm) and the slip S to be constant, the velocities in the working circuit are proportional to the diameter D. Hence the kinetic energy given up by unit mass in the fluid passing through the driven member is proportional to D^2. Also the mass of the fluid varies with the volume of the working circuit, which in turn varies with D^3. Therefore, the horsepower transmitted at a given rpm and slip is proportional to

$$D^2 \times D^3 = D^5 \qquad (3).$$

Summing up the foregoing, it is possible to write the equation of horsepower transmitted as:

$$HP = (rpm)^3 \times D^5 \times C,$$

in which C takes into consideration the slip of the coupling, the fluid used, and the internal design of the working circuit. This factor can be determined only by test after the design of the working circuit is established. It will be noted from Fig. 20 that the torque-slip curve approaches a straight line between 94% and 100% input speed. Therefore, in that range, the power transmitted is directly proportional to the slip; hence C will vary as the slip.

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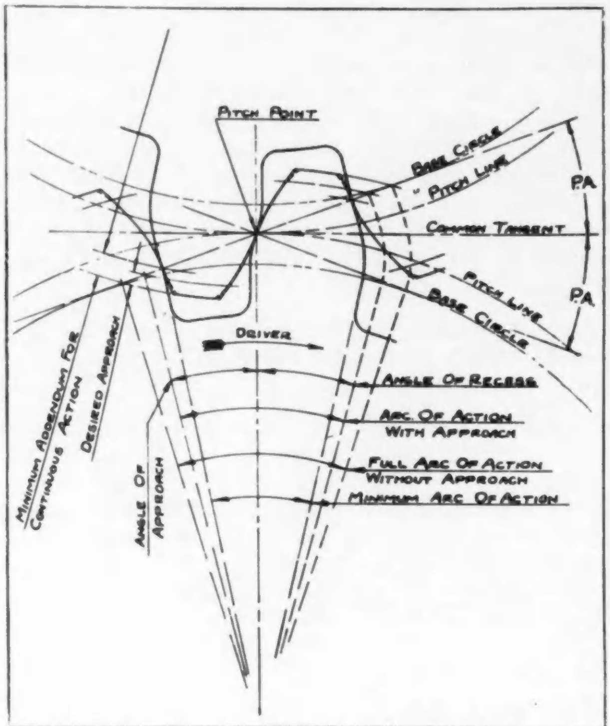
Correct Weight for Willys 1941 "Americar"

On page 445 of the paper "1941 Car Design Trends," published in the November, 1940, issue of the SAE Journal, the shipping weight of the four-door sedan of the 1941 Willys Americar was given incorrectly as "2370 lb, 115 lb more than last year's Willys 440." The correct dry weight of the 1941 Willys Americar is 2254 lb, only 20 lb more than the correct dry weight of 2234 lb for the 1940 Willys.



■ Fig. 1 – Common terms in involute gear design

GROUND FORM

THIS paper covers some of the more important points in the manufacture and care of hobs. The differences in manufacture and application between machine-relieved and relief-ground hobs are pointed out clearly. Emphasis is placed on the relief-ground type, recommended for use for gears where silent running, most exact uniformity of rotation, or close fitting is required.

The author gives special attention to the elements that must be watched in order to maintain accu-

IN modern automobile and machine construction it has been found advantageous to pay close attention to the inspection of component parts, as in this way much time can be saved in the final assembly.

Gears, however, being more intricate than most parts, must be made and inspected with as much, or more, precision than other machines or automobile components.

There are a good many different ways by which inaccuracy can creep into gears and cause them to be noisy; for instance, steel not properly handled, improper heat-treatment, inaccurate machines, careless operators, and inaccurate generating tools.

Past experience has taught all of us that we cannot deviate very far from the fundamental law of gearing and still expect to get quiet gears.

In order that a pair of gears may transmit a constant velocity ratio, their tooth curves must be such that a normal to the common tangent of the teeth at the point of contact will always pass through the pitch point.

Fig. 1 shows a method of calculating the arc of contact that any two mating gears have; also, the length of contact that the involute curve has with its mating gear.

In order to produce gears as just described, one of the many necessary requirements is to have accurate hobs.

In following the hobbing process, it is of paramount importance that the lead of the hob be made accurate. By accurate lead we not only mean that the thread of the hob must be advanced a given amount in one convolution, but that the lead must not deviate more than 0.0005 in. from the true helical path of the thread in making a convolution.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 17, 1940.]

Since "quiet gears" is the watchword of the transmission builder of today and since noisy gears are always found to be inaccurate, "how to manufacture accurate gears," is the problem that confronts the gear manufacturer of today, especially the transmission builder. The fact is that a gear having tooth contours which comply with the geometrical

DIAMETRAL PITCH			1 to 1¾	2 to 2¾	3	4	5	6 to 12	13 to 50
HOLE DIAMETER									
Must be lapped straight, parallel, not less than 75% bearing length.									
Hole within (plus only)		A	.0008	.0005	.0003	.0003	.0002	.0002	.0002
		B	.001	.0007	.0005	.0005	.0005	.0003	.00025
HUB RUN OUT									
Face within		A	.0008	.0005	.0003	.0002	.0002	.0002	.0002
		B	.001	.0007	.0005	.0003	.00025	.00025	.00025
Diameter within		A	.001	.0005	.0004	.0003	.0003	.0002	.0002
		B	.0012	.0008	.0005	.0003	.0003	.00025	.00025
O. D. Run-Out	Up to 3" Diameter	A						.001	.001
		B						.001	.001
	3" to 4" Inclusive	A				.0015	.001	.001	.001
		B				.0015	.0015	.001	.001
	Over 4" Diameter	A	.003	.002	.0015	.0015	.001	.001	
		B	.004	.003	.0015	.0015	.0015	.001	
Spacing from flute to flute within		A	.004	.003	.0025	.002	.0015	.001	.001
		B	.005	.005	.004	.003	.002	.001	.001
Accumulated error in one revolution must not exceed		A	.010	.009	.008	.007	.005	.003	.003
		B	.012	.010	.009	.008	.006	.005	.004
CUTTING FACES									
Must be radial within (Cutting Depth)		A	.003	.0015	.001	.0008	.0006	.0005	.0003
		B	.005	.0025	.0015	.001	.0008	.0007	.0005
THREAD									
Lead variation in any one complete turn must not exceed		A	.0025	.002	.001	.0008	.0006	.0005	.0004
		B	.0035	.0025	.0015	.0013	.0010	.0008	.0006
Lead variation from tooth to tooth must not exceed		A	.0007	.0005	.0004	.0003	.0002	.0002	.0002
		B	.001	.0008	.0007	.0006	.0005	.0004	.0003
TOOTH PROFILE OR PRESSURE ANGLE									
Must be symmetrical within		A	.001	.0007	.0005	.0004	.0003	.0002	.0002
		B	.0015	.001	.0006	.0005	.0004	.0003	.0003
Straight portion within		A	.001	.0005	.0003	.00025	.0002	.00015	.0002
		B	.0016	.0008	.0005	.0004	.0004	.0002	.0002
Start of approach within plus or minus		A	.020	.018	.015	.010	.010	.010	.005
		B	.022	.020	.015	.010	.010	.010	.005
Approach on each side of tooth symmetrical within		A	.015	.0120	.010	.0080	.006	.005	.005
		B	.015	.012	.010	.008	.006	.005	.005
Tooth thickness within (minus only)		A	.003	.002	.0015	.0015	.001	.001	.001
		B	.003	.002	.002	.002	.002	.001	.001
MULTIPLE THREADS									
Circular Pitch Within		A B	.005	.003	.0020	.0015	.0010	.0007	.0005

■ Fig. 2 – Limits for single and multiple thread ground finishing (spur and spiral) gear hobs – The classification of a hob is determined by the finest limit of any one feature

FINISHING HOBS

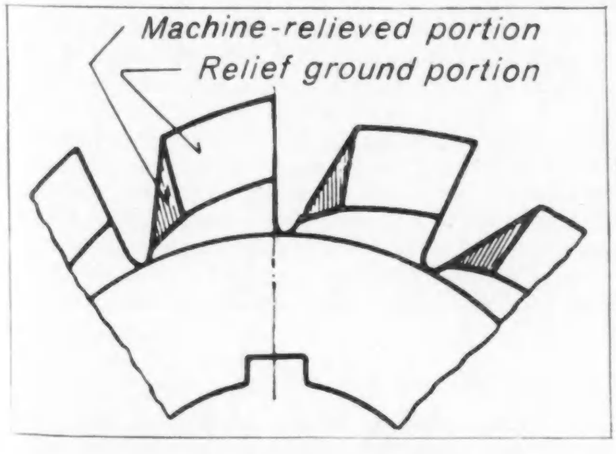
by CHARLES R. STAUB
Chief Engineer, Michigan Tool Co.

racy; the care that must be taken in sharpening hobs, as well as in mounting them on the hobbing machine for cutting gears; and the importance of accuracy in hobbing machines and in gear blanks, for quiet and efficient gears.

Among the various testing instruments described that indicate how closely the finished hob approaches the ideal conditions, are a lead-testing machine and a contour-testing fixture. A chart is presented for use in estimating gear-cutting time.

laws underlying its construction, will make a better gear than one not having teeth so formed. Considering the many factors that enter into the making of accurate gears, keeping the fundamental facts as mentioned in mind will help the manufacturer a great deal to eliminate gear trouble, provided, of course, that the system of gearing used is theoretically correct.

The constantly increasing demand as regards accuracy in gearing has given rise to the utmost accuracy in hob manufacture. The machine-relieved hobs formerly used exclusively will still suffice perfectly well for many purposes, but not in cases where silent running, most exact uniformity of rotation, or close fitting is stipulated. For such purposes, relief-ground hobs should be used. They differ from the machine-relieved type in that the flanks and tops of the teeth, often also the root, are ground after hardening. This procedure has the effect of removing all



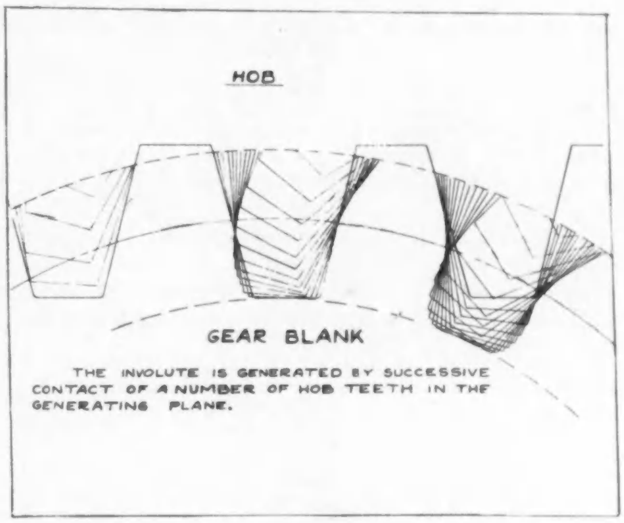
■ Fig. 3 – Hob with relief-ground teeth

inaccuracies occurring in hardening; the lead is improved and uniform tooth form is ensured for all teeth.

Fig. 2 shows the hob tolerances that are used by the hob manufacturers of today. This tabulation gives both Class "A" and Class "B" tolerances.

Roughing as well as finishing hobs may be ground on the teeth, although it is usual to limit form-grinding of hobs to a special quality of work where real accuracy is required.

It is not possible to lay down rules that specify where ground hobs must be used or where machine-relieved hobs will suffice. The deciding factor is the degree of accuracy needed by the producer of the gears or other components hobbed. Thus, for example, many spline-shaft hobs are form-ground, although most hobs for special profiles are



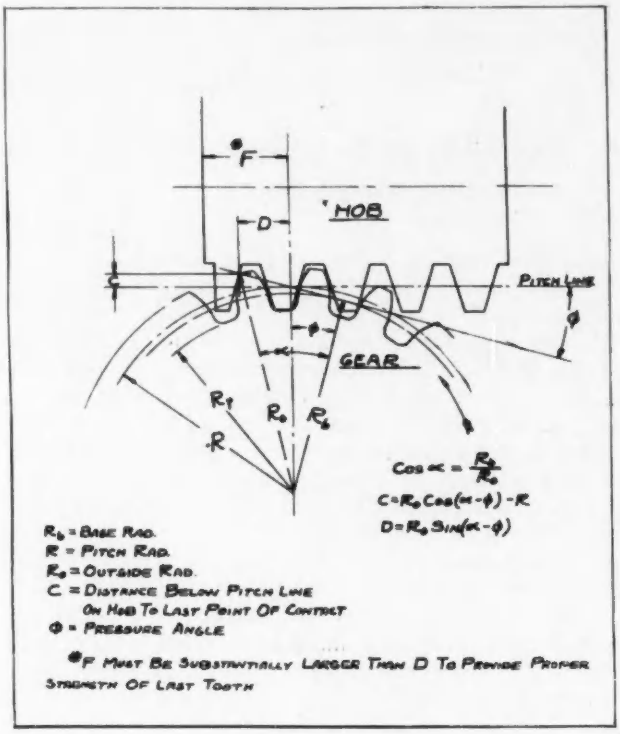
■ Fig. 4 – Gear tooth as generated by a hob

only machine-relieved; this condition exists despite the fact that the short, wide spline-hob teeth distort very little in hardening by comparison with other tooth forms.

Hob teeth that are to be ground are relieved at the back to clear the run-out of the grinding wheel, as shown in Fig. 3. By this type of relief the hob tooth is left just as strong as formerly without widening the flutes.

There have been developed a number of test fixtures to control the manufacturing process of hobs based on the fundamental laws of gearing. These fixtures also will be of unlimited value to the users of hobs to control the quality of tools coming into their plants. To design and use a hob intelligently, it is necessary to understand the geometrical laws involved.

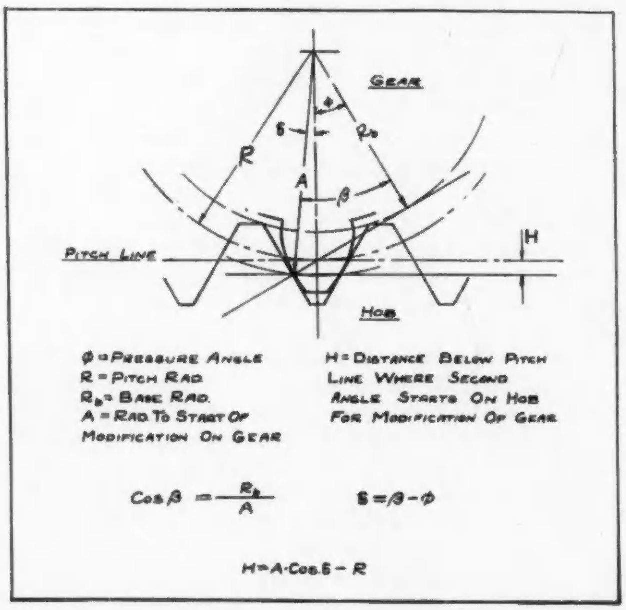
Fig. 4 shows the increments of cuts as a hob generates a gear tooth.



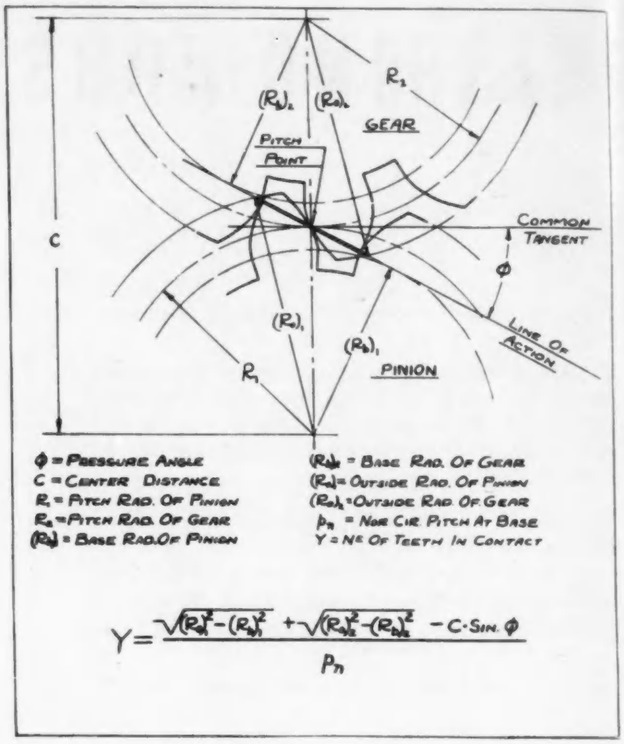
■ Fig. 5 – Method of calculating the last point of contact of a gear tooth with a hob tooth, and of finding the last cutting position for setting a hob

Fig. 5 shows a method of calculating the last point of contact that a gear tooth has with a hob tooth, also the last cutting position that a hob may be set and still obtain full generating action.

In designing the hob it is necessary to know what these contacts are, as the involute gear as it is made today, especially in the automobile field, has a modified tooth, and the hob has to be designed to form these modifications,



■ Fig. 6 – Method of finding the start of modification on a hob for a known modification of the gear



■ Fig. 7 – Method of finding the number of teeth in contact for external gears

or approach curves (as they are sometimes called) without breaking the arc of contact with the mating gears.

Fig. 6 is a formula showing how to find the start of modification on a hob for a known modification on the gear, which is predetermined by an analysis of the gear combination in question and by use of the formula shown in Fig. 7.

After making these calculations, a hob then must be made that will generate the involute and modify it correctly; therefore, it must have the correct contour on all of its cutting teeth. It must have all of its cutting teeth in the true helical path and all of its cutting edges concentric. In modifying gear teeth, care must be taken not to modify them excessively because any modification of the involute form reduces the amount of involute overlap, or the number of teeth in contact.

To check hobs for these fundamental requirements of correct contour of teeth, proper location of contour on the helical path of the thread, and uniform distance of contour from the axis of the hob, it is essential to have checking equipment that is designed for this purpose and is theoretically correct.

The testing instruments that have been developed will indicate how closely the finished hob approaches the ideal conditions.

Fig. 8 shows a hob on the lead-testing machine which uses a sine bar that can be set to check any lead from 0.144 to 2.6 in. by using the sine-bar method of checking. The possibility of lead screw errors showing up in the hob is eliminated.

■ Design Principle

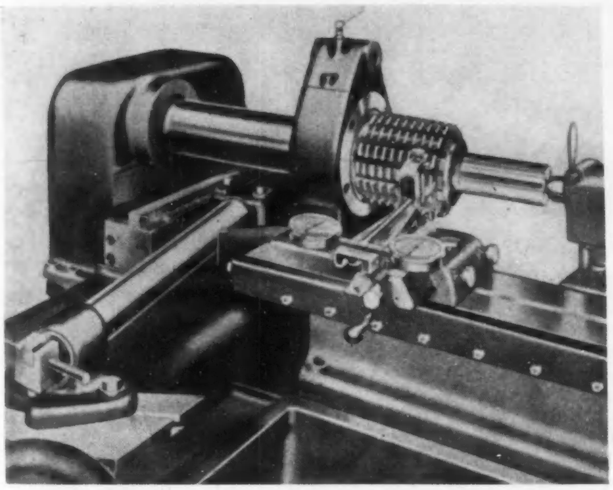
The main components of the machine are shown in Fig. 8. These are the hob spindle, the indicator slide, and the sine-bar table.

In order to check the lead of a given hob, the spindle must make one complete turn while the indicator slide travels a distance equal to the axial lead of the hob to be checked. This we obtain in the following manner:

The rotation of the spindle and the movement of the indicator slide are both controlled by movement of the sine-bar table. This table, clearly shown in Fig. 8, carries the sine bar and also a precision master rack. Driven by this master rack is a pinion which, through precision-ground change gears, causes the spindle to rotate, at the same time the sine bar actuates the movement of the indicator slide.

Only two simple operations are required to set this machine up for a given lead – the proper change gears for driving the spindle and the sine-bar setting which is determined by the axial lead of the hob.

The indicator arrangement on this machine also has a



■ Fig. 8 – Hob on lead-testing machine which employs the sine-bar method of checking

5:1 ratio, and will, therefore, register ten thousandths of an inch. The contact point of the indicator is brought against the cutting edge of the hob tooth, and the indicator dial is set at zero. As the table of the machine is advanced, the hob starts to rotate, and the slide carrying the indicator arrangement moves transversely away from the table. The amount of movement of the slide in one convolution of the hob is controlled by the setting of the sine bar, and is equal to the lead of the hob being checked. This setting is determined by the helix angle of the hob, and is positive.

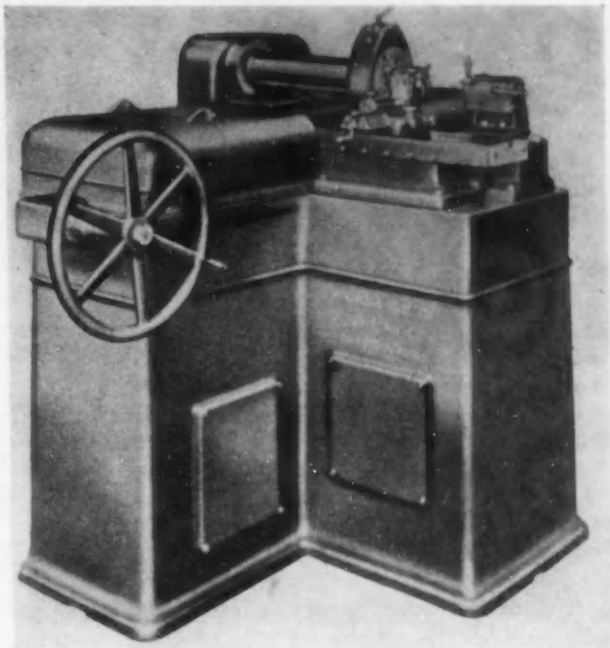
Fig. 9 gives a general view of the hob lead checker.

If the hob has been made correctly, the indicator point will register zero on all cutting edges. If there is a deviation of the hob teeth from the true helical path due to grinding, or if the hole in the hob is large and does not fit the arbor, or if the gashes in the hob are not properly spaced, the indicator will register a great variation in the cutting edges as they pass by the indicating points.

Fig. 10 is a lead chart showing how the lead of the hob can be plotted graphically so that it can be seen at a glance just where and how great the errors are.

■ Hob Contour

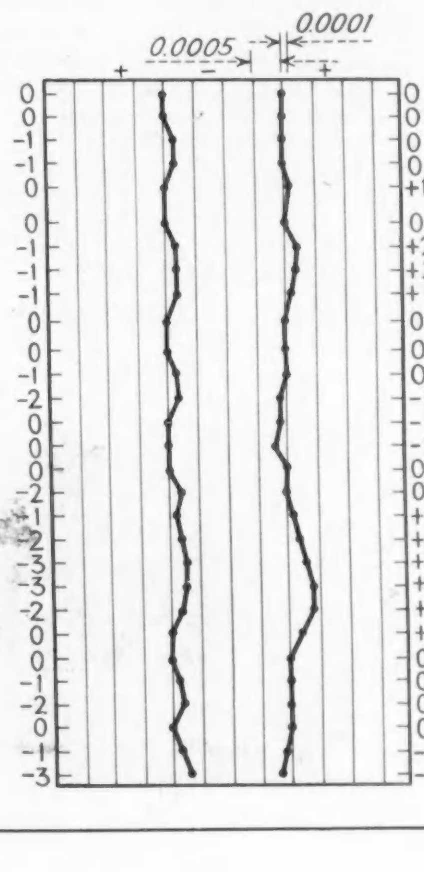
Next of importance is the angle of inclination, or the pressure angle of the hob, and its modification if any.



■ Fig. 9 – Hob lead-checking machine

The bearing produced on the gear to be cut is largely controlled by the contour of the hob, and may be varied for different applications.

Fig. 11 shows a hob on the contour-testing fixture; the indicator arrangement is constructed so that a movement



■ Fig. 10–Typical lead inspection chart of gear hob

of 0.001 in. in the contact point is registered as 0.005 in. on the dial indicator. Therefore, a movement of 0.0002 in. on the point of contact will register 0.001 in. on the dial indicator. By this amplification it is possible to check the contour of hob teeth to ten thousandths of an inch.

This is a very sensitive instrument and, having a keen-edged contact finger, it actually will register such depressions and elevations as are caused by rough grinding. It also will show any deviation from the true pressure angle by means of a sine bar setting.

The indicator slide can be moved inward in increments of 0.010 or 0.015 in., and readings can be taken at the different points and plotted on a graph chart, thus showing any variation in the contour of the hob tooth.

Fig. 12 is a contour chart showing a modified hob tooth form, the centerline of modification being the master form, the other two lines being tolerance lines beyond which the form must not go. However, if the hob should check to either extreme, the hob will still cut gears without interference.

Fig. 13 is a formula sheet which shows the method of figuring correction in pressure angles for both straight and spiral gash hobs for extreme accuracy.

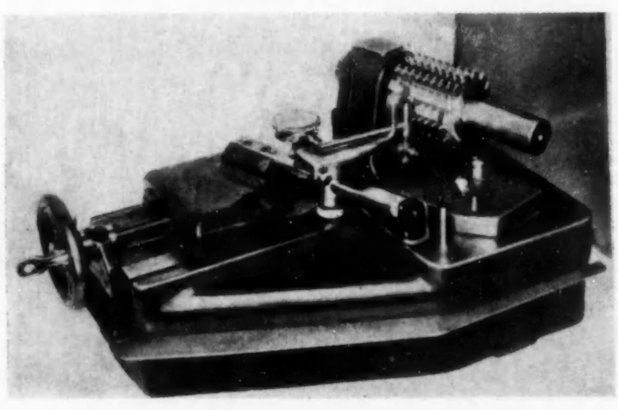
Sharpening of Hobs

The most important rule for the maintenance of hobs is *sharpen frequently*. It is always, and especially in the case of hobbing, more economical to resharpen more often, removing only a little from the face of the tooth, than the reverse as, after the cutting edges become blunt, deep pitted portions of the tooth flanks very soon are effected by the wear. The time apparently gained by running the hob longer is turned into loss by the more rapid wear and by the longer grinding time, consequently less tool life.

It is very important that a hob be sharpened properly, as a hob sharpener can spoil a good hob by not sharpening it concentrically. It is therefore necessary that the hob user provide himself with the means of checking hobs so that these errors can be detected.

We might mention here that, in sharpening hobs, every care should be taken not to try to remove too much metal at a time with the grinding wheel. If too much metal is removed at once, it will cause excessive heat and check marks in the hob teeth as shown in Fig. 14.

There are two very important reasons why hobs should be kept sharpened properly:

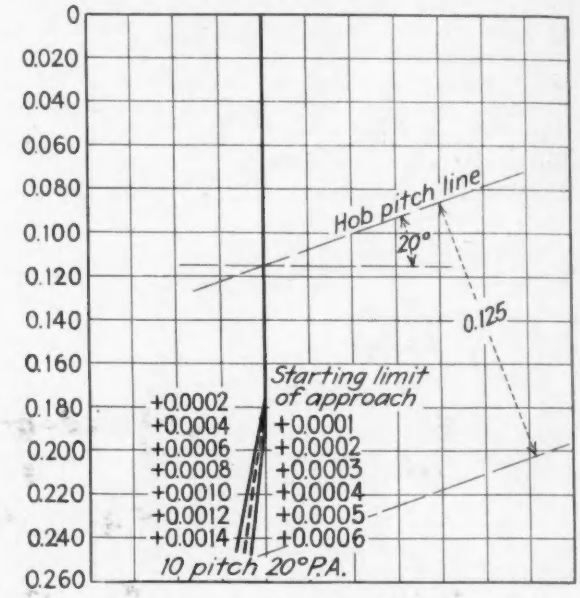


■ Fig. 11 – Hob on contour-testing fixture

(1) Economy: A sharp hob will cut much faster, use less power, and wear much longer.

(2) Better Work: A sharp hob will produce more accurate work with a finish far superior to that obtainable from a hob not in first-class condition.

A correctly sharpened hob tooth, *A*, has neither of the common errors shown at *B* and at *C* in Fig. 15. When a tooth is ground with a hook *B* it will cut a gear with a



■ Fig. 12 – Contour chart showing modified hob tooth form

decreased pressure angle. When ground ahead of center, as at *C*, the gear tooth will have an increased pressure angle. This condition is better demonstrated in the enlarged views shown in Fig. 16. Either of these errors will produce an incorrect form and should be avoided carefully.

Care must be used to maintain correct flute spacing. If some of the hob teeth are ground farther back than others, *D* in Fig. 15, causing a reduction in tooth height, *E*, only the remaining teeth will do all the work. This condition means faster wear and irregularity of cutting action, and the form of the gear teeth will not be the same on both sides.

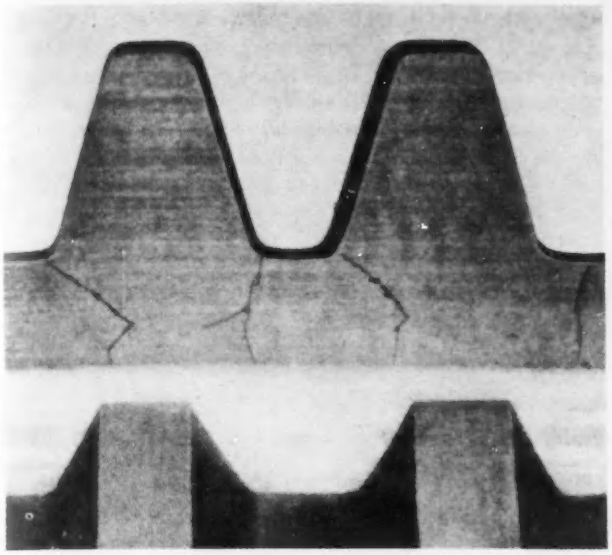
Hobs made with a hook (for special work) can be sharpened satisfactorily provided the amount of hook, marked on the hub, is maintained carefully.

For grinding, the beveled side of the grinding wheel should be used, with a medium grain of soft grade – yet hard enough to prevent flying grit. The wheel should be kept clean; a glazed wheel also may crack the hob teeth.

The flat side of a grinding wheel cuts away the face of the tooth as it touches the spiral surface far from the center point (in Fig. 17 the crossing point of work axis and wheel axis) and grinds there. The larger the spiral angle of the grinding flute, the larger must be the bevel angle of the grinding wheel in order to provide sufficient curvature away from the surface to be ground.

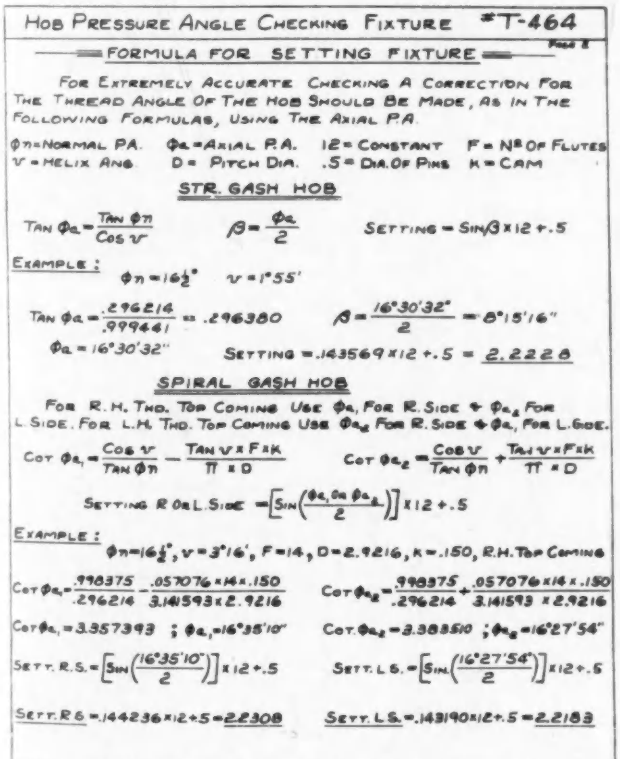
■ Hob Centering Gages

The function of hob centering gages is supposedly to make the gear teeth symmetrical or "balanced." Centralizing either a hob tooth or space with the common center-



▪ Fig. 14 – Check marks in hob teeth caused by removing too much metal at once in sharpening the hob

line between hob and blank will cause the series of flats comprising the tooth forms to be theoretically even on each side. In practice, these flats, caused by the successive hob teeth generating the involutes, are so close together that they merge into smooth curves. Usually they cannot be detected by the most sensitive profile measuring devices, and ordinarily cannot be seen when a gear is rolled with its mate to observe the bearing.



HOB PRESSURE ANGLE CHECKING FIXTURE #T-464

FORMULA FOR SETTING FIXTURE

FOR EXTREMELY ACCURATE CHECKING A CORRECTION FOR THE THREAD ANGLE OF THE HOB SHOULD BE MADE, AS IN THE FOLLOWING FORMULAS, USING THE AXIAL P.A.

ϕn = NORMAL P.A. ϕa = AXIAL P.A. 12 = CONSTANT F = Nº OF FLUTES
v = HELIX ANG. D = PITCH DIA. .5 = DIA. OF PINS K = CAM

STR. GASH HOB

$$\text{TAN } \phi a = \frac{\text{TAN } \phi n}{\text{COS } v} \qquad \beta = \frac{\phi a}{2} \qquad \text{SETTING} = \text{SIN } \beta \times 12 + .5$$

EXAMPLE:

$\phi n = 16\tfrac{1}{2}°$ $v = 1°55'$

$$\text{TAN } \phi a = \frac{.296214}{.999441} = .296380 \qquad \beta = \frac{16°30'32''}{2} = 8°15'16''$$

$\phi a = 16°30'32''$ SETTING = .143569 × 12 + .5 = 2.2228

SPIRAL GASH HOB

FOR R.H. THD. TOP COMING USE ϕa_1 FOR R. SIDE & ϕa_2 FOR L. SIDE. FOR L.H. THD. TOP COMING USE ϕa_2 FOR R. SIDE & ϕa_1 FOR L. SIDE.

$$\text{COT } \phi a_1 = \frac{\text{COS } v}{\text{TAN } \phi n} - \frac{\text{TAN } v \times F \times K}{\pi \times D} \qquad \text{COT } \phi a_2 = \frac{\text{COS } v}{\text{TAN } \phi n} + \frac{\text{TAN } v \times F \times K}{\pi \times D}$$

$$\text{SETTING R OR L. SIDE} = \left[\text{SIN}\left(\frac{\phi a_1 \text{ OR } \phi a_2}{2}\right)\right] \times 12 + .5$$

EXAMPLE:

$\phi n = 16\tfrac{1}{2}°$, $v = 3°16'$, F = 14, D = 2.9216, K = .150, R.H. TOP COMING

$$\text{COT } \phi a_1 = \frac{.998375}{.296214} - \frac{.057076 \times 14 \times .150}{3.141593 \times 2.9216} \qquad \text{COT } \phi a_2 = \frac{.998375}{.296214} + \frac{.057076 \times 14 \times .150}{3.141593 \times 2.9216}$$

COT ϕa_1 = 3.357393 ; $\phi a_1 = 16°35'10''$ COT. ϕa_2 = 3.383510 ; $\phi a_2 = 16°27'54''$

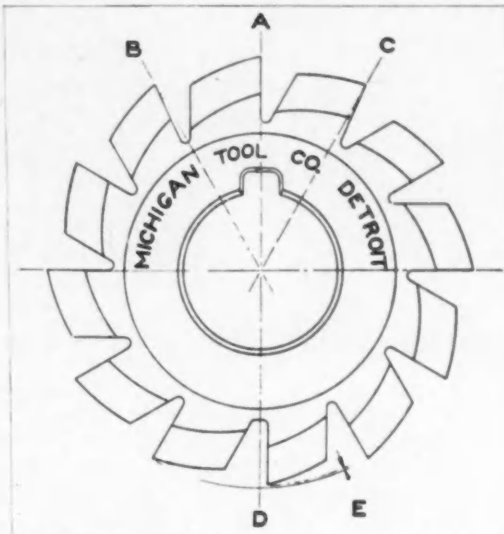
$$\text{SETT. R.S.} = \left[\text{SIN}\left(\frac{16°35'10''}{2}\right)\right] \times 12 + .5 \qquad \text{SETT. L.S.} = \left[\text{SIN}\left(\frac{16°27'54''}{2}\right)\right] \times 12 + .5$$

SETT. R.S. = .144236 × 12 + .5 = 2.2308 SETT. L.S. = .143190 × 12 + .5 = 2.2183

▪ Fig. 13 – Method of figuring correction in pressure angles for both straight and spiral gash hobs for extreme accuracy

When visible flats do appear on hobbed work, the remedy is not to resort to centralizing gages, as is commonly supposed. Such a procedure occasionally may result in better balanced forms, but it will not eliminate the flats. This practice is carried over from the time when most gear teeth were produced by milling with disc cutters, the latter requiring accurate centralization to secure balanced forms.

It is far better to eliminate the cause of such visible flats, rather than attempt to balance the flats. The presence of flats is caused commonly by a combination of too few a number of gashes in the hob, and a low number of teeth



▪ Fig. 15 – Hob with a correctly sharpened tooth, *A*; tooth ground with a hook, *B*; and tooth ground ahead of center, *C*

in the blank, particularly on coarse pitches. In this case the obvious remedy is to increase the number of gashes in the hob.

If the number of teeth to be cut is extremely low, as in automobile starter pinions having five or six teeth, it may not be practical to increase the hob gashes sufficiently to eliminate all visible flats, but such pinions are not used at high speeds and, therefore, accuracy of forms is not so important.

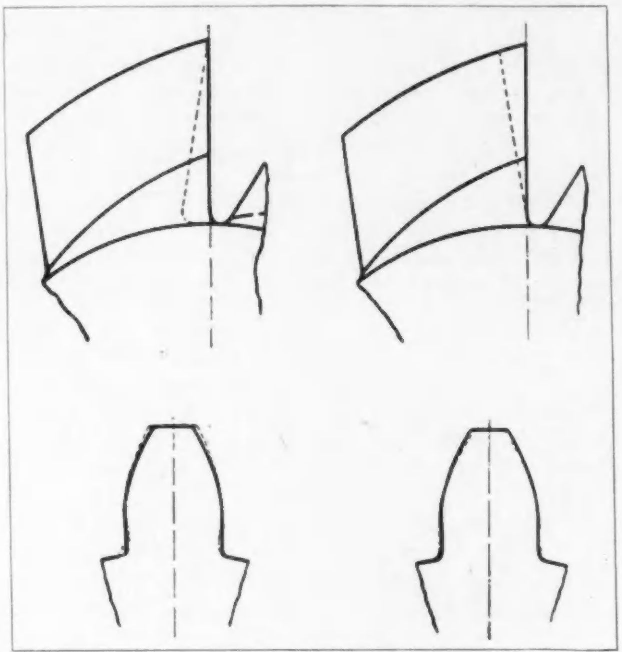
Likewise, when infeeding a worm gear of a low number of teeth where frequently a small worm diameter limits the number of gashes that it is practical to provide in the hob, appreciable flats may be present. These can be eliminated by tangential feeding rather than infeeding.

Inaccurate hobs, particularly unground hobs, sometimes will generate visible flats because of high lead errors. The remedy is obvious.

Faulty sharpening of the hobs, resulting in hob run-out or in erratic spacing of the flutes, may cause the same effect.

Faulty mounting of the hob in the hobbing machine is another possible cause of flats.

For the general run of spur and helical gears, therefore, a hob-centering gage is not necessary. It has been our experience that it is much more important to have the hob run true than to centralize on any particular tooth or space and, for very precise work, to select the most accurate zone in a given hob by axially shifting it to various positions. The errors produced in tooth forms from eccentrically run-



■ Fig. 16 – Enlarged views of hob-tooth grinding errors shown in Fig. 15

ning hobs, and from lead, spacing, and form errors inherent in the hobs, sometimes amount to many times the error from generating flats, for it should be remembered that even if a hob is set central, it does not necessarily follow that accurate tooth forms will be produced. The combined effect of the errors in the hob, and particularly the hob setting, must still be considered.

■ Hob Setting for Precision Gears

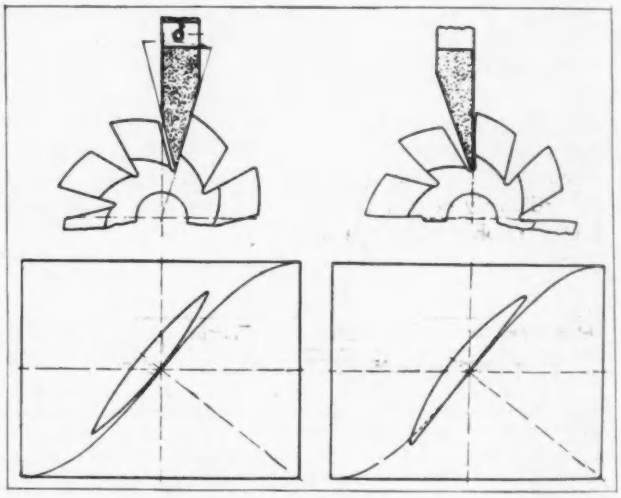
Many users of hobbing machines fail to realize the full importance of proper hob setting. The necessity of having the hob running true, and centralizing it on a tooth or a space when low numbers of teeth are cut, is common knowledge; but the selection of the best cutting zone in a given hob to produce tooth forms of the highest possible accuracy is not so widely understood nor appreciated.

It is not recommended that all of the refinements in hob setting hereinafter described should be employed indiscriminately in the set-up for small lots of "commercial" accuracy, because of the added expense involved. However, for gears requiring the utmost quietness and smoothness of operation, as in turbine reduction sets, automobile transmissions, timing gears, and printing-press machinery, the proper hob setting is often responsible for the satisfactory performance of gears that otherwise would have been rejected.

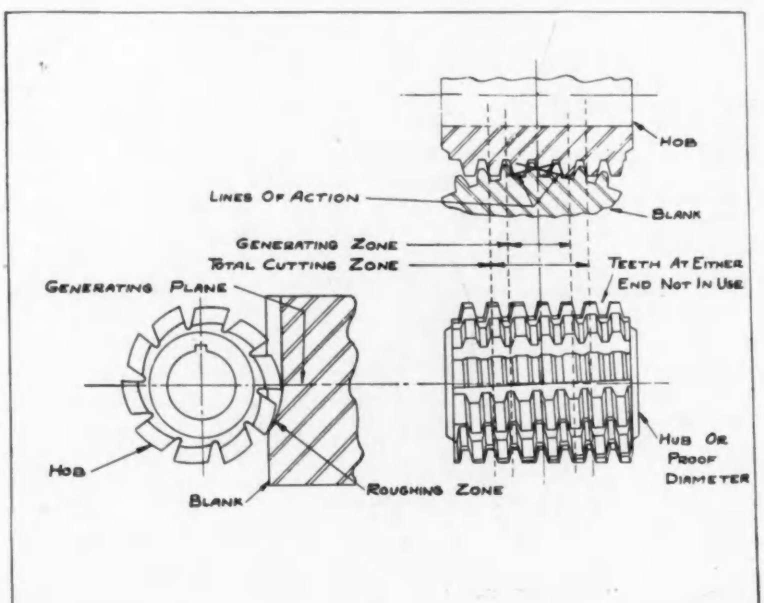
■ Hob Run-Out

Obviously the first consideration in setting a hob is its fit on the arbor, which it should fit without play; otherwise it may shift under cut and the most careful set-up will be of no avail.

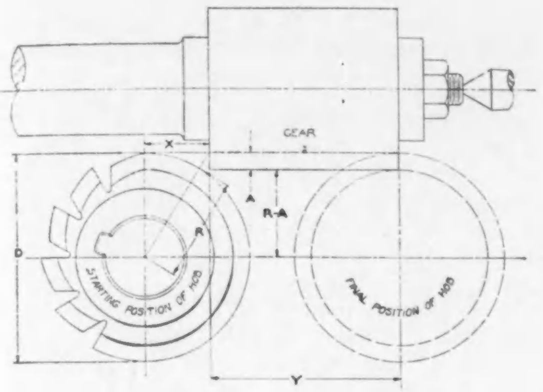
The next step is to clamp the hob on the arbor, using a



■ Fig. 17 (above) – The illustration shows how the flat side of a grinding wheel cuts away the face of a hob tooth as it touches the face far from the center point



■ Fig. 18 (left) – Hob teeth in roughing and finishing zones



■ Fig. 19 – Method of estimating time required for hobbing gears

X — The distance hob must feed from time it starts cutting until it reaches full cutting depth.
N — Number of teeth in gear.
Y — Length of face of gear or gears.
L — Number of threads in hob.
S — Revolutions per minute of hob.
F — Feed per revolution of gear blank.
A — Total depth of cut.
D — Diameter of hob.

FORMULA

(1) $\text{Time} = \dfrac{N\,(Y + X)}{S \times F \times L}$ (2) $X = \sqrt{A\,(D - A)}$

So formula may be written: $\text{Time} = \dfrac{N\,(Y + \sqrt{A\,(D-A)})}{S \times F \times L}$

EXAMPLE

GEAR:
10 D.P.
30 Tooth
.2157 Whole Depth
Face Width 1.5″

HOB:
3″ Dia.
Double Thd.
90 R.P.M.
.050″ Feed

$$\text{Time} = \frac{N\,(Y + \sqrt{A\,(D-A)})}{S \times F \times L}$$

$$\text{Time} = \frac{30\,(1.5 + \sqrt{.2157\,(3 - .2157)})}{90 \times .050 \times 2}$$

$$\text{Time} = 7.58$$

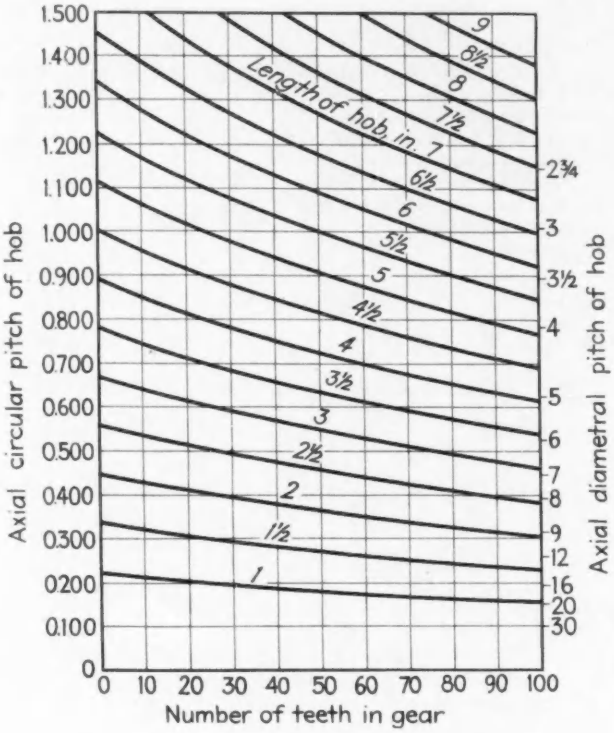
minimum number of spacing collars. The outer bearing should be in place before applying the wrench to the clamping nut; otherwise, the arbor may be damaged by springing it out of line.

The hub or proof diameters should be checked by means of an indicator graduated in ten-thousandths of an inch. An indicator graduated in half-thousandths or thousandths cannot be relied upon for this class of work. Run-out should be within 0.0002 in., and the high points of the run-out on each end of the hob should be in line. Any eccentricity will produce an error in tooth form, particularly when the high points of the run-out are opposite on the two ends of the hob.

If the run-out of the hub diameters exceeds the limit stated, it can be corrected readily by loosening the clamping nut, rotating the spacing collars, and re-clamping. The slight errors in parallelism of the faces of the collars are thereby utilized to bring the hob into correct alignment. It may be necessary to repeat this process a few times before the hob runs satisfactorily. With a little practice the oper-

■ Fig. 20 – Hob cutting speeds

Diam. Inches	FEET PER MINUTE																			
	15	20	25	30	35	40	45	50	55	60	65	70	80	90	100	110	120	130	140	150
	REVOLUTIONS PER MINUTE																			
1/4	229	306	382	458	535	611	688	764	840	917	993	1070	1222	1375	1528	1681	1833	1986	2139	2292
3/8	153	204	255	306	357	407	458	509	560	611	662	713	815	917	1019	1120	1222	1324	1426	1528
1/2	115	153	191	229	267	306	344	382	420	458	497	535	611	688	764	840	917	993	1070	1146
5/8	92	122	153	183	214	244	275	306	336	367	397	428	489	550	611	672	733	794	856	917
3/4	76	102	127	153	178	204	229	255	280	306	331	357	407	458	509	560	611	662	713	764
7/8	65	87	109	131	153	175	196	218	240	262	284	306	349	393	437	480	524	567	611	655
1	57	76	96	115	134	153	172	191	210	229	248	267	306	344	382	420	458	497	535	573
1 1/8	51	68	85	102	119	136	153	170	187	204	221	238	272	306	340	373	407	441	475	509
1 1/4	46	61	76	92	107	122	138	153	168	183	199	214	244	275	306	336	367	397	428	458
1 3/8	42	56	70	83	97	111	125	139	153	167	181	194	222	250	278	306	333	361	389	417
1 1/2	38	51	64	76	89	102	115	127	140	153	166	178	204	229	255	280	306	331	357	382
1 5/8	35	47	59	71	82	94	106	118	129	141	153	165	188	212	235	259	282	306	329	353
1 3/4	33	44	55	66	76	87	98	109	120	131	142	153	175	196	218	240	262	284	306	327
1 7/8	31	40	51	61	71	82	92	102	112	122	132	143	163	183	204	224	244	265	285	306
2	29	38	48	57	67	76	86	96	105	115	124	134	153	172	191	210	229	248	267	287
2 1/4	25	34	42	51	59	68	76	85	93	102	110	119	136	153	170	187	204	221	238	255
2 1/2	23	31	38	46	54	61	69	76	84	92	99	107	122	138	153	168	183	199	214	229
2 3/4	21	28	35	42	49	56	63	70	76	83	90	97	111	125	139	153	167	181	194	208
3	19	25	32	38	45	51	57	64	70	76	83	89	102	115	127	140	153	166	178	191
3 1/2	16	22	27	33	38	44	49	55	60	66	71	76	87	98	109	120	131	142	153	164
4	14	19	24	29	33	38	43	48	53	57	62	67	76	96	96	105	115	132	134	143
4 1/2	13	17	21	26	30	34	38	42	47	51	55	59	68	76	85	93	102	110	119	127
5	12	15	19	23	27	31	34	38	42	46	50	54	61	69	76	84	92	99	107	115
5 1/2	10	14	17	21	24	28	31	35	38	42	45	49	56	63	70	76	83	90	97	104
6	9	13	16	19	22	26	29	32	35	38	41	45	51	57	64	70	76	83	89	96
7	8	11	14	16	19	22	25	27	30	33	36	38	44	49	55	60	66	71	76	82
8	7	10	12	14	17	19	22	24	26	29	31	33	38	43	48	53	57	62	67	72
9	6	9	11	13	15	17	19	21	23	26	28	30	34	38	42	47	51	55	59	64
10	5	8	10	12	13	15	17	19	21	23	25	27	31	34	38	42	46	50	54	57
11	5	7	9	10	12	14	16	17	19	21	23	24	28	31	35	38	42	45	49	52
12	4	6	8	9	11	13	14	16	18	19	21	22	26	29	32	35	38	41	45	48



■ Fig. 21 – Recommended length of worm-gear hobs

ator will be able to true up the hob in a few minutes' time. This is one of the most important elements in hob setting, and it will be found that the extra time required will be justified amply by results.

It is very necessary, in the first place, to make hobs accurately when precision gears are required, and it is equally important to see that the hob is sharpened properly and mounted properly on the machine. All the good that has been put into the hob is destroyed when it is mounted inaccurately. In other words, if a hob runs out on a hobbing machine, it has the same effect as a drunken lead.

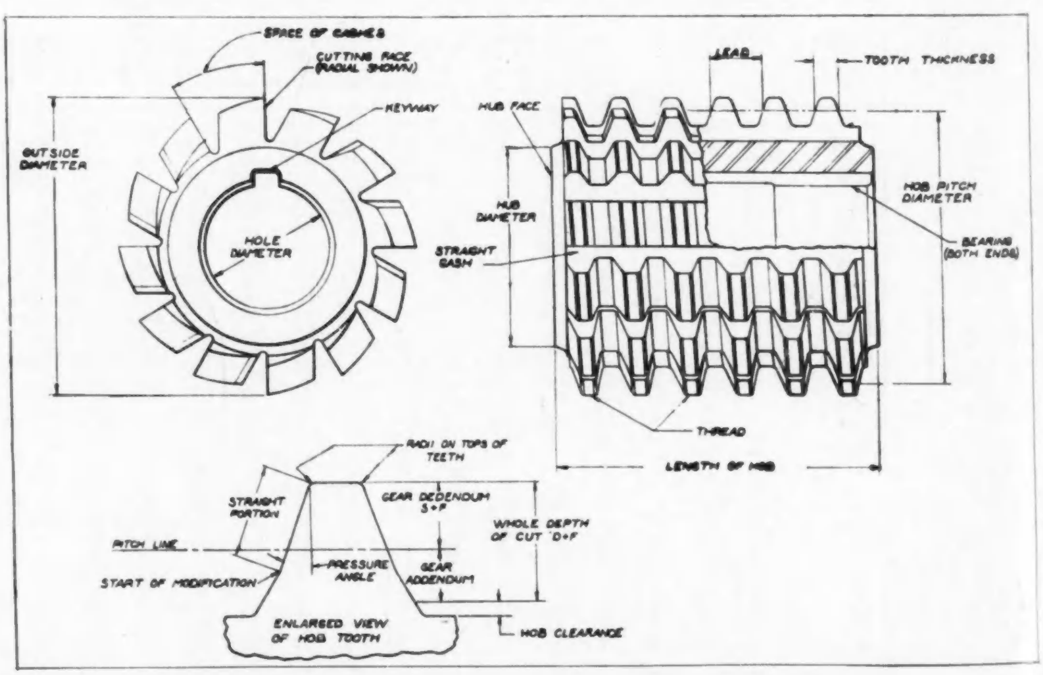
■ Composite Effect of Hob Cutting Teeth

The composite effect of errors in the hob teeth is another important element influencing tooth form. Even though the most accurate ground hob obtainable is used, the errors of which may be within 0.0002 in. for form, tooth spacing, and weave, the combined effects of these errors may affect the form several times the amount of the individual hob tooth errors. For example, suppose a hob has 60 teeth, 25 of which comprise the cutting zone when producing a certain gear; of the 25 teeth, perhaps 10 serve as roughing teeth only, and the remaining 15 actually generate (see Fig. 18). Each of the 15 teeth will form a portion of the involute curve. See Fig. 4.

It will be seen readily that, although each tooth may be individually accurate, errors in spacing will cause certain teeth to cut too much on one side and not enough on the other, thus modifying the involute considerably. It may appear that, with 15 teeth in the generating zone, a wide departure from the true involute is possible; in practice, however, the errors cancel each other to a large extent. By shifting a hob axially, different composite sets of teeth of the hob comprise the generating zone; by this means it is possible to select one or more best positions of a given hob when cutting a gear of a given number of teeth.

■ Recording the Hob Positions

When satisfactory positions of a hob have been determined, they should be recorded by means of a special micrometer setting device attached to the machine. The high points and amount of run-out may be marked on the hob with blue vitriol solution. Thus, on future jobs, the hob positioning can be duplicated without resorting to trial or dummy gears. Frequently several good positions will be found on one hob, all of which then can be utilized before sharpening is required. Sharpening the hob may affect the



■ Fig. 22 – Nomenclature of hobs

form slightly; it is therefore best to repeat the trial cuts and check for form after each sharpening.

Another important point in hobbing accurate gears is to be sure that the hobbing machine is in good mechanical condition. By good condition, we do not mean only good ways and bearings on the machine, but also accurate index and feed gears. This is very important, as the accumulated errors in the index and feed gears and also the lead screw will show up in the gears being cut.

It would be a good policy every so often to check the angular velocity of the hobbing machine with instruments provided for this purpose.

It is the practice at times when cutting gears of heavy pitches to take a rough and finish cut, thus obtaining better accuracy. In roughing, single, double, and triple-thread unground hobs are used. It is not good practice to use greater than a triple-thread hob for roughing because there will be no economy effected. The additional cost and variation of hobs greater than triple thread will more than offset the time saved in cutting, and the quality of work will not be as good.

Fig. 19 is a chart for use when estimating cutting time required for hobbing a load of gears, and is useful when making a time study of a job.

Fig. 20 is a very useful chart when setting up a new job. We might add here that it is just as important to run hobs at the proper speeds as it is to mount the hobs properly. To run hobs too fast or too slow, the highest efficiency cannot be obtained. The determining factors for hob speeds are the type of material being cut, its hardness, physical properties, and the quality of finish required.

Fig. 21 is a graph showing the recommended length of worm gear hobs. The length can be determined easily when the axial circular pitch and the number of teeth in the worm gear to be cut are known.

Fig. 22 shows the nomenclature of hobs. The terminology shown here is used generally in connection with hobs, and is followed as standard by all hob manufacturers.

Severe Engine Conditions – Effect on Oil and Fuel

WITH the passing of the Model T Ford, special compounded oils disappeared from the market. Then specifications appeared to be the recommended practice and even the SAE Handbook contained minimum specifications to assure motor oil quality. Next the Sligh and Indiana oxidation tests were introduced because engine conditions were becoming more severe and the sludge or dirty-oil problem was to the fore. About this time the SAE campaign for the use of 10-W and 20-W motor oils was started, and the corrosion of copper-lead and leaded-bronze bearings was introduced. Not alone was corrosion confined to the leaded-bronze or alloy bearings, but the cadmium-silver bearings operating with crankcase temperatures as high as 350 F at high speeds brought this corrosion phase of severe engine conditions to a head. The problem was ably reported, and corrective methods and measures were taken both by the automotive and the oil companies.

In 1937, the trend toward high-output aircraft and solid-injection diesel engines imposed severe conditions on the lubricant, and Wright aircraft and Caterpillar diesel engines required the Wright compounded aircraft oils and the Caterpillar diesel oils for severe-duty engine service. This condition was brought about because marked improved performance of such lubricants in these makes was necessitated by certain design and operating conditions and to give freedom from ring-sticking and reduce excessive ring and liner wear. It soon was found that it was necessary to take current design requirements and study the service effects on the lubricating oil.

At the SAE World Automotive Engineering Congress in New York, May, 1939, the writer discussed the test made on a fleet of 1½-ton Chevrolet trucks hauling 14,000 lb between Chicago and Pittsburgh and return, and it was stated that no straight mineral or commercial oil on the market would lubricate these engines for 2000 miles without excessive lacquer on pistons and valve mechanism. This operation presented a new problem of oxidation caused by high crankcase oil temperatures and, with it, the introduction of the Underwood oxidation test. About this time it became apparent that used oil analyses were not always definitely related to the condition of the engine, for it is possible to have a very bad used oil analysis with a clean engine and it is also possible to have a good used oil analysis with a dirty engine.

Today the problem of severe duty has been extended to the GMC 71 diesel engine which has some of the characteristics of the diesel engine piston lacquer and ring-sticking tendencies and the oxidation and bearing corrosion conditions of the high crankcase oil temperatures of the gasoline engine.

The ever increasing severity of high-duty engine conditions makes a cycle approaching the period of the Model T Ford.

It appears that a Caterpillar oil, a Wright aircraft oil, a GMC 71 diesel oil, a Hercules diesel oil and a heavy-duty truck oil might be needed. However, full-scale engine tests, single-cylinder engine tests, and laboratory tests are being run with the view of studying anti-oxidants, lacquering and corrosion action in the hot crankcase oil and for the detergent or purging action of the piston and rings; to keep products of incomplete combustion from adhering to the piston surfaces.

There is going on today one of the most intensive studies of the oil and fuel problem in the petroleum, chemical and automotive industries ever witnessed in the field of severe duty engine conditions. In the year 1939 over 200 patents were issued for addition agents for oils and fuels; yet the final, utopian addition agent for present day needs for clean engines is still on the proving ground in truck, bus and tractor operators' equipment.

Excerpts from the paper: "Severe-Duty Engine Conditions as Related to Oil and Fuel," by C. M. Larson, chief consulting engineer, Sinclair Refining Co., presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 14, 1940.

An Instrument for Continuous

Measurement o

THE problem of developing an instrument for the continuous measurement of piston temperatures by the thermocouple method can be divided into three parts: 1. The measuring instrument or instruments necessary; 2. The installation of the thermocouples in the piston; 3. The necessary electrical circuit connecting the two.

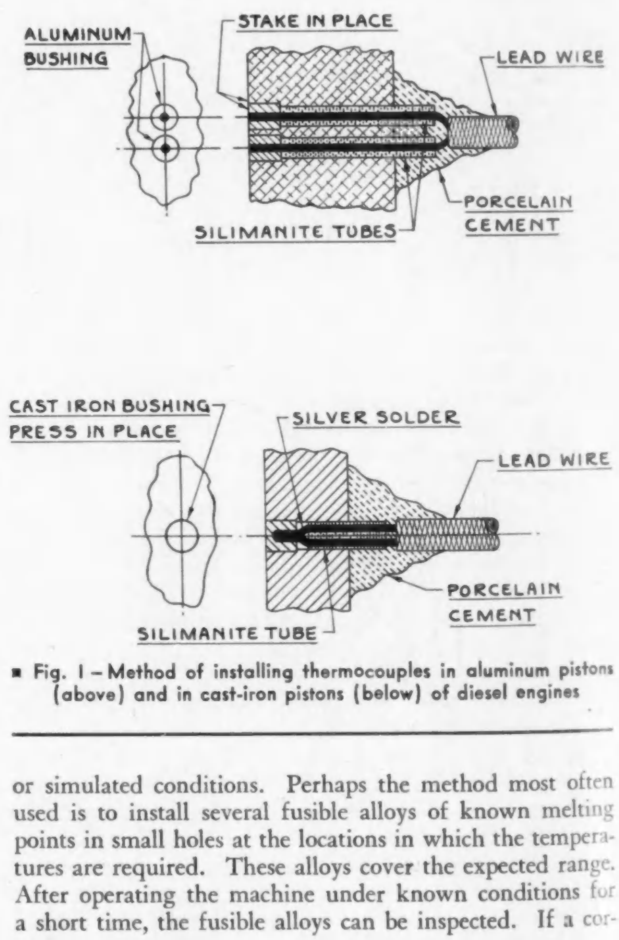
After reviewing the principle of operation of the potentiometer method of reading temperatures, the authors describe the standard light-beam galvanometer selected for use with the intermittent contacts. The sensitivity of the galvanometer, they explain, is controlled by means of a variable-ballast resistor connected across its terminals. A multiple switch makes possible the continuous measurement of one temperature or the intermittent measurement of several temperatures.

Test data obtained with the instrument presented in this paper include temperature gradients of both aluminum and cast-iron pistons; the effect of load variation on piston temperatures; and the effect of spark advance on piston temperature.

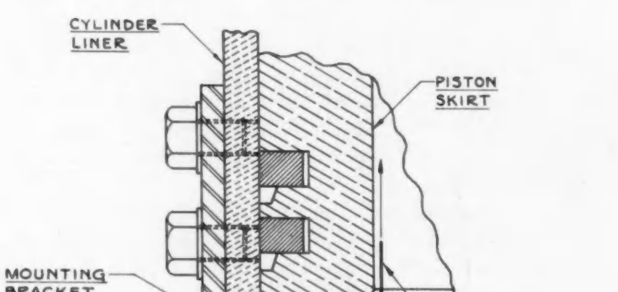
They announce that a production engine fitted with the final design of the instrument has been run 25,000 miles at 60 mph on a dynamometer with no attention or adjustments, and that a test-car installation has shown that measurements can be made continuously on the road.

THE need for a method to determine the temperatures of moving parts of an engine and those of reciprocating parts in particular, has been established as a fact for a long time. It has become more and more evident that the development of modern high-output engines has been retarded by the inability of pistons to withstand the high temperatures to which they are subjected. This is true of both gasoline aviation engines and two-cycle diesel engines.

A survey of the technical literature will disclose many methods which have been used with varying degrees of success to determine piston temperatures under operating



■ Fig. 1 – Method of installing thermocouples in aluminum pistons (above) and in cast-iron pistons (below) of diesel engines



■ Fig. 2 – Piston thermocouple contactor

or simulated conditions. Perhaps the method most often used is to install several fusible alloys of known melting points in small holes at the locations in which the temperatures are required. These alloys cover the expected range. After operating the machine under known conditions for a short time, the fusible alloys can be inspected. If a cor-

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 14, 1940.]

# of PISTON TEMPERATURES

by ARTHUR F. UNDERWOOD and A. A. CATLIN
Research Laboratories Division, General Motors Corp.

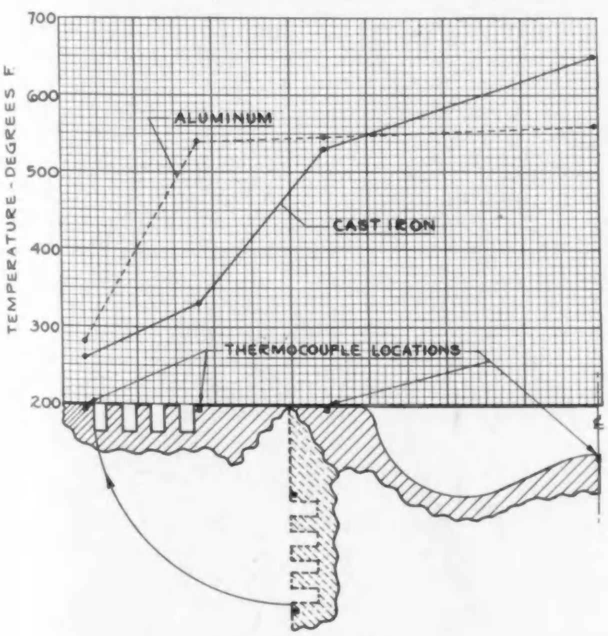
rect guess has been made, it will be found that melting of one or more of the lower alloys has occurred and that these alloys have fallen out, while the remainder are still in place. It is then reasonable to assume that the actual operating temperature is within the range between the highest to melt and the lowest to remain. Considerable time and trouble are often required in obtaining the desired results. Often the alloys do not have definite melting points so that inspection after a test shows a particular high-temperature alloy has disappeared although a lower-temperature one is still in place. Also, some of these alloys are subject to deterioration. Ordinarily, only one determination can be made from each set-up; therefore, the time required to obtain a curve showing the effect of several variables may consume several days or weeks.

A better method has been the use of a thermocouple as is employed for the determination of the temperature of stationary objects and of liquids. To maintain a continuous electric circuit, numerous investigators have used various types of lever schemes, commonly referred to as "grasshopper actions," along which the thermocouple lead wires were lashed. Obviously, this method requires more room than is available in the ordinary engine crankcase, and it is not suitable where high inertia forces are involved.

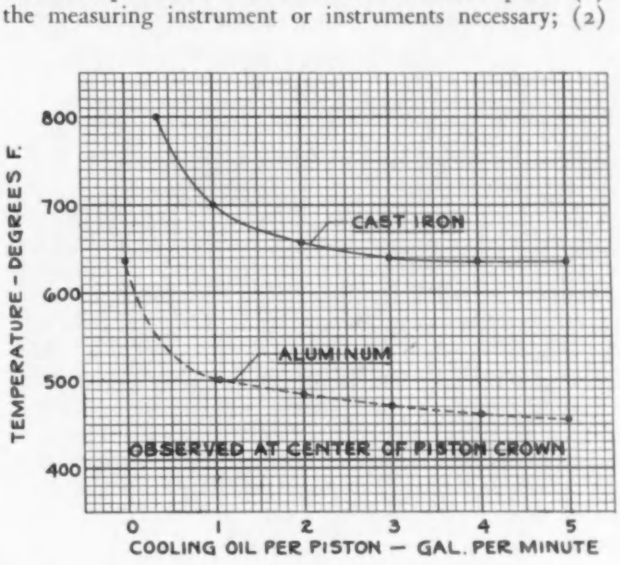
During the development of high-output two-cycle diesel engines, it became quite important to measure the temperature of pistons so that the operating conditions might be controlled, and so that the causes of ring-sticking and carbon formation on the underside of the piston might be better understood. Some five years ago, the development of a more suitable method for the continuous measurement of these temperatures was undertaken.

The problem of measuring piston temperature by the thermocouple method can be divided into three parts: (1) the measuring instrument or instruments necessary; (2) the installation of the thermocouples in the piston; and (3) the necessary electrical circuit connecting the two.

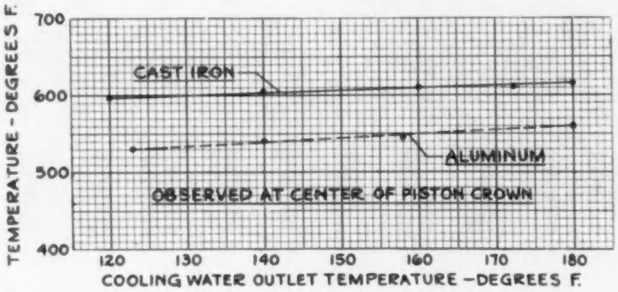
It is necessary first to recall the potentiometer method of reading temperatures to understand the principle of operation. An iron-constantan junction located at the point where the temperature is to be read produces a voltage which depends on its temperature. The wires are led to a potentiometer which, in effect, is a Wheatstone bridge. By means of the adjustable slide wire in the potentiometer box, the voltage of a standard cell is balanced against the junction voltage until a galvanometer in the circuit shows that no current is flowing. When this adjustment has been accomplished, the resistance of the circuit has no



■ Fig. 3 – Temperature gradients of aluminum and cast-iron pistons – 8 x 10 in., 2-cycle diesel engine, 105 lb per sq in. mep, 750 rpm



■ Fig. 4 – Effect of cooling oil on aluminum and cast-iron piston temperatures – 8 x 10 in., 2-cycle diesel engine, 105 lb per sq in. mep, 750 rpm



■ Fig. 5 – Effect of engine coolant temperature on aluminum and cast-iron piston temperatures – 8 x 10 in., 2-cycle diesel engine, 105 lb per sq in. mep, 750 rpm

influence. The slide wire is calibrated in F to correspond to the correct temperature of the iron-constantan junction.

Therefore, it is only a question of determining that no current is flowing during the interval that the circuit is closed. If intermittent contacts are made, a more sensitive galvanometer may be required to determine when the circuit is balanced because of the shorter contact interval. However, as will be brought out later, the time interval of contact can be decreased too greatly no matter how sensitive the galvanometer is made. The regular galvanometer supplied with the commercial potentiometer selected for use was replaced by a standard light-beam galvanometer. The sensitivity of the latter was controllable by means of a variable ballast resistor connected across its terminals. A multiple switch completed the instrumentation and made possible the continuous measurement of one or the intermittent measurement of several temperatures.

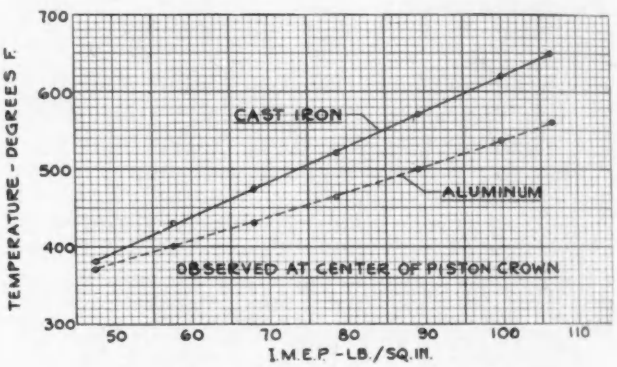
The general methods of installing thermocouples in both the aluminum and cast-iron diesel engine pistons are shown in Fig. 1. Numerous variations of the installations shown have been used. Each design of piston has presented its own installation problems. It is imperative that the lead wires from the thermocouples be secured tautly to the piston to prevent them from fatiguing and causing an open circuit. They also must be well insulated. Cotton-covered enameled wire soon chafes through, causing a short-circuit. Rubber insulation is likewise unsatisfactory even when covered with an impregnated cotton loom. The use of double "spaghetti" covering has been quite satisfactory, but it chars readily at elevated temperatures. Surprisingly, the installation of the thermocouples has been the most troublesome part of this temperature-measuring scheme.

The contactor shown in Fig. 2 was used to complete the electrical circuit. One pair of contact points was mounted in a bakelite block which was fastened securely to the lower edge of the piston skirt below the wristpin boss. A second pair of points was mounted likewise in a bakelite block mounted on a bracket attached rigidly to the cylinder liner. This second pair of points was spring-loaded in the vertical plane with the potentiometer lead wires connected to them through the springs. Thus, it can be seen that the thermocouple circuit will be closed at the bottom of each stroke and that, with the galvanometer as an indicator, it is possible to balance the voltages and read the temperatures as in an ordinary arrangement.

Before installation on an engine, a single element of this device was tested thoroughly by a bench set-up. Contact points made of the basic thermocouple material were used. Satisfactory operation was obtained at all speeds up to 1800 rpm with a cam having a maximum lift of 0.025 in. and a contact duration of 25 deg of shaft rotation. No perceptible wear of the points was observed, nor did oil on them affect the results.

This equipment first was installed on a single-cylinder, two-stroke cycle, diesel engine which had a bore of 8 in., a stroke of 10 in., and a rated speed of 750 rpm. Thus, the average piston speed was 1250 fpm. The engine was equipped so that the intermittent observation of six temperatures was possible.

During the course of a general engine-development program, numerous piston temperature data were obtained. The temperature gradients of both aluminum and cast-iron pistons are shown in Fig. 3. These gradients are typical and vary somewhat with design and other factors. It has been found that, in well-designed pistons, these gradients are essentially symmetrical with respect to the axis of the piston. For conversational purposes, the entire crown of the aluminum piston above the top compression ring had a uniform temperature. On the other hand, the cast-iron piston had a sharp temperature gradient in this region.



■ Fig. 6 – Effect of load on aluminum and cast-iron piston temperatures – 8 x 10 in., 2-cycle diesel engine, 750 rpm

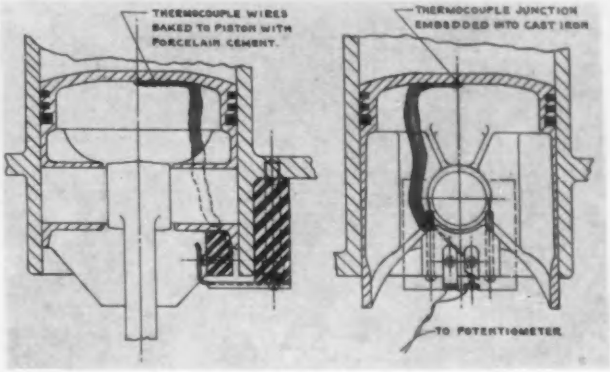
The use of oil to cool the pistons has many advantages among which are: the general reduction of the temperature of the piston crown which reduces the thermal stresses present; the use of less piston running tolerance; and an increase in ring life. The amount of cooling oil supplied to these pistons had a marked effect on the temperatures observed. The thermocouple located in the center of the piston crown was found to be the most critical in this respect, although thermocouples located elsewhere in the crown indicated similar tendencies to a somewhat less degree. This effect is shown by Fig. 4. As previously discussed, the curve for the center of the aluminum piston crown is representative of the entire crown temperature. A small amount of oil, delivered to the piston for cooling purposes, oxidized rapidly and formed varnish or carbon deposits on the hot surfaces. The use of a baffle in the piston to trap a small amount of the cooling oil, thus simulating a "cocktail-shaker action" to induce scrubbing, tends, in general, to reduce the piston temperature and also the oxidization of the oil. It is interesting to note that any reasonable quantity of piston cooling oil will not keep the aluminum piston below the coking temperatures

of ordinary oils immediately adjacent to the top compression ring.

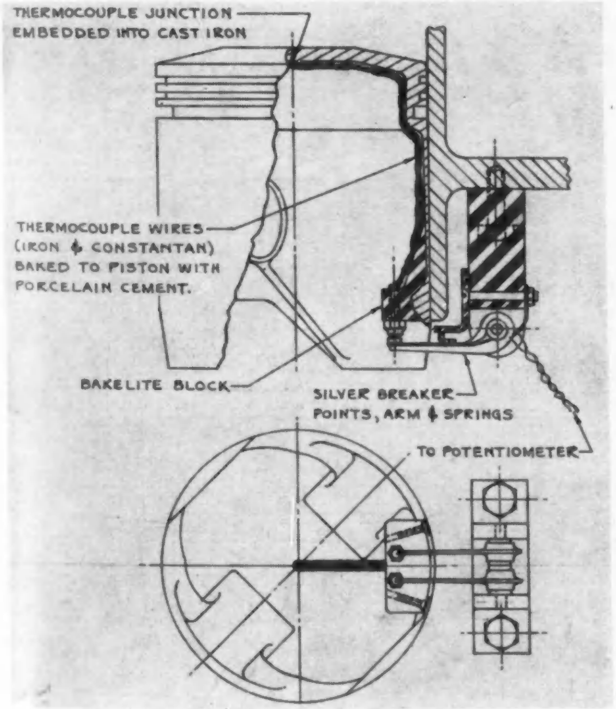
As shown in Fig. 5, the engine coolant temperature has but little influence on the temperatures of either aluminum or cast-iron pistons. Other variables being constant, the piston temperatures are directly proportional to the coolant temperature within the normal range. The effect of load variation on piston temperatures is shown in Fig. 6. As with the engine coolant, these temperatures are also directly proportional to load.

Many hundreds of hours' operation clearly demonstrated the practicability of this device. Numerous sundry difficulties were overcome readily, but the method of installing the thermocouple lead wires in the diesel engine pistons was never satisfactory. Surprisingly, the contactor was never a major problem and operated satisfactorily for two or three hundred hours without repair. With this instrument it was usually possible to duplicate results from day to day within the limit of experimental error.

During tests on the stability of lubricating oil, the need for continuous measurement and control of the piston temperature was evident as it was known that the degree of heat had the effect on the piston head of a "hot plate" within the engine which cooked the oil thrown against it. Thus, too hot a piston caused ring-sticking and formation of carbon on the underside of the piston, which falls into the oil pan. An investigation for a suitable arrangement was started. It was decided that the engine should operate at least to 3500 rpm which approximates where full horsepower is ordinarily obtained on automotive engines, and that it could be read continuously over long periods of operation with no maintenance. Therefore, a life of not under 10,000 miles was taken to insure a thorough test on an engine without disturbing it for repairs. Such freedom



■ Fig. 8 – Set-up for high speeds employing silver sliding contacts

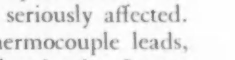
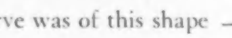


■ Fig. 7 – Set-up for preliminary investigation

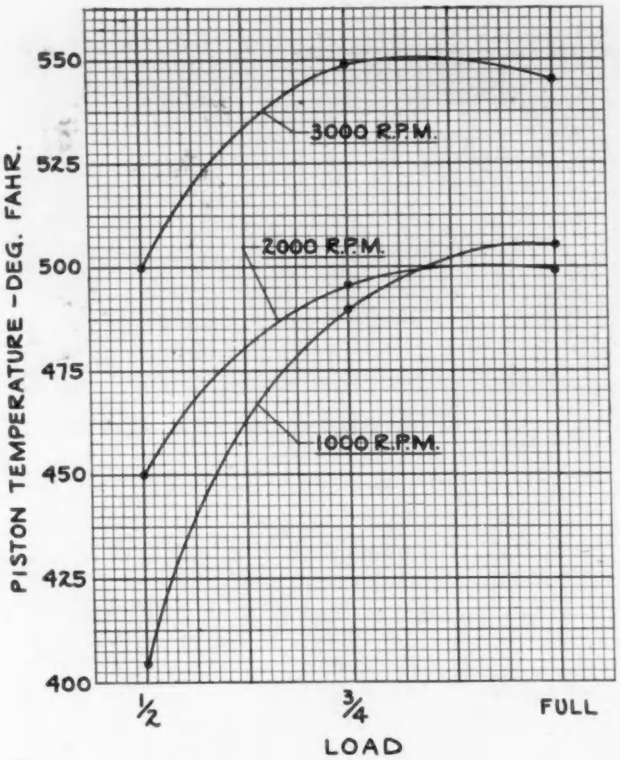
from mechanical difficulties would be especially valuable to the West Coast on road testing.

The basic method of employing intermittent contact was used with suitable changes so that the temperatures of pistons at 3500 rpm could be read for long periods of operation. Fig. 7 shows the first arrangement with the thermocouple located at the center of the piston head. The wires were placed against the inside wall and covered with a porcelain cement which was baked into place. By means of two countersunk-head screws, a bakelite block was attached to the bottom of the piston skirt. To the engine block was attached another bakelite block. On these two insulated blocks were supported two regular breaker points which had contacts made of the same material as the wires that were attached, namely, iron and constantan, so that no extraneous voltages possibly could be developed.

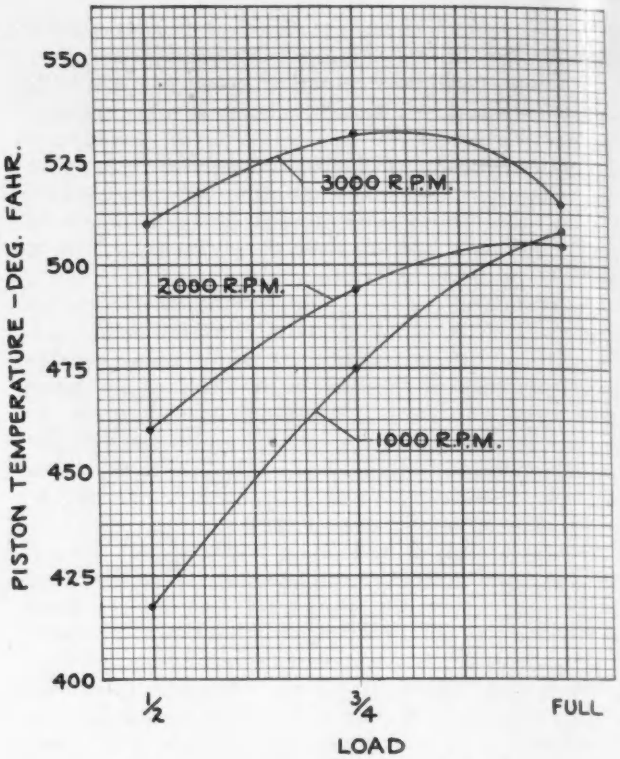
Before installation in an engine, it was believed advisable also to investigate this device on a set-up which was essentially a single-cylinder apparatus that could be motored over with a dynamometer up to about 2500 rpm. It soon was discovered that the contacts did not make good electrical connections, and very erratic results were obtained at these higher speeds. Silver was suggested as the best material since it does not oxidize readily and does have high conductivity. Because both contacts and the nearby lengths of wire are at the same temperature, it was realized that there would be no error involved. Pure silver showed itself to be entirely satisfactory from an electrical viewpoint.

A 1938 6-cyl engine then was fitted with the arrangement shown in Fig. 7. Piston temperature at the center of the head could be read at all speeds up to about 2600 rpm, at which point the sensitivity was seriously affected. An oscillograph was attached to the thermocouple leads, and a small voltage was impressed on the circuit. It was discovered that, at slow and moderate speeds (up to 2000 rpm), the voltage curve was of this shape  rapidly building up to a maximum and, at 2600 rpm, the curve had changed to . The voltage curve did not reach its maximum value and was very erratic. A large part of the trouble may have been caused by bouncing of the points.

For the slow speeds these contacts are suitable. Therefore, for use in diesels and moderate-speed engines, they



▪ Fig. 9 – Piston temperature versus load and speed – 6-cyl 1938 engine – 57-octane fuel



▪ Fig. 10 – Piston temperature versus load and speed – 6-cyl 1938 engine – 70-octane fuel

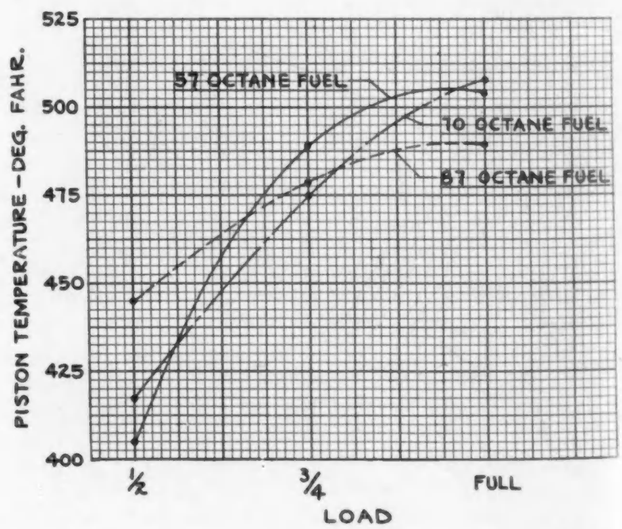
make a conveniently obtainable arrangement. Silver should replace the standard tungsten metal; it can be soldered to the steel of the breaker arm and screw.

It was decided that, at higher speeds of 3500 to 4000 rpm, the time interval would be far too short unless the movement of the breaker arm was greatly increased. Sliding contacts seemed to be the better method because a long contact could be made with very little deflection of the component parts.

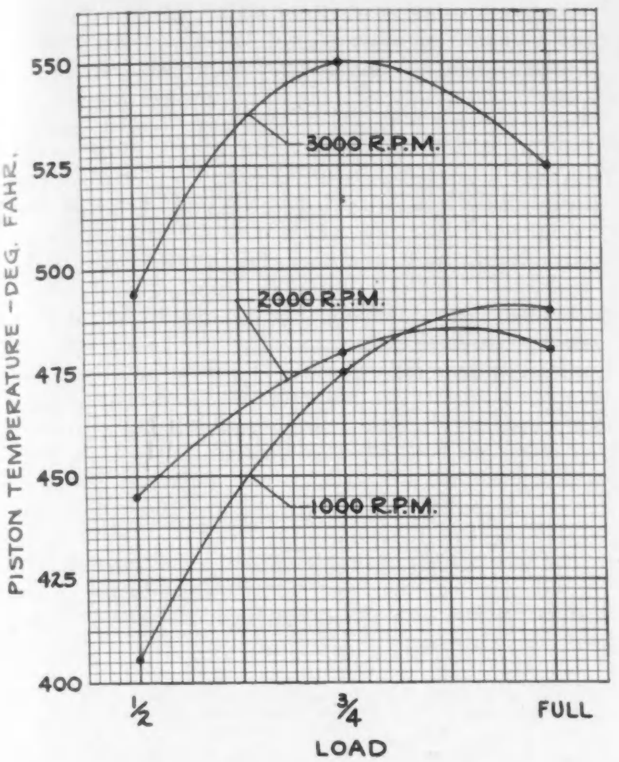
By reducing the deflection, a corresponding reduction in inertia forces would be gained to maintain better contact. Fig. 8 illustrates this type. The contacts have been shifted to the side of the insulating block of the piston. The insulated block can be attached to the piston at any convenient location that will not interfere with the crankshaft or connecting rod. In the installation shown, the block is screwed to the underside of the piston pin boss. Bent "fingers" of heat-treated spring steel are used as springs. Only 0.010-in. deflection is allowed, thereby reducing the movement of the springs to give a better "following" characteristic. Pure silver slides and points were used.

This arrangement was used for extensive tests on an engine of 3¾-in. stroke on which oil-stability tests were being run. To show the type of results which could be obtained, the effects of several variables on piston temperature were measured. It is to be remembered that these data are not intended to be exact as no special precautions were taken to control all the variables. Rather, the curves were made to indicate how the thermocouple could be used to study the effect of various factors on piston temperatures. The thermocouple was located at the center of the piston head as the hottest temperature was desired for the work on oil stability.

Figs. 9, 10 and 11 show the effect of load and speeds on the temperature of No. 5 piston for each of three gasolines having 57, 70, and 87-octane rating, respectively. The same data are plotted on Figs. 12, 13, and 14 to show more directly the effect of octane number and load on piston temperature at three different speeds. It is interesting to note that an increase in speed and load does not always produce a higher temperature. Figs. 9, 10, and 11



▪ Fig. 12 – Piston temperature versus load and octane number – 6-cyl 1938 engine – 1000 rpm



■ Fig. 11 – Piston temperature versus load and speed – 6-cyl 1938 engine – 87-octane fuel

show a slightly lower temperature at 2000 rpm as against 1000 rpm for full load. On the same curves, an increase in load from ¾ to full, shows a decrease in temperature when running at 3000 rpm. The use of 87-octane gasoline shows a reduction in piston temperature under conditions when knock is most likely to occur. Full load at 1000 to 2000 rpm on Figs. 12, 13 and 14 shows this effect.

The results on Fig. 15 are given to illustrate the effect of spark advance on piston temperature when running on each of three fuels. The temperature rises rapidly with spark advance except on 87-octane fuel. Some unknown combination of factors caused the temperature to decrease on the latter fuel at the detonating point. To show further what type of data can be obtained, Fig. 16 has been included for demonstrating the relationship of temperature to piston cooling by means of a jet of lubricating oil impinging against the underside of the piston head. As the pressure on the jet is increased to throw a greater volume of oil up to the piston, the temperature drops at a decreasing rate.

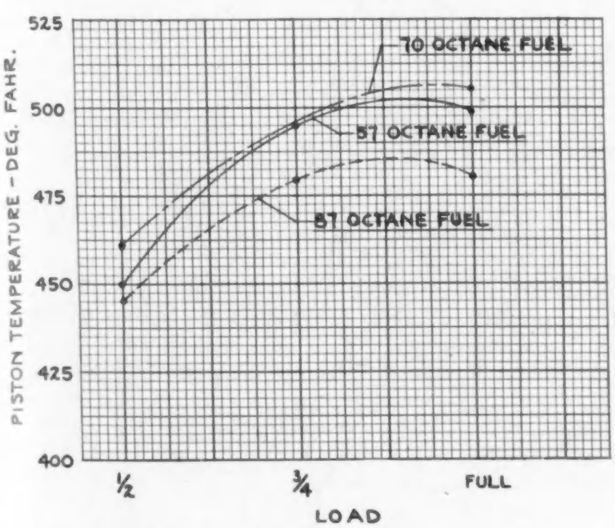
As soon as this testing was done, the engine was returned to oil-stability tests during which the piston temperature was to be maintained at 500 F. This engine was used as a piece of apparatus to test lubricating oil. To give controlled conditions, it was desirable to keep the hot piston head at a constant known value.

Considerable trouble was encountered in that the engine clearance increased rapidly and the oil temperatures could not be maintained at 280 F. Investigation brought out that the porcelain cement holding the wire to the inside of the piston had spalled off in minute amounts. The lubricating oil circulated the grains and actually wore the engine out. As an example, the piston ring gap became as much as ⅛ in. Also, it was established that the pure silver contacts would wear out in 2000 to 3000 miles.

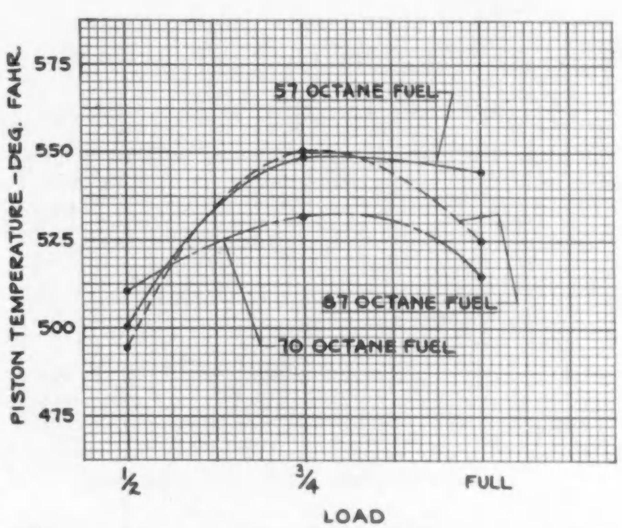
■ Present Design

To have a thoroughly practical instrument, it was proposed that one should be developed that would run 10,000 miles without attention. The first requirement was to remove the porcelain cement. Inquiry brought out the fact that we could not obtain a cement which would withstand the high temperature and be free from mineral filler; therefore, it seemed best to remove the thermocouple wire from the hot piston surface so that an ordinary gasket cement could be used. Secondly, the life of the contacts would have to be extended.

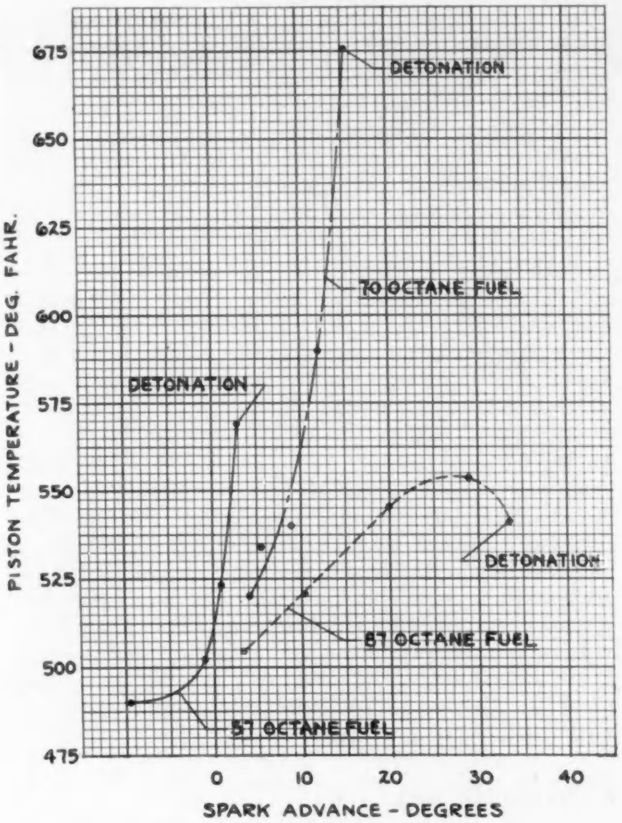
The final design is shown in Fig. 17. By use of the



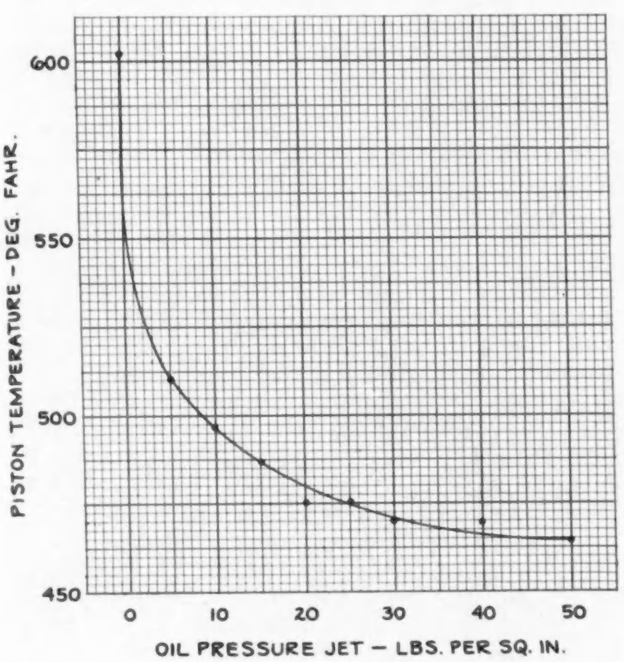
■ Fig. 13 – Piston temperature versus load and octane number – 6-cyl 1938 engine – 2000 rpm



■ Fig. 14 – Piston temperature versus load and octane number – 6-cyl 1938 engine – 3000 rpm



■ Fig. 15 – Piston temperature versus spark advance and octane number – 6-cyl 1938 engine; water, 194-200 F; oil sump, 197-205 F; 1000 rpm; full load

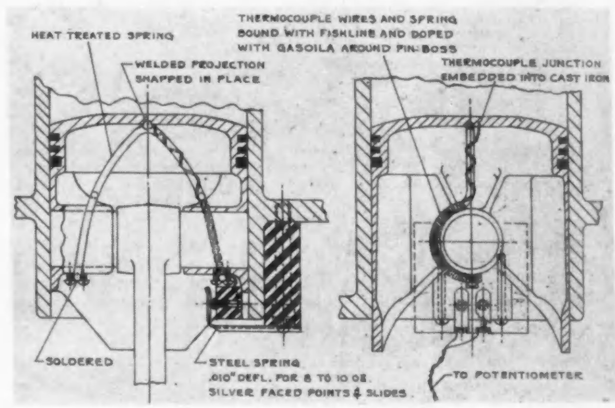


■ Fig. 16 – Piston temperature versus pressure on jet of lubricating oil impinging against the underside of the piston head – 6-cyl 1938 engine

"wishbone" supports made of spring wire and snapped into place, the thermocouple wire is led away from the hot piston head. It has been our practice to wind the thermocouple wires around the steel wire, lash with fish line, and cement together with a gasket cement such as "gasoila." The "wishbone" also can be made of tubing and the thermocouple wires led down the center. To fill the core with cement, a vacuum is applied to one end of the tube. Of course, the location in which it is desired to place the thermocouple will determine the actual shape of the "wishbone."

A silver alloy, Elkonium No. 1, supplied by the P. R. Mallory Co., Indianapolis, Ind., has been found to be perfectly suited for the contacts. This material is furnished as contact points (Cat. No. S-142-L-1) and in strips ¼ x 6 x 0.030 in. The points are pressed into holes drilled at the ends of the bolts which go through the insulating block on the piston. A soft solder is used to "sweat" the slides to the heat-treated steel spring. All wire connections must be soldered thoroughly.

The steel springs are made to have a deflection of 0.010 in. with an 8 to 10-oz pull at the center of the sliding surface. It has been found that this rate gives a spring



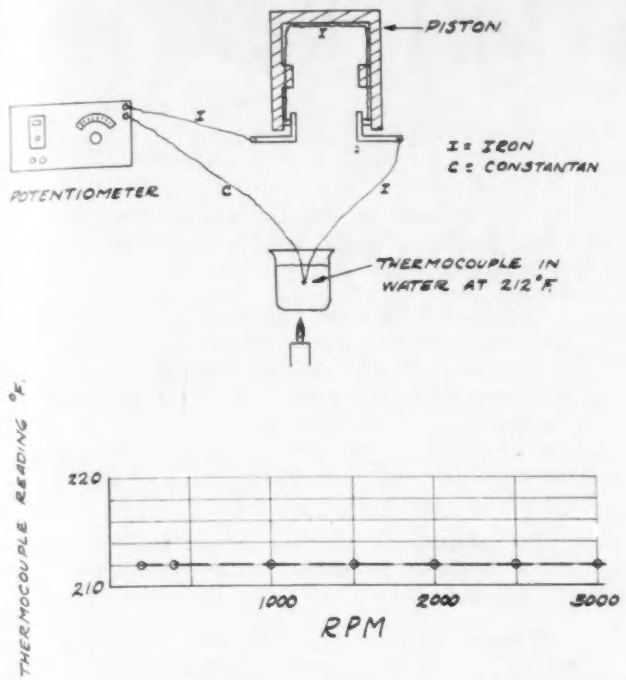
■ Fig. 17 – Final design of contacts for instrument for continuous measurement of piston temperatures

that has long contact life and that has no undesirable vibration period. Dimensions of the spring will depend on the particular application which may require a longer or shorter length.

For speeds up to 4000 rpm, a contact angle of 70-80 deg of crankshaft rotation has been used. In the case of our 3¾-in. stroke engine, this amounts to a sliding contact on the silver strip of about ⅜ in.

A production engine fitted with this set-up has been run 25,000 miles at 60 mph on a dynamometer with no attention or adjustments, thereby demonstrating the endurance and practicability of the device. Several laboratory test engines also have been equipped with the apparatus for investigating piston design up to 4000 rpm and for fuel study. A test car has shown that measurements can be made continuously on the road.

For laboratory work, a galvanometer having considerable sensitivity is desirable but, with care, the galvanometer which comes with the commercial potentiometer will be sufficiently accurate for automotive work. Galvanometers



■ Fig. 18 – Method for determining accuracy of readings

of the latter type are required for road work because of vibration.

By changing the design of contacts, it is entirely possible that temperatures of any rotating or reciprocating mechanism may be obtained. This would allow measurement of the temperatures of bearings, valves, and so on. A reciprocating motion is perhaps more desirable because of the slow motion at the ends of the stroke.

■ Test for Accuracy

During the first portions of the work, the question often arose as to the accuracy of the reading. While it could be shown that the potentiometer read a definite or certain temperature, say 300 F, it could not be shown that there was not an error caused by the contacts. To investigate this problem, the wiring of the circuit was changed as shown in Fig. 18. The iron wire is carried from the potentiometer through both contacts and to a beaker outside the engine. The constantan wire runs only from the beaker to the potentiometer. A liquid in the beaker is heated by a gas flame.

With this arrangement, the thermocouple is in a bath the temperature of which can be determined accurately. To determine the effect of having the two sliding contacts in the circuit, it is only necessary to run through the speed range and compare the potentiometer readings with the true bath temperatures. Fig. 18 shows the results obtained. The perfect agreement demonstrates that there is no error introduced by the mechanism.

■ Use of Common Return Wire

To reduce the number of contacts at the bottom of the piston skirt, it was proposed during the investigation that all the iron wires be fastened to one contact. This arrangement would give a common wire for all thermocouples, thereby requiring only 6 contacts instead of 10 for five locations. It was demonstrated that this procedure would lead to errors of as much as 25 F if the thermocouple junctions are in electrical contact with the part on which the temperature measurement is being taken. Under these conditions, the several thermocouples are in parallel and a true temperature is not taken. By insulating the junctions, correct readings can be obtained, but care must be observed not to produce another error by reducing the heat flow.

Settling Disputes by Integration

MAY I suggest the procedure of integration in contrast to arbitration or domination in handling differences or conflicts? The easiest way is by domination. This method requires the least ability and the least analysis from the point of view of good management. Many controversies are supposedly settled by domination – but are they settled? It is just another method of holding the lid on until the pressure becomes too much for the lid, or until conflict becomes chronic dissension.

A second way of handling conflict is by compromise. This procedure is so frequently used by management in dealing with men that the terminology needs no interpretation. To me, it savors very much of listening to two old cattlemen whom I once knew, trying to trade horses. Each was keenly aware of the strong points of his own and keenly interested in pointing out the weak points of the other man's horse. Both sides search for complaints to pile up, concessions to be traded and, the more concessions each has to trade, the more difficult it is for each side to find out what the other really wants. If a compromise is finally reached, each side has had to give up something that it really wanted. Differences settled in this manner have a tendency to return another day or in another form.

The third – and from the point of view of theory the best – method of settling conflict is by integration. M. P. Follett, in her application of psychology to problems of conflict, states: "I should like to ask you to agree for the moment to think of conflict as neither good nor bad; to think of it not as warfare, but as the appearance of difference – difference of opinions, of interest. For that is what conflict means – difference. As conflict – difference – is here in the world, as we cannot avoid it, we should, I think, use it instead of condemning it, we should set it to work for us."

This philosophy means that we do not have to accept the fact that differences cannot be worked out to the satisfaction of all parties concerned. Three managers may have different ways of approaching a problem. No way is probably the best way. A process of integration would mean such an exchange of ideas that the better parts of each plan would be accepted, and the result would be a new plan, different but better. This is improvement. It does not necessarily involve giving up something.

Excerpts from the paper: "The Growing Emphasis on Human Administration," by N. J. Aiken, director of Placement Bureau, State College of Washington, presented at the Pacific Coast Regional Meeting of the Society, Aug. 16 and 17, 1940.

WEAR-RESISTANT COATINGS

THE influence of chemical surfacing upon the running-in of diesel cylinder liners is discussed in this paper.

The newly assembled diesel engine requires a run-in before being placed into service operation. Because it is difficult to lubricate the smooth, honed liner bore during the period of structural change, "scratching" or "scuffing" is imminent. The problem of scuffing during the run-in may be largely eliminated through a caustic-sulfur treatment for the honed liners before assembly.

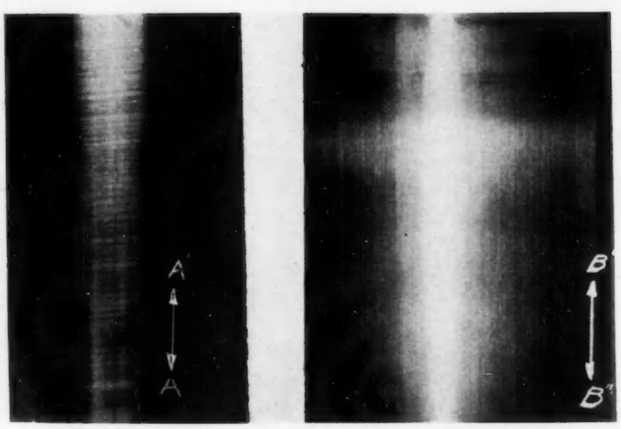
The nature of this coating, before and after the run-in, is shown by photomicrographs illustrating this paper.

Photomicrographs are also used to show the honed and run-in surfaces of the untreated liner and the surface of a liner "as scratched."

■ ■ ■

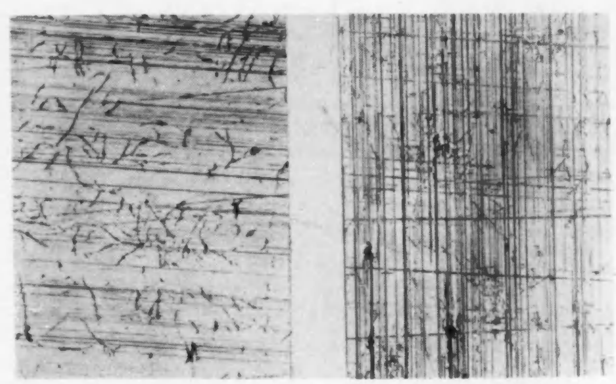
THE run-in given a diesel engine as a concluding phase of assembly is designed toward the achievement of several objectives. In a general way, the period of the run-in is an opportune one in which to complete certain final steps of engine production – those steps toward the promotion of equability in the ultimate product. Under the close watch and diligent instrumentation of skilled attendants during the run-in, the engine is inspected carefully to make certain that it has met all manufacturing specifications placed upon its production. Through the adjustment of occasional production abnormalities, as noted by the attendants, a nearer uniform product is assured. Specifically, the run-in implies operation of the engine through a prearranged time schedule of crankshaft speeds and horsepower in order to bring about a "mating" of all bearings. At any rate, all new bearings of the engine are not in a suitable state for satisfactory service operation until they, in the engine, have been given a preliminary run. The term "mating," in this sense, does not refer to any improvement or change in the dimensional fit between bearings, but does refer to an improvement in the compatibility between the contact surfaces of all bearing couples. It is intended that the term "bearing" will include all friction surfaces of the engine whether rolling, simple sliding, or reciprocating. One bearing surface moving in contact with a related bearing surface forms a bearing couple. Only through proper coordination and balance of the elements composing the operation schedule can the run-in be the most effective and most efficient. The rate at which the run-in may be conducted can be no greater than permitted by the most sensitive part of the engine. The total length of the run-in period, on the other hand, must be sufficient completely to run-in the part of the engine slowest to change. It must be kept clearly in mind that, at the present time, there are no specific units of measure with which to evaluate or express run-in results – that the value index of any run-in is determinable only by the relative indices associated with each objective for which the run-in was initiated.

A portion of the engine requiring a well-developed run-in is that including the piston, piston rings, and associated cylinder liner bore. In coordination with the run-in requirements of other parts of the engine, it is necessary that this run-in will cause the piston-piston ring-liner as-



A FINISH-HONED B RUN-IN

■ Fig. 1 – Bore of a 5¾ x 8-in. finish-honed cylinder liner before and after run-in



A FINISH-HONED 100X B RUN-IN 100X

■ Fig. 2 – Photomicrographs to show the changes produced by running-in an untreated cylinder liner

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 13, 1940.]

# of Diesel Cylinder Liners

by J. E. JACKSON
Caterpillar Tractor Co.

sembly to seal-in early and effectively and will conduct the liner and piston-ring bearing surfaces safely through the interval of the structural transformation.

The sealing-in of the piston-piston ring-liner assembly is necessary in order to preserve the initial working compression ratio of the engine from the earliest operation. Apart from the contribution to the future thermodynamic performance of the engine, other effects make it desirable to prevent gases from blowing past the piston rings to the crankcase. Lacquers formed by combination of hot oxygen in the blowby gases with lubricating oil in the ring-belt area may result in very undesirable ring-sticking[1]. Large, irregular flow of gases about the compression rings will cause them to become unstable in operation. This instability may lead the rings to rock in the ring grooves, with resultant poor bearing between the ring face and liner bore, or it may contribute to or cause ring oscillation in the ring groove to such extent that failure would result. In addition to the damage to ring faces by erosion, any large flow of blowby gases turbulently interferes with the oil-control functions of the oil rings. Inadequate lubrication through impaired spreading of the lubricating oil is very undesirable.

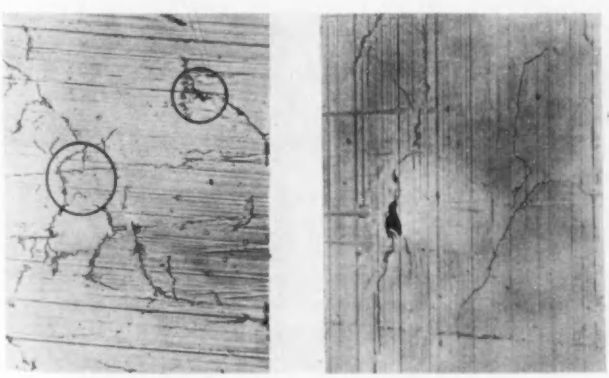
The run-in schedule must be designed so that the liner and piston-ring bearing areas will pass through their respective critical transformations without "scratching" and that there will be no further danger of them becoming "scratched" by future operation under normal working conditions. The critical transformation is a metallurgical one where the immediate rubbing surfaces experience certain important changes in physical properties[2]. When these changes are produced properly in the running-in period, the liner and rings are not only made "scratch-resistant" but also will be "surface-conditioned."[3] It is from this latter result of the run-in that much is contributed to the future wear performance of the piston, piston rings, and associated cylinder-liner bore.

That changes are wrought in the surface configuration of the cylinder-liner bore during the run-in may be seen by contrast of Fig. 1A with Fig. 1B. The former is a segment of the bore surface from a 5¾ x 8-in. cylinder liner after production finish-honing to 1½-3 micro-in. The roughness reading was taken along the surface parallel to the bore axis as indicated by the arrows $A' - A''$. A 5¾ x 8-in. production finish-honed liner with the same roughness and appearance as that of Fig. 1A was run-in through the 10-hr schedule in Table 1 as outlined for the engine using this size liner. The surface after break-in is shown in Fig. 1B with the direction of piston reciprocation indicated by the arrows $B' - B''$. Material composing the liner surface was displaced to allow the alteration in the orientation of the scratches on the surface from those in helical form, as produced by honing, to those arranged straight and axially from operation through the run-in.

Table I – Break-In Schedule for 5¾ x 8-In. Cylinder Liners

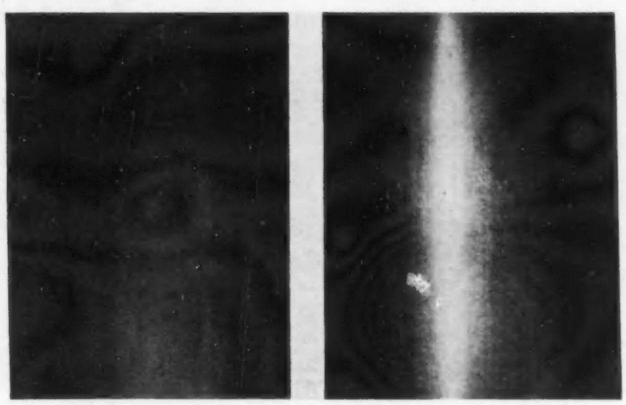
500 RPM Idle	1½ hr
700 RPM Idle	1 hr
20% Rated Load, 700 RPM	1 hr
50% Rated Load, 850 RPM	1 hr
75% Rated Load, 850 RPM	1 hr
100% Rated Load, 850 RPM	½ hr*
100% Rated Load, 850 RPM	4 hr**

* End of run-in for treated liners.
** End of run-in for non-treated liners.



A FINISH-HONED 500X B RUN-IN 500X

■ Fig. 3 – Photomicrographs to show the changes produced by running-in an untreated cylinder liner

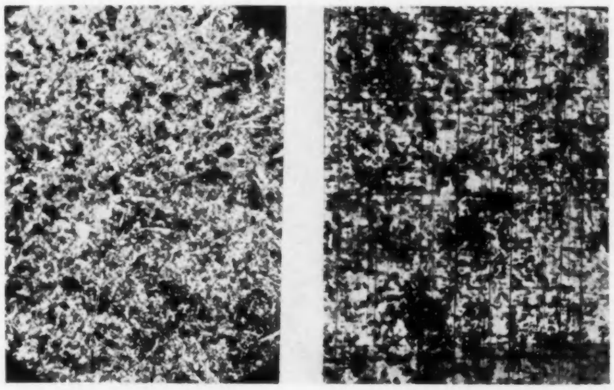


A CHEMICALLY SURFACED B RUN-IN

■ Fig. 4 – Bore of a 5¾ x 8-in., finish-honed and chemically surfaced cylinder liner before and after run-in

[1] See "Cylinder Lubrication of High-Speed Diesel Engines as Influenced by Temperature and Combustion Process," by A. T. McDonald and L. A. Blanc, presented at the Twentieth Annual Meeting of the American Petroleum Institute, November, 1939.
[2] See British Iron and Steel Institute Special Report No. 24: "An Electron Diffraction Study of Oxide Films on Iron," by R. Jackson and A. G. Quarrell.
[3] See "Cylinder Wear," a lecture by Prof. G. Ingle Finch at the Imperial College of Science and Technology.

Figs. 2A and 2B are in respective order from the same liners of Fig. 1 and are photomicrographs at 100X to show not only the changes in direction and character of the scratches in greater detail but also some of the changes in material arrangement surrounding the graphite flakes in



A BEFORE RUN-IN 100X B AFTER RUN-IN 100X

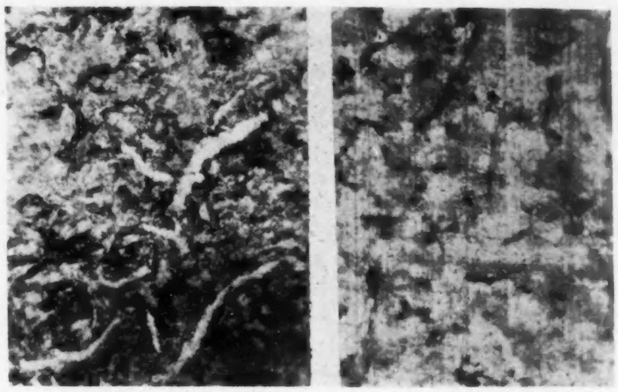
■ Fig. 5 – Photomicrographs to show changes produced by running-in a liner treated in the caustic-sulfur solution

the cast iron. The general, rugged surface configuration of the finish-honed liner is shown in the photomicrograph of Fig. 3A (from the same surface as Fig. 2A) with the black circles showing where the flow of metal by scratching is very noticeable. The honed surface as shown in Fig. 3A may be contrasted with the liner surface after run-in in Fig. 3B at 500X. The surface of Fig. 3B is from the liner of Fig. 2B which was run through the 10-hr schedule. It may be noted that many of the honing marks yet remain upon the surface. Since the engine was operated at full load and at rated speed throughout the last 4½ hr of the run-in, it is reasonable to conclude that the liner will pass safely through the next 4 hr which has been found by previous test to be necessary for removal of all honing scratches.

■ Scratches and Deformation

The scratches that are prominently noticeable upon the liner bore surface after honing and after run-in are directly associable with deformation of the surface metal beyond the yield point. Such deformation adjacent to each scratch fulfills basic requirements of work hardening. Since honing subjects the surface to deformation through metal removal as well as through flow by scratching, then it is conclusive that the honed bore surface is coated with a thin layer of work- or strain-hardened metal in random distribution. Many of the wear-resisting properties of the bore surface are derived from a coating of strain-hardened metal upon the surface; it is the formation of such metal coating, properly oriented in the direction of reciprocation, that constitutes "surface conditioning." By the same measure that this strain-hardened metal layer by run-in resists future wear, so does the strain-hardened metal layer by honing resist change of surface configuration by the run-in.

It has been stated that any piston-piston ring-liner assembly run-in must be coordinated with the run-in requirements of the remaining parts of the engine. For those liners with the "as finish-honed" bore surface the run-in period is so hazardous that such coordination is very difficult to attain. In the beginning, the surface of the liner is notoriously difficult to lubricate, for not only is the dynamic surface energy lowered by adsorbed gases and machine oils but also there are few regions on the surface to serve as capillaries. The honing scratches are very shallow in proportion to their width; therefore, they offer very little aid in the retention or spreading of lubricating oil. The ability of the graphitic carbon of the cast iron to aid in either holding or spreading lubricating oil over the new liner bore surface has been, in the writer's opinion, overestimated; this property of the graphite is not developed until after considerable engine operation, that is, the oil-retaining properties of the graphite do not appear until after the graphite has been fully exposed by the removal of the metal flowed over the flakes by honing and by running-in (Fig. 3B).



A BEFORE RUN-IN 500X B AFTER RUN-IN 500X

■ Fig. 6 – Photomicrographs to show changes produced by running-in a liner treated in the caustic-sulfur solution

Transformation of the liner bore surface begins soon after the start of the run-in – a time when poor distribution of the lubricant is least desired. When the metal forming the delicate bearing surfaces of the rings and liner is physically deformed into new orientations during the transformation, metal will be brought to the surface from out of the mass by the deformation. At the time when the molecules of this now newly exposed metal were deep in the mass, their fields were at equilibrium with the fields of surrounding metal molecules. Immediately upon their exposure to form the new surface, however, the molecules are not at equilibrium and have, by way of their unsatisfied fields of force, strong attraction for any other molecules brought into their proximity[4]. The force which caused the deformation was produced through the moving contact of the other member of the bearing couple. In the event that this other member also is deformed and active metal brought to its surface, seizure may occur while the two areas of active metal are in proximity of each other. To prevent such seizure, or to minimize its effects, an adequate supply of the lubricating agents should be present. As will be discussed, liners are chemically surfaced in order to assure positively the presence of these lubricating agents.

When any amount of metal at the surface is deformed into a new orientation, work must have been done upon

[4] See "Science of Petroleum," Vol. IV, 1938 edition, pp. 2566-2575: "Chemical and Physical Forces in Lubrication," by G. L. Clark, R. P. Sterrett, and Bert H. Lincoln.

the metal so deformed. When the amount of this work is large with respect to the deformed mass, and when this work is done at a rapid rate, the metal involved will experience a rapid rise in temperature. Unless the two bearing surfaces are sufficiently isolated by the lubricant, welding will occur. At any time the conduct of the run-in must be so gradual that seizure or welding will never become of such magnitude to cause "scratching" or "scuffing."

Many of the problems associated with the running-in of honed cylinder liners have been alleviated by giving the liners chemical treatment as a final operation prior to engine assembly. The liners, after finish-honing, are cleaned and then immersed in a concentrated water solution of sodium hydroxide and a small amount of sulfur. The process is one of etching and, literally, might be called "a controlled pitting process." Any free ferrite at the bore surface (and free ferrite should not be in any bore surface) is etched from the matrix. The carbide constituent of a pearlitic matrix is also reacted upon by the solution and fine lamellar pearlite is etched away leaving small pits filled with etching end-products.

Before any further discussion is given to the matter of chemical surfacing, it should be stated emphatically that the treatment is no cure-all, and that by use of chemical surfacing the choice of both lubricating oil and liner materials is subject to no less rigid specifications.

The 5¾ x 8-in. liner of Fig. 4A was 1½-3½ micro-in. after finish-honing and was 9-13 micro-in. after chemical treatment. Note that the shiny bore with the helical honing scratches has become a deep grey, almost black in color. The liner of Fig. 4B is the bore of a 5¾ x 8-in. liner also honed to 1½-3½ micro-in. and treated to 9-11 micro-in. and then given a run-in through the 6-hr schedule for the engine using this size treated liner (Table 1). Roughness of the bore after the run-in was 7-9½ micro-in. It may be noted that after the run-in the bore surface has not recovered to the bright shiny appearance which it showed before treatment.

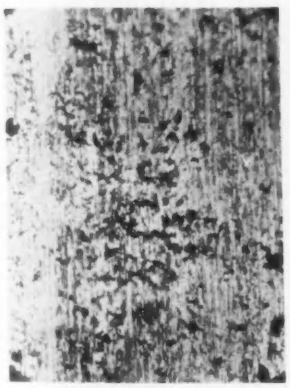
The dullness in the appearance of the treated bore after the run-in is attributed to the residual pits in the surface. These pits were too deep to be removed during the run-in. The photomicrograph of Fig. 5A shows (at 100X) that the surface becomes rough and pitted by treatment. Fig. 5B (100X) shows the residual pits on the surface after the run-in. The area of Fig. 6A (500X) is from or near the field of Fig. 5A and shows more clearly the ruggedness of the treated surface. Fig. 6B (500X) is from the same field as Fig. 5B and shows more clearly the shape of the residual pits and the nature of the scratches formed on the bore surface during the run-in.

The methods by which chemical treatment makes the liner bore surface more susceptible to safe run-in are: by removal of the undesirable components of surface composition; by deposition of certain chemical end-products upon the surface; and by change in the surface configuration.

The important surface metal to be removed by the etching is the strain-hardened metal formed on the surface by honing. When the layer by honing is removed by etching, then the running-in process, in so far as the transformation is concerned, begins with non-strained metal as it is exposed from under the etched coating.

■ Free Ferrite Undesirable

It is held that the most readily welded constituent of the cast iron of either piston rings or liner is free ferrite. For this reason, the presence of free ferrite in either ring or liner bearing surface is undesirable. However, cast iron is a heterogeneous mixture in which a small amount of free ferrite may exist. When the ferrite grains are small in size and amount, no harm in the way of scuffing will come from it during service operation. During the run-in, on the other hand, free ferrite in any amount adds to run-in difficulties. The removal of this free ferrite in the process of chemical surfacing is a definite aid toward the administration of a safe run-in.



■ Fig. 7 – The "furrow" of pits on the bore surface of a scuffed, untreated cylinder liner – 50X

■ Fig. 8 – Pits in the "furrow" formed when an untreated liner is scuffed – 200X

■ Fig. 9 – A pit 0.0004 in. deep formed when an untreated cylinder liner is scuffed – 500X

Although some honing methods are superior to others in regard to the stresses placed into the surface structure, each does transmit certain honing pressures and shearing stresses to the surface in order to remove metal. Under these metal-removing loads, certain of the polyhedral grains of the cast iron are either loosened or distorted in the matrix but not removed. After brief operation under the piston rings, these loosened grains become removed and serve to promote scuffing and scratching. These loosened

grains are among those certain undesirable components of surface composition to be removed by chemical treatment.

The coating deposited or formed upon the liner bore by etching in the caustic-sulfur solution serves in several capacities. The composition of the coating is ferrous oxide and ferrous sulfide tightly adhered to the unetched underlayer of iron. Not only does the matte surface of the sulfide-oxide coating facilitate rapid surface spreading of lubricating oil, but also lubricating oil is retained in the porous inner structure of the coating. This dual feature of retention and spreading of lubricating oil insures safety from seizure or welding during the early part of the run-in.

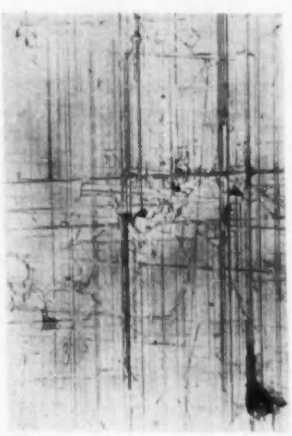
When the temperature of a small surface area is caused to increase rapidly through deformation, sulfur, from decomposed ferrous sulfide adjacent to the hot surface area, will aid the lubrication oil to prevent welding.

■ Oil Held in Pits

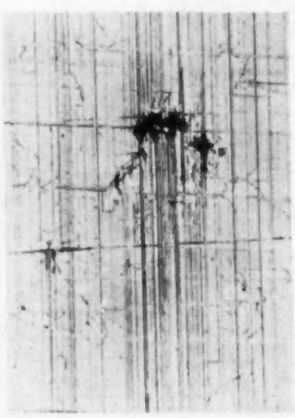
When free ferrite and small grains of fine pearlite are etched by the caustic sulfur solution, pits are formed which will be filled partially by the sulfide-oxide end products. These pits, with the porous sulfide-oxide contents, serve as reservoirs to hold a supply of lubricating oil to meet local surface needs during any period of boundary lubrication. The superficial sulfide-oxide coating is removed during the run-in, but the pits, on the other hand, last for many hours after the run-in (Figs. 5 and 6).

Through the aid of the various properties of the caustic-sulfur treatment the run-in is given ample opportunity to produce, by "surface-conditioning," the most favorable wear-resistant coating which the liner material is capable of forming. Truly, the surface-conditioned coating will experience wear – but at a very low rate. As a small amount of the surface is worn away, a new layer of the wear-resisting coating will be formed by the next reciprocations of the piston. With uniform chemical treatment and uniformity in the administration of the run-in schedule all engines will have the same operating expectancy, so far as liner wear and scuffing are concerned, for all will start their operation life from the same reference position.

For the engine with untreated liners, however, the future is not so bright. Because of the non-uniformity in the changes produced on the surface of each liner bore by the run-in, no assurance is held as to the liner's future performance. There is always the danger of scuffing.



■ Fig. 10 – "Scratching" on bore of untreated liner after a 10-hr run-in – 200X



■ Fig. 11 – This scratching could progress easily into scuffing on the bore of this untreated cylinder liner – 200X

As may be observed by the unaided eye, scuffing on the cylinder bore is seen to be a furrow in the surface extending in the direction of reciprocation. In contrast with the shiny areas of the bore, the scuffing is greyish-matte in appearance. Upon close visual examination in oblique light, the furrow is seen to contain a dense cluster of pits and deep scratches. Fig. 7 is a photomicrograph taken at 50X to show the arrangement of the pits in the furrow. It may be seen that the pits are quite large and close together, and their long axis does not necessarily extend in the direction of reciprocation. Fig. 8 (200X) shows more detail in regard to the size of the pits and the general disruption experienced by the surface. It may be seen that the general structures adjacent to the pits show considerable weakness with respect to their fastening in the matrix. The large pit shown at 500X in Fig. 9 is the same depth as the majority of those pits shown in Figs. 7 and 8, and was 0.0004 in. as measured by the vernier focusing scale of the microscope. These photomicrographs of Figs. 7, 8, and 9 were from an untreated liner which scuffed in a single-cylinder test engine operated at full load without a previous run-in.

Early stages of scuffing, when it should probably be termed "scratching," are shown in Figs. 10 and 11. This 5¾ x 8-in. liner was not treated and was operated through the 10-hr run-in. Note that the metal is flowed on the surface in both directions of reciprocation from the point where the particles were removed from the surface. The flow of the surface metal during the run-in is shown by the partial obliteration of many honing scratches.

The caustic-sulfur treatment process used to prevent scuffing or scratching of liners (as discussed in this paper) is the "Surfide Process" developed by the Standard Oil Co. of Calif.

DISCUSSION

Advantages of Faster Break-In

– A. L. Beall

Wright Aeronautical Corp.

IT is desired to suggest a qualification of the concept of surface conditioning as proposed by the author.

The paper discusses a tremendous amount of apparently useful work and is well supported by the experience cited.

It is suggested that the break-in schedule involving 2½ hr idle time may produce an absorbed film or glaze of the surfaces which it is difficult to remove with consequent reconciliation of the ring face and cylinder barrel surface.

In break-in of an engine, it is desired to remove a minute amount of material from the ring surface and burnish the cylinder bore as rapidly as possible to reduce blowby to a minimum. For this purpose, it would seem that a schedule which reaches a substantial increase of load in the first hour would be more effective than the long idle time. In other words, the work to be done in reconciling the surface should be completed as promptly as may be without scuffing the surfaces. It is possible that 20% or even 50% load could be achieved in the first hour with a better break-in and without scuffing.

Experience with several types of surface coatings indi-

(*Concluded on page 40*)

The C.U.E. Cooperative Universal Engine for Aviation Single-Cylinder Research

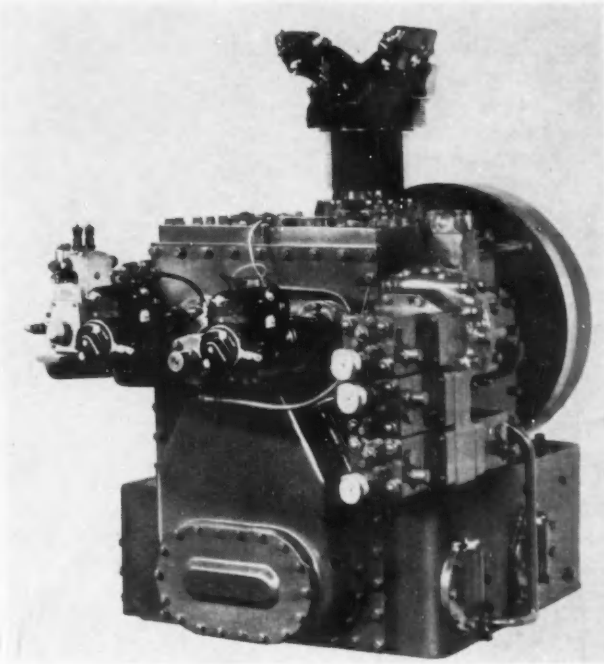
by A. W. POPE, JR.
Waukesha Motor Co.

COOPERATIVE METHOD – The C.U.E. engine design can be used to illustrate the effectiveness of the cooperative method. An appreciation of this method by the Society of Automotive Engineers and its members is far more important than the C.U.E. engine design itself. The formula for a cooperative project seems to be: first the recognition of a need; second, the desire to fill this need; and third, the willingness of a group to work together on it.

Some years ago the SAE Ignition Research Committee expressed its need for a standardized single-cylinder engine which would take actual full-scale aviation engine cylinders into the laboratory under accurately controlled operating conditions for aviation spark-plug research. The writer and his company wanted to build such an engine. The result was an engine displayed at the SAE White Sulphur Meeting in the spring of 1937. While it met the requirements for aviation spark-plug testing and was purchased on the spot by a spark-plug manufacturer, it did not fill the needs of the petroleum industry which was interested primarily in lubricating oil and fuel tests.

The interest, however, shown at this meeting definitely indicated the need for such an engine. There was a desire to build the engine. So the only missing link required to complete the formula was a group willing to work together. A volunteer group was organized at White

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 11, 1940.]



■ Fig. 1 – C.U.E. engine complete with Wright Cyclone cylinder

SOME years ago the SAE Ignition Research Committee expressed its need for a standardized single-cylinder engine which would take actual full-scale aviation engine cylinders into the laboratory under accurately controlled operating conditions for aviation spark-plug research. Such an engine was built by the author's company and displayed at the SAE Semi-Annual Meeting in 1937. Although this engine did meet the needs of the spark-plug manufacturers, it did not fill the needs of the petroleum industry, which was interested primarily in lubricating oil and fuel tests.

A volunteer group, organized at the 1937 SAE Semi-Annual Meeting, first listed the features which such a design must include, and then a small design subcommittee was formed to study and coordinate the requirements. Five complete designs were made before a combination was developed which appeared to meet all requirements in a satisfactory manner.

The single-cylinder crankcase design was laid out to provide for the Wright Cyclone air-cooled cylinder, $6\frac{1}{8} \times 6\frac{7}{8}$ in. bore and stroke, the largest in general use at the time.

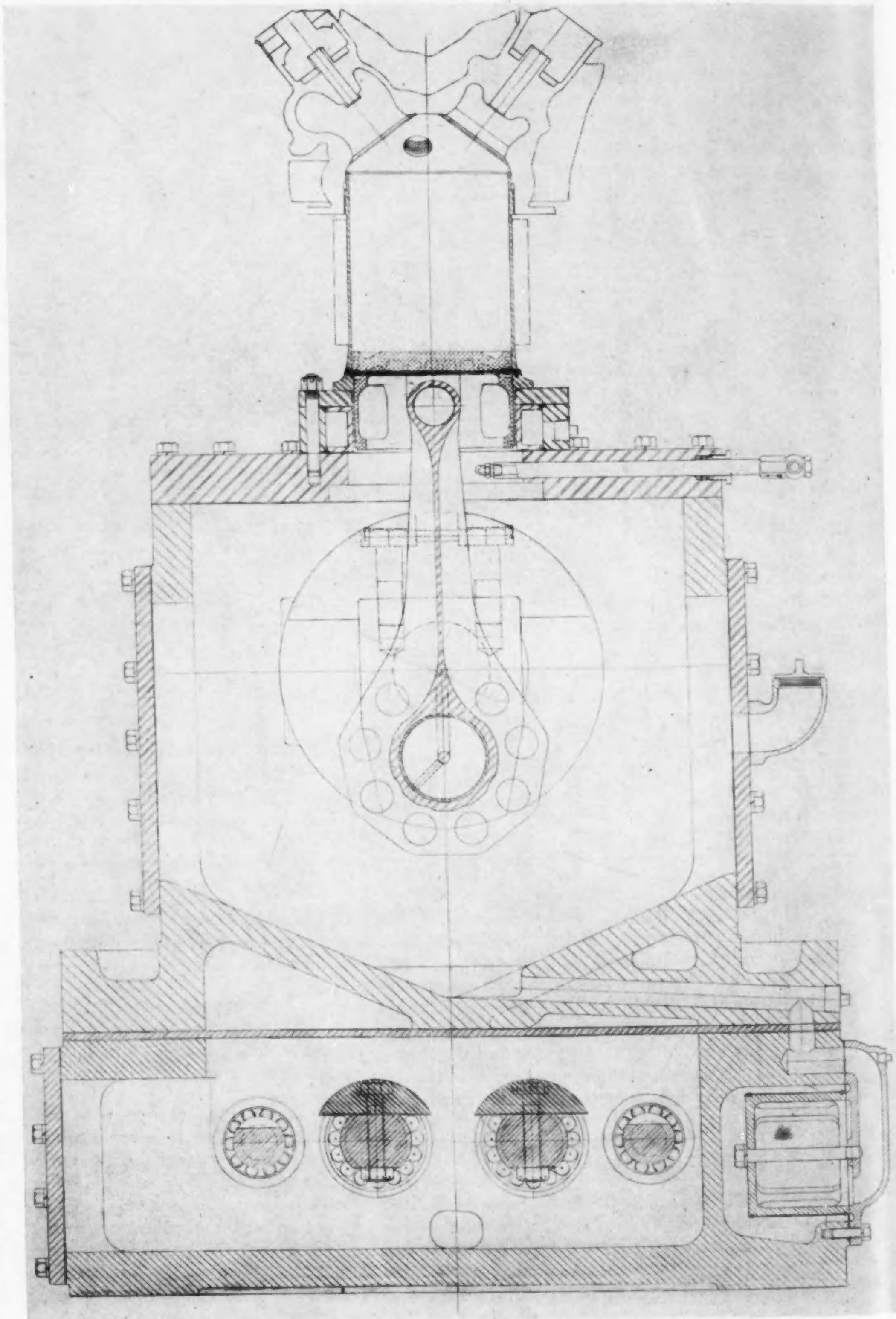
A detailed description of the C.U.E. engine design and construction comprises the major part of Mr. Pope's presentation, covering such features as: the balancing system; main roller bearings; smooth-walled crank chamber; lubrication; crankcase construction; temperature control; crankcase seal; accessory drives; crankshaft; connecting rod; speed; spark indicator; flywheel; reverse rotation; dynamometer coupling; and fuel injection system.

Sulphur, and the first meeting went a long way toward defining the problem by listing the features which the design must include.

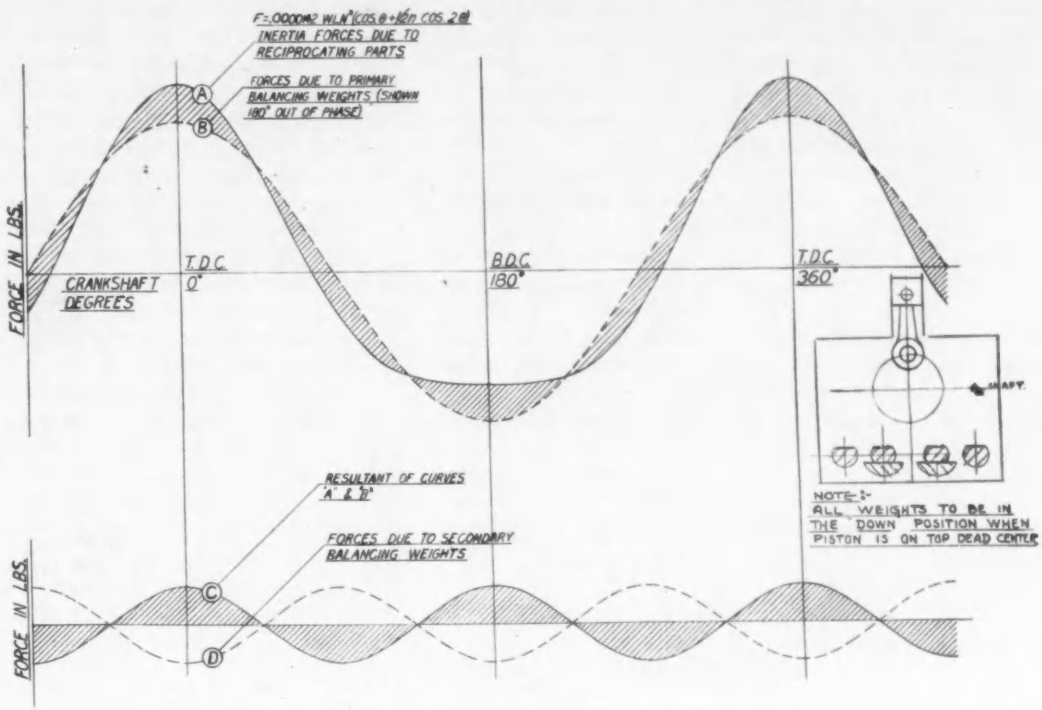
A small design subcommittee was formed to study and coordinate the many miscellaneous requirements. Counting the original spark-plug test engine, five complete designs were drawn up and submitted to the group for consideration before a combination was developed which appeared to meet all requirements in a satisfactory manner. Three months of intensive design work were required.

Size of Engine – Since the Wright Cyclone air-cooled cylinder was the largest in general use at the time and a great deal of research work was under way on this cylinder, the single-cylinder crankcase design was laid out large enough to provide for this cylinder which is a 6⅛-in. bore and 6⅞-in. stroke, giving a displacement of 202 cu in. The complete engine as it is being built today is shown in Fig. 1 with Wright Cyclone cylinder installed. The weight is 4900 lb.

Balancing System – A basic feature of any satisfactory single-cylinder high-speed laboratory engine is the balancing mechanism. The reciprocating-type counter-weight construction which gives theoretically ideal balance and was entirely satisfactory for a spark-plug test engine was unsuited for lubricating-oil tests because of the mechanism in the crank chamber which made it difficult to control the amount of crank-chamber splash and also seriously complicated the problem of cleaning out the case thoroughly between oil tests. The Lanchester-type balancing system adopted for the C.U.E. engine leaves the crank chamber clean and smooth since the mechanism is contained in a sub-base entirely separate and independent from



■ Fig. 2 – Cross-section of front end of C.U.E. engine showing balancing weight positions



■ Fig. 3 – **Balancing curves for C.U.E. engine**

the main engine lubricating system. Two sets of rotary weights are provided, one running at engine speed to balance the primary reciprocating forces and the other pair of weights running at twice engine speed to balance the secondary unbalanced forces. The crankpin and connecting-rod big end rotating mass is balanced by weights on the crankshaft proper. Fig. 2 is a section from the front end and showing balancing weight positions.

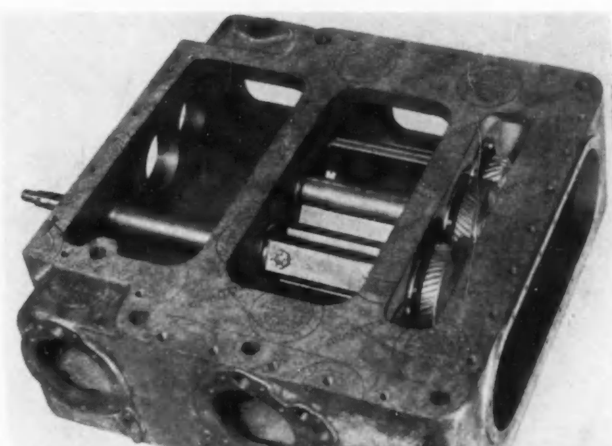
The primary balancing weights are bolted to their shaft and thus can be changed to meet the requirements of different piston weights. The secondary balancing weights are machined from the solid.

Timing of the balancing system is provided for by means of a multiple-hole vernier-type coupling located on the front of the drive shaft.

The balancing mechanism is driven from the crankshaft by a six-strand roller-type chain. An idler sprocket is provided for tension adjustment, and provision is made for mounting the adjustable sprocket on either side of the engine center to permit operation of the engine in either direction.

Fig. 3 is a diagram illustrating the unbalanced forces and the forces created by the balancing mechanism to cancel them. It will be noted that the symmetrical sine wave representing the balancing force set up by the primary weights rotating at engine speed is exceeded by the reciprocating piston weight at top dead-center and exceeds the reciprocating forces at bottom dead-center. There is also an unbalanced component in the 90-deg range. These secondary unbalanced forces occurring four times per revolution are illustrated in the lower part of the figure. They are balanced by the pair of small rotating weights operating at twice crankshaft speed. This analysis is based on the assumption that the connecting-rod and piston assembly can be segregated into reciprocating and rotating masses. While this system is only an approximate balancing method, for practical purposes it is quite satisfactory and provides a very smooth-running single-cylinder engine at high speeds.

Fig. 4 is a view of the balancing weights and housing with the engine removed. Fig. 5 shows a side section through the engine and balancing system. The complete isolation of the balancing mechanism from the engine proper should be noted, and a positive seal at the crankshaft extension opening prevents any interchange of oil at this point. The constant oil level in the sump is below the weights but oil thrower bolts and discs dip and thus provide an oil mist for lubrication of the roller bearings, gears, and six-strand roller drive chain and idler. SAE 40 oil is



■ **Fig. 4 – Balancing weights and housing – C.U.E. engine**

supplied through the filler on the left-hand side of the balancer case (when the engine is viewed from the front) and maintained at the correct level indicated by the pipe plug opening at the rear left corner 3¼ in. from the lower edge. As the balancer weights require accurately timed relation with the crankshaft, an adjustable coupling is provided at the front end of the drive shaft. Vernier bolt-hole arrangement permits positive timing means. Tension on the roller drive chain is maintained by an adjustable idler on the slack side.

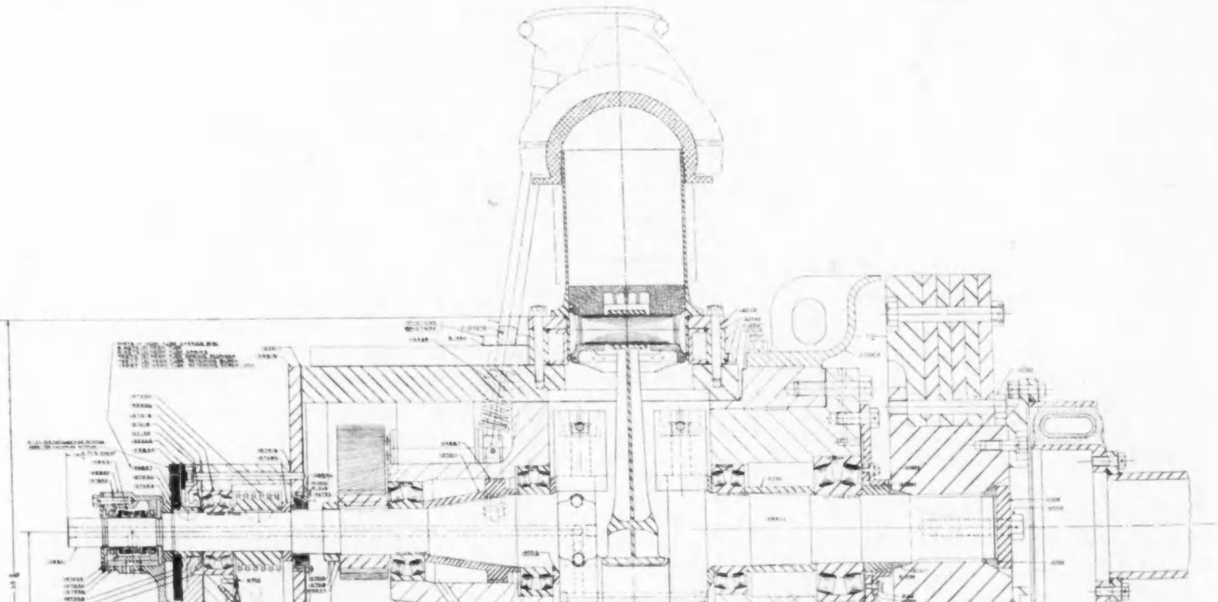
Main Roller Bearings – A basic requirement for lubricating-oil tests is positive control of the oil supply to the main crank chamber. This condition practically dictated the use of roller crankshaft bearings instead of plain bearings which would require pressure lubrication with uncontrolled oil throw-off. Roller bearings operate with no lubrication other than the oily atmosphere of the crank chamber. Fig. 5 illustrates the generous double-roller self-aligning bearing dimensions and the widely spaced double bearings on each end of the crankshaft. This construction successfully eliminates the typical single-cylinder crankshaft whip which sometimes occurs with single bearings.

Smooth-Walled Crank Chamber – Ability to clean out the crankcase thoroughly between oil tests is important, and to make this possible the crank chamber is made with sloping walls giving good drainage and with smooth surface. The pockets between roller bearings and between the rollers themselves can be flushed out with solvent oils and with air pressure through drilled channels provided for this purpose.

Lubrication – Accurate control of lubricating-oil supply to the engine required three separate supply systems in addition to the constant-level splash system in the balancer base. Fig. 6 illustrates diagrammatically the oil circuits. Each one is equipped with an individual gear-type pressure pump. In addition to the three pressure pumps, a fourth scavenge pump is required. All pumps are driven from an auxiliary half-speed gear train on the right-hand side of the engine (when viewing the engine from the front).

This layout permits complete flexibility in measuring, filtering, cooling, or controlling the oil in any desired manner. Individual laboratories provide their own oil conditioning and control equipment.

The crankpin oil circuit enters the front end of the crankshaft through a gland, follows a drilled passage to the crank cheek where a cross drilling takes the oil to a sediment trap bolted on the side of the cheek, then back to the crankpin bearing hole which is located 45 deg on



■ Fig. 5 – Side section through C.U.E. engine and balancing system

the leading side. The sediment trap which is accessible through the crankcase side door makes a convenient means of measuring sludge formation.

Oil Spray Circuit

An oil spray circuit is provided to regulate the amount of splash oil in suspension in the crankcase. This circuit connects to a jet in the crankcase top deck. This jet can be adjusted as to position and size and is removable from outside the engine.

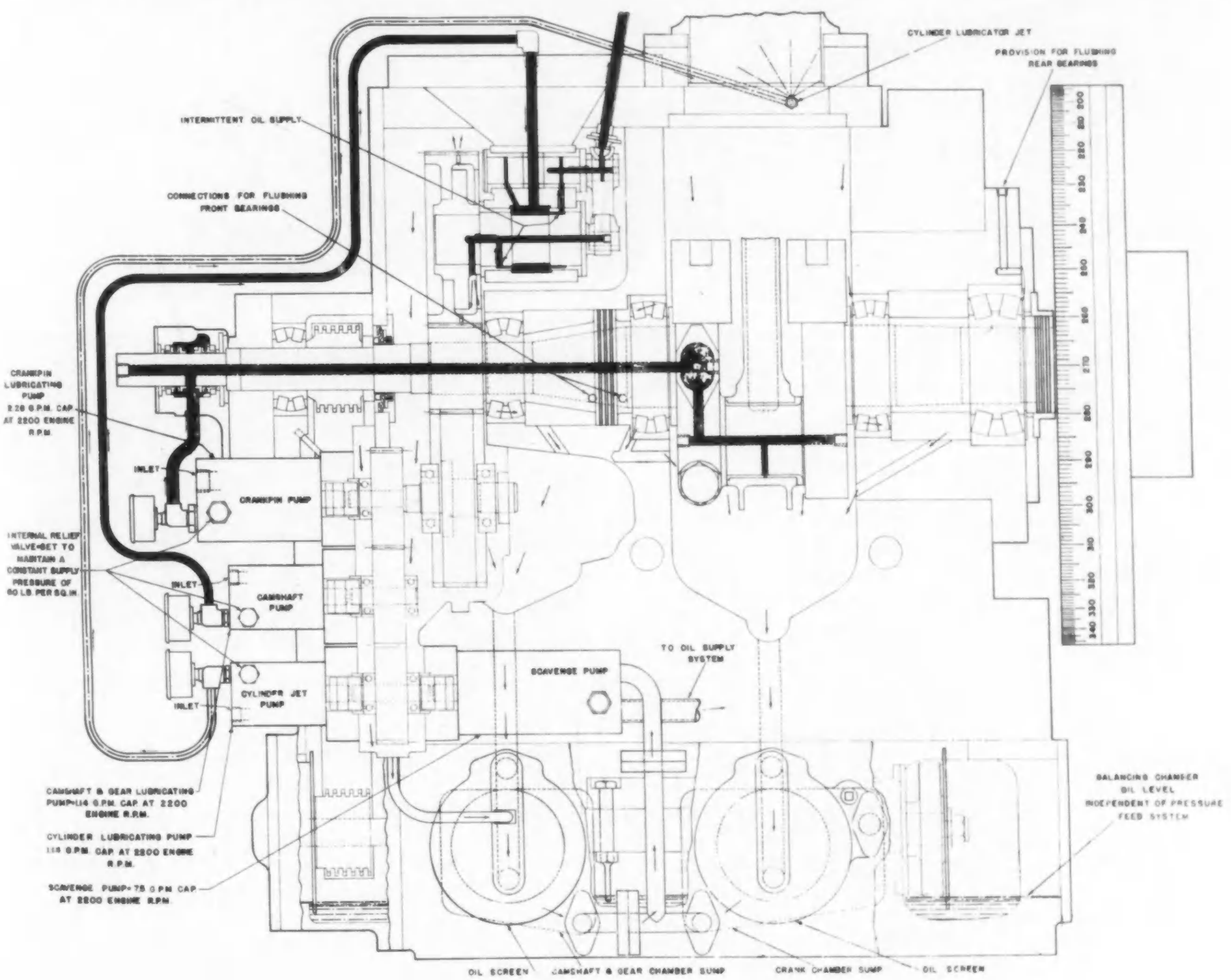
A third oil circuit leads to the timing-gear housing where intermittent oil feed is controlled by registration of grooves and holes in the camshaft as indicated, one lead going to the timing gear and another lead going to the rocker arm and valve push rods. The timing-gear oil can be kept entirely separate from the crank-chamber oil since a piston-ring type of seal is provided between the two front main bearings which positively separates the gear chamber from the crank.

The scavenge pump can be connected to draw oil from either or both of two sumps located in the sub-base. One of these collects oil from the crank chamber and the other from the gear chamber. Cylindrical screens attached to the sump covers are accessible for cleaning. If it is desired to keep the timing-gear oil separate from the crank-chamber oil, the pipe connecting these sumps is disconnected. The main scavenge pump then draws oil from the crank chamber, and the gear chamber can be scavenged by its pressure pump itself. If a scavenge pump is required for this circuit, it is possible to provide a double-type scavenge pump. When operating with the two sumps connected to a common scavenge pump, it is, of course, necessary to balance the gas pressure in the crank chamber and gear chamber by a connection above the oil level in order to prevent accumulation in one chamber or the other.

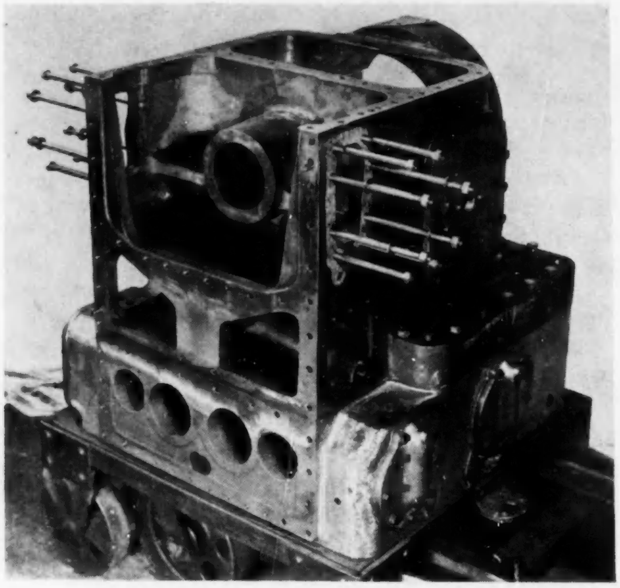
Universal Crankcase Construction – An important requirement of the design was its adaptability to a wide range of cylinder sizes and types to make it a truly universal engine. This was accomplished by making the top of the crankcase open and suspending the camshaft and gears from the crankcase cover. Fig. 7 shows a view of the basic box-type crankcase structure with the top plate removed. This box construction with $1\frac{1}{2}$-in. cast sections makes an extremely rigid structure, and the side plates give convenient accessibility to the crank chamber. The open top gives complete freedom of design for everything above the crankshaft without disturbing the main structure.

The crankcase 2-in. steel top plate provides great latitude in adopting cylinder types to this crankcase.

Temperature Control – Temperature control of the cylin-



■ Fig. 6 – Lubricating oil system – C.U.E. engine

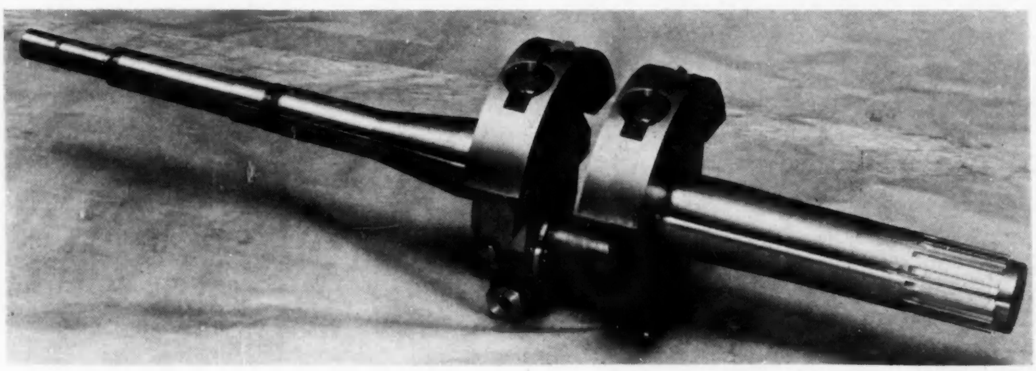


■ Fig. 7 – C.U.E. engine crankcase with the top plate removed

the two-piece crankshaft construction. This arrangement permits operation with the standard aircraft engine connecting-rod big-end bearing. However, unless there is some specific purpose in using the actual aircraft-engine connecting-rod bearing, it is much more convenient to have a connecting rod with split big end, and a connecting rod has been designed and is available as extra equipment. Fig. 9 shows the I-beam section rod. An interesting feature is the greatly improved stress distribution and bearing stiffness obtained by turning the I-section 90 deg to the conventional. This arrangement also makes a very simple machining problem and also provides convenient bolt head locating means between flanges. Precision-type copper-lead bearings are provided.

Speed – While the normal operating speed with the Wright cylinder is approximately 2300 rpm, the crankshaft and balancing system should operate for a reasonable life at 3000 rpm. Higher speeds can, of course, be used but the bearing life will be reduced. There is little doubt, however, that connecting-rod bearing life will be the controlling factor on permissible speed rather than the crankshaft bearings.

Spark Indicator – A neon-type spark-position indicator is mounted on the crankshaft front extension and is of double construction to indicate independently the spark position for each magneto.



■ Fig. 8 – C.U.E. engine two-piece crankshaft with clamped crankpin

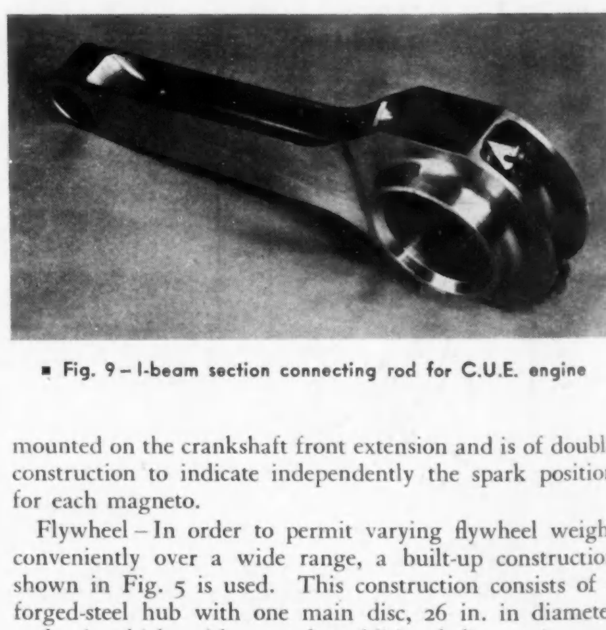
der mounting flange and the crank chamber is provided for by steam jacket space in the cylinder flange and the crankcase base. Holes $1\frac{1}{2}$ in. in diameter extending from side to side of the engine in the lower part of the crank chamber are provided for electric cartridge-type heaters.

Crankcase Seal – Piston blowby measurement can be made accurately since the crank chamber is positively sealed with piston-ring-type joints. A positively driven poppet-type breather valve is provided and can be timed to control the crank-chamber pressure or, if desired, the engine can be operated with an atmospheric vent.

Accessory Drives – Adequate accessory drives are required for laboratory use, and this engine is equipped with four one-half speed drives suitable for magneto or injection pumps, a fifth tachometer drive running at one-half engine speed, and a crankshaft front extension for indicator drive operating at engine speed.

Crankshaft – Crankshafts can be provided either of a one-piece construction or of two-piece construction with the typical Wright clamped crankpin design, which has proved to be simple and reliable, Fig. 8. The correct tension on the clamp bolt is determined by measuring its stretch, 0.007 to 0.009 in. stretch giving adequate clamping effect. Nitralloy steel is used to provide hard wearing surfaces comparable with aircraft construction.

Connecting Rod – The standard one-piece radial-engine connecting rod can be used in this engine as the result of



■ Fig. 9 – I-beam section connecting rod for C.U.E. engine

Flywheel – In order to permit varying flywheel weight conveniently over a wide range, a built-up construction shown in Fig. 5 is used. This construction consists of a forged-steel hub with one main disc, 26 in. in diameter and 2 in. thick, with space for additional discs to increase the weight if desired. The periphery of the flywheel is graduated at 1-deg intervals for convenience in engine timing. The weight of the main disc is 210 lb, and it has a

radius of gyration of 10.4 in. The ¾-in. auxiliary discs weigh 92.6 lb and have a radius of gyration of 9.9 in.

The forged-steel flywheel hub is located on the crankshaft end by a 3½-in. diameter 16A spline. Double taper aviation-type clamp construction keeps the hug tight on the shaft. The tapered clamping rings are pulled into place by three small bolts and a plate instead of the customary single large nut. This construction facilitates clamping and locking.

Reverse Rotation – It is desirable for the crankshaft to rotate in relatively the same direction as in the actual aircraft engine. It, therefore, is necessary to make provision for rotation in either direction. Standard rotation is considered to be anti-clockwise when viewed from the timing-gear end of the engine. Reverse rotation is clockwise when viewed from the timing-gear end. The following changes are necessary to provide for reverse rotation:

A. Inlet and Exhaust Camshafts: Inlet and exhaust camshafts must be re-timed in accordance with reverse rotation.

B. Balancing Weight Drive-Chain Idler Location: The balancer chain idler must be moved from the left-side location to the right side and, at the same time, the balancer driveshaft must be shifted from the right side to the left-hand side. When making this shift, the balancer weights must be re-timed carefully so that all four weights are down when the crankshaft is at top dead-center.

C. Oil Pump Reversal: All four oil pumps must have their inlet and outlet connections reversed and the pressure-regulating valve in the pump body shifted to the opposite side.

D. Crank-Chamber Relief-Valve Timing: If the crank-chamber relief valve is to be used, its cam should be re-timed to open at approximately 70 deg ATDC.

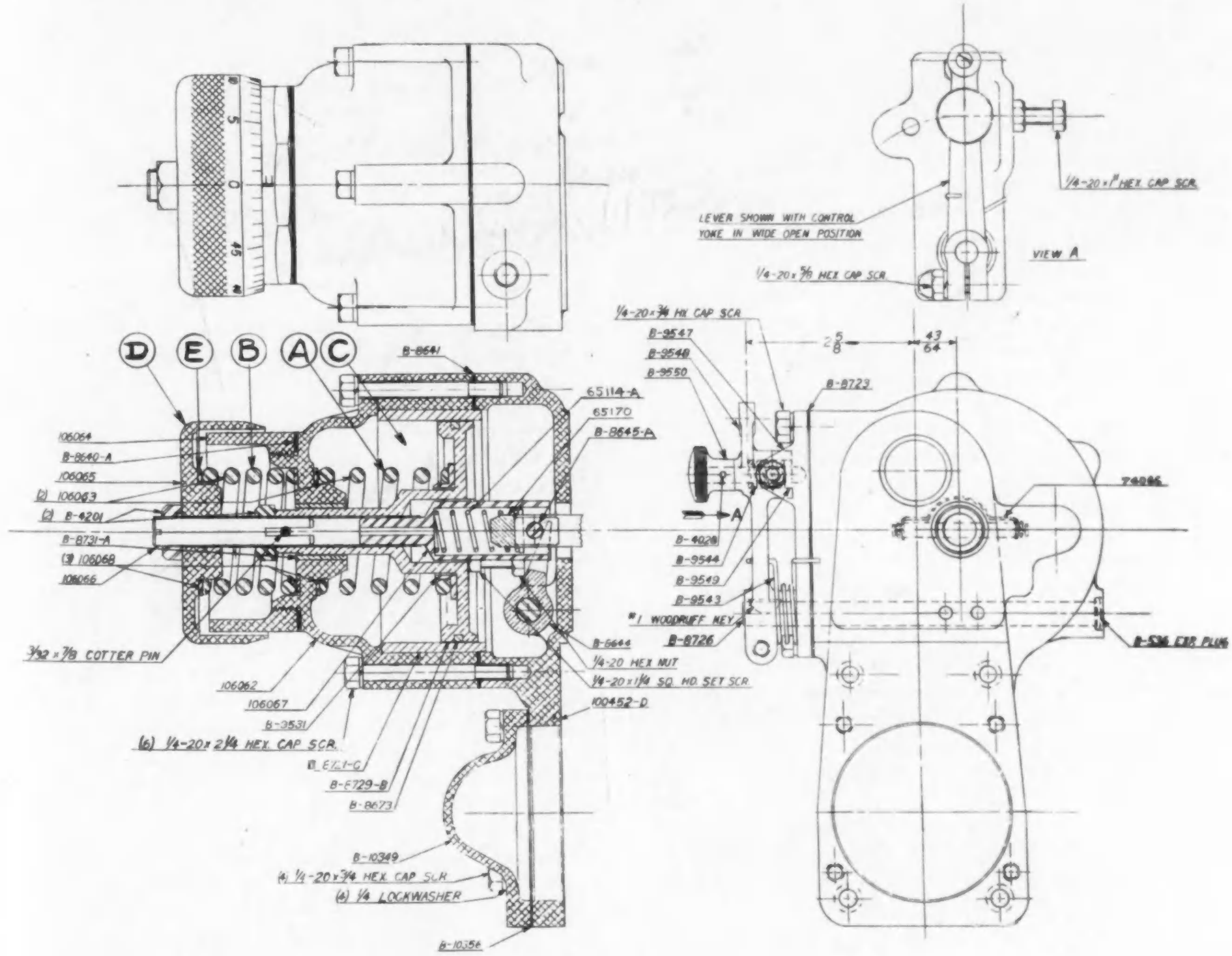
E. Magneto Rotation: Magneto breaker mechanism must be reset for reverse rotation.

F. Spark Indicator Dials: Neon-tube spark indicator disc should be shifted 60 deg in a clockwise direction. The spark indicator dials will then read in reverse order.

G. Tachometer Drive Rotation: Tachometer head for reverse rotation (clockwise when viewing the tachometer driveshaft on the engine) will be required.

H. Flywheel Marking: Flywheel timing marking will be in reverse order.

I. Crankpin Oil Hole: The oil hole in the crankpin is located 45 deg on the leading side. When the engine rotation is reversed, this hole will be 45 deg on the trailing side. For ordinary operation this may not be harmful but, to obtain best lubricating conditions, the hole probably should be relocated. The nitralloy shaft can be



■ Fig. 10 – Pneumatic control system for maintaining a constant fuel-air ratio for supercharging

drilled if the thin hard case is ground away enough to permit starting the drill.

Dynamometer Coupling – While no special dynamometer coupling has been designed for this engine, this item is extremely important for a successful laboratory set-up. Experience to date indicates the most successful coupling for this purpose to be the Falk air-flex type of pneumatic coupling illustrated in Fig. 5. The principle of this coupling involves the use of an inflated rubber ring of tire construction vulcanized between the driving and driven coupling members. Air pressures of 80 to 100 lb per sq in. are used, and the typical strength, flexibility, and long life experienced with pneumatic tires as compared with solid rubber tires seems to be obtained with this coupling as compared with couplings using solid-rubber cushion members.

Fuel Injection System – While the first demand for this engine appears to have been spark-plug and lubricating-oil service, it seems probable that its ultimate and most useful purpose will be that of high-output fuel research. When the design requirements for spark-plug and lubricating-oil tests had been met, the engine was equally well suited for fuel testing. Supercharging is of course required in all tests and, since single-cylinder carburetor operation is erratic and a carburetor is difficult to balance, manifold injection was considered the most practical means of fuel supply. A Bosch B size 2-cyl injection pump with special camshaft having both cams in the same position and with special 11-mm plungers will deliver 100 lb of gasoline per hr at 2300 rpm engine speed. The two plungers are, of course, connected to a single injection line. A special Bosch low-pressure nozzle is located in the induction pipe to inject fuel against the air stream. Injection occurs during the intake stroke, starting shortly after top dead-center. Further development is needed to indicate the optimum injection timing and spray characteristics. This low-pressure nozzle materially reduces the load on the injection pump as compared with operation with the conventional high-pressure pintle-type injector.

With gasoline injection there is some tendency to leakage past the pump plungers with resulting dilution of the oil in the pump sump. For this reason the Bosch Co. requires a circulating oil system for the pump if gasoline is to be used. A small gear-type circulating pump is driven from the end of the injection pump camshaft to circulate lubricating oil from outside sump to the injection-pump cam chamber.

In order to maintain conveniently a constant air-fuel ratio, a pneumatic control system has been designed for application to the standard Bosch pump. This system is illustrated in Fig. 10. Induction line air pressure applied to one side of the control piston regulates the pump rack position in proportion to the induction air pressure. This control maintains approximately constant air-fuel ratio from sub-atmosphere up to any desired degree of supercharging. The spring rate, of course, must be designed to match the engine cylinder and injection-pump plunger size. Adjustment of the air-fuel ratio at any intermediate position is obtained by turning the large graduated knob on the end of the pneumatic control.

■ Conclusions

The fact that 15 C.U.E. engines have been built and 15 more are under construction demonstrates the effectiveness of the cooperative method in stimulating and executing cooperative research.

The growth of engineering and science seems to have reached a phase approaching maturity which requires coordinated and cooperative effort if it is to continue and be effective. The life history of a science can be paralleled to that of man. First, the individualist phase of youth; second, the family phase developing the relationship between individuals; and third, the social phase of maturity requiring cooperation with society in order to live effectively. Science went through its individualist stage in the periods of such men as Newton, Faraday, and Pasteur. Its second or family phase occurred during the industrial period typified by corporation groups like families developing group units. We are now entering the phase of maturity requiring the cooperative method which gears individual efforts together and makes them effective.

Since the formula for a successful cooperative project appears to be the discovery of a need, the desire to fill it, and finally the willingness of a group to work together, may it be suggested that the primary purpose and object of the SAE is to provide the means of originating, stimulating and executing cooperative projects?

Discussion of Wear-Resistant Coatings

(*Concluded from page 32*)

cates that they delay break-in of the surfaces and, while they may be helpful in preventing scuffing, they do require an excessive time for run-in.

Augments Author's Discussion

– G. L. Neely

Standard Oil Co. of Calif.

PROBABLY the most significant feature of Mr. Jackson's discussion was in regard to the deleterious effect of cold working and deformation of liner surfaces in the final machining and honing operations, which often cause the grains of cast iron to become loosened or distorted without being removed from the surface. Our experience shows that this is a major cause of scuffing during break-in and that the Surfide treatment either removes these particles or renders them incapable of causing scuffing.

This treatment removes free ferrite and certain other undesirable constituents in the metal surface, but it does not attack the graphite particles. It is evident, therefore, that the graphite is exposed by the treatment and the beneficial effects of this material are immediately available. In addition, the treated surface is coated with a sulfide film which prevents welding and promotes spreading of the oil film. The surface pits produced by the action of the chemicals serve as reservoirs for the oil.

This process is well past the experimental stage, as shown by the fact that several of the large engine and engine parts manufacturing companies have been licensed to use the process and one of these, the Caterpillar Tractor Co., has used the process for more than three years. One of the most recent applications is the treatment of used piston rings in large-bore gas engines by the Natural Gasoline Department of the Standard Oil Co. of Calif. The use of new treated rings has proved so beneficial in reducing scuffing, blowby, and lacquer formation in these engines, that a Surfiding plant was installed to treat the used rings at the annual overhaul period.

OIL COOLING –
Its Relation to Bearing Life

by R. A. WATSON
Service Engineer, Federal-Mogul Corp.

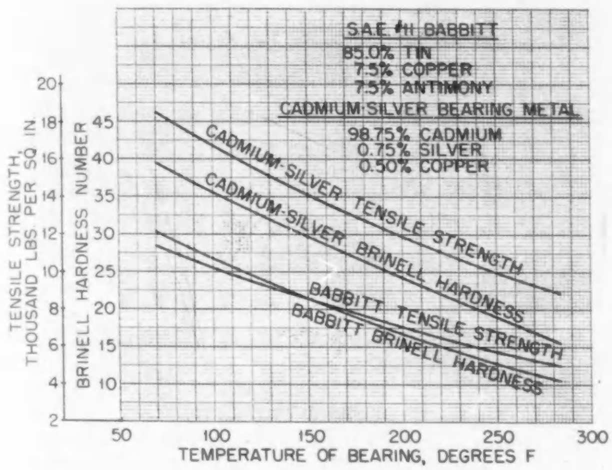
IN his discussion of oil cooling and its relation to bearing life the author attempts to answer four questions: (1) Why temperature affects bearing life; (2) What the actual bearing temperatures are in service; (3) What factors cause high oil and bearing temperatures to be generated; and (4) How to reduce these temperatures most conveniently and effectively.

The best method of cooling the oil, he finds, is by the use of radiators in the air blast. Heat exchangers utilizing the engine cooling water, he believes, are not practical for engines where oil temperature control to 180 F maximum is desired. Also, jacketing the oil pan or employing cooling-water coils in the oil pan are not considered feasible methods.

Pointing out that all authorities on automotive lubrication recognize that a lubricating oil temperature of 180 F gives optimum overall engine performance, he concludes that it is possible to maintain the lubricating oil temperature at or below this value in a large automotive diesel engine. In his discussion of oil coolers he brings out their influence on bearing life, on cylinder lubrication, and on effective life of lubricant.

IN the past three years premature bearing failures have been given more thought than ever before by manufacturers and operators, both of whom have become more critical of performance as the performance of other engine parts has risen, and because the duty imposed on the bearings has been steadily increasing.

Believing that it is not sufficient to accept the condition that oil temperatures are now running higher than ever before and thus probably contributing to bearing failures,



■ Fig. 1 – Effect of temperature on strength of cadmium-silver and babbitt bearing materials

we have studied the matter in the light of the following questions:

1. Why temperature affects bearing life;
2. What the actual bearing temperatures are in service;
3. What factors cause high oil and bearing temperatures to be generated; and
4. How to reduce these bearing temperatures most conveniently and effectively.

Of the most practical interest to the operator is the finding that the best method of cooling oil is by the radiators in the air blast, while heat exchangers utilizing the engine cooling water are not practical for engines where oil control to 180 F maximum is desired. Similarly, it is not feasible to jacket the oil pan or to employ cooling water coils in

[This paper was presented at a meeting of the Northern California Section of the Society, San Francisco, Calif., April 9, 1940.]

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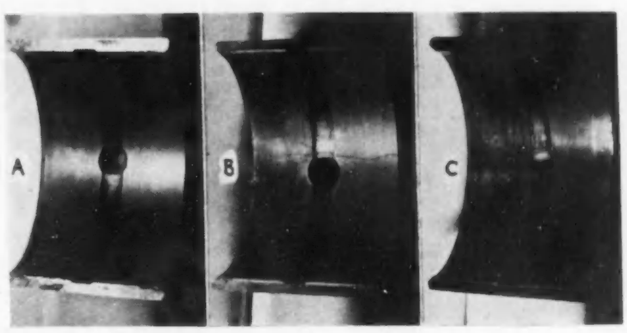


■ Fig. 2 – Progressive bearing failure due to high localized temperature

Table 1—Relative Deterioration of Diesel-Engine Lubricants when Used in 4-Cyl., 4¼-In. Bore Engine With and Without An Oil Cooler

	Without Cooler	With Cooler	Without Cooler	With Cooler	Without Cooler	With Cooler	Without Cooler	With Cooler
Oil Hours	60	56½	61½	60	60	60	60	60
Engine Hours	66	62¾	127½	245	242½	365	307½	425
Gravity, API at 60 F	17.5	18.7	17.9	19.1	17.5	19.3	17.8	19.2
Viscosity, S.U. at 100 F, sec	978	880	1063	723	1113	703	1172	693
Viscosity, S.U. at 130 F, sec	333	309	353	262	383	258	380	255
Viscosity, S.U. at 210 F, sec	64	64	65	59	69	59	68	59
Insoluble, mg/gr	12.49	5.55	17.47	0.97	5.93	0.98	114.72	1.01
KOH Abs., mg/gr (Acidity)	6.0	3.62	8.20	2.43	5.46	2.43	6.65	4.05
Ash, %	0.15	0.17	0.19	0.16	0.14	0.17	0.22	0.17

the oil pan. Sufficient engineering data are available to make it possible to predict what the effect of these coolers will be on a large transport engine.

It is concluded that it is quite possible to maintain the lubricating oil temperature in a large automotive diesel engine at or below 180 F, at which temperature, because of the relatively high strength of the bearing metal, much longer bearing life may be expected.

In endeavoring to point out the advantages accruing due to the use of an oil cooler in connection with the lubrication of internal-combustion engines, it immediately becomes evident that the subject is more in the light of a discussion of the evils of high-temperature operation than of the merits of an oil cooler.

The function of a properly selected oil cooler, that is, one of the correct type and design to fit the particular job involved, is that of obtaining and maintaining the most effective operating temperatures for an engine without danger of reaching temperatures which may bring about structural changes in the metal parts that result in a lessening of their ultimate strength, as well as that of preserving those properties of a lubricant which play a large part in a crankcase oil's effective life.

To illustrate the foregoing, the subject may be divided into three parts:

1. The influence of the use of oil coolers on bearing life.

2. The influence of the use of oil coolers on cylinder lubrication.

3. The influence of the use of oil coolers on the effective life of a lubricant.

In order to illustrate Item No. 1, it is necessary to call attention in Fig. 1 to the structural weakness occurring when bearing metals are subjected to high temperatures. This effect is less with some bearing metals and greater with others but, in all cases, there is a definite, measurable, falling off of strength with increased temperature.

Fig. 2 (*A*, *B*, and *C*) illustrates a bearing failure occurring due to high localized temperature. In *A* no crack is discernible, although there is a bright shiny spot evident. *B* has developed a microscopic crack, and *C* is close to the point of failure. This occurred 122 hr after the bright spot was first noted. The total operating time was exactly 196 hr. The engine was operated in an atmosphere of 125 F without an oil cooler at full load.

Contrast this result with that from an engine operated under the same conditions but equipped with an oil cooler. Fig. 3 shows bearing shells from this test after 500 hr of operation. There can be no doubt that the use of the cooler greatly increased the life of the bearings.

With respect to cylinder lubrication, there are two primary factors governing the formation of deleterious materials – oxidation and temperature. Little can be done about controlling the supply of oxygen available for reaction, but much has been done, and very much more can be done, about the control of temperature to reduce oxidation since the rate of oxidation under the same atmospheric conditions increases rapidly with increased temperature.

By this is not meant oxidation of lubricants to sludges by means of various laboratory test methods, but rather the reactions occurring in an engine which contribute to carbon and lacquer formation. Fig. 4 is an illustration showing the effect of the use of an oil cooler on an engine operating under severe conditions of temperature (125 F atmosphere) compared with the same engine operating without the aid of a cooler. The piston at the left was operated with, and the one at the right without the cooler. Little need be said about this illustration except that the overall condition of the engine exhibited the same improvement in the case of the engine operated with a cooler as does the piston shown in the illustration.

To illustrate the effect of an oil cooler on oil deterioration, Table 1 is offered which shows the analyses of crankcase drains from two new engines operated under full-load conditions of an atmospheric temperature of 125 F using the same lubricant. Duration of test was 500 hr. Oil was drained every 60 hr.

The insoluble content given in this table is a measure of the solid material only and does not include materials soluble in the oil. A glance at the data shows in every case an appreciable reduction in deterioration of the lubricant in every respect.

1. Investigation of bearing temperatures obtained in the field revealed that maximum temperatures of sufficient magnitude are reached to cause a serious loss in tensile strength of the bearing metal.

2. The laboratory investigation revealed the factor having the greatest effect on bearing temperature to be the lubricating oil temperature and that, through its control, bearing temperatures could be reduced.

3. It is the opinion of many authorities on automotive engine bearing design and lubrication that the ideal crankcase lubricating oil temperature is about 180 F. With the lubricating oil maintained at this temperature in a diesel engine, the tensile strength of the bearing metal is from 20 to 40% greater than that obtained under some conditions of trucking operation. Bearing life with the lubricating oil controlled at 180 F should be at least 20 to 40% greater, with an improvement of several-fold in some cases

not seeming unreasonable when viewed in the light of the short bearing life obtained during hot-weather operation.

4. Several methods of cooling the lubricating oil were studied:

a. It is estimated that the minimum useful size for a lubricating oil cooler using the engine cooling water is one in which the product of the area and the heat-transfer factor is 1700 Btu per hr, per deg F. Such a cooler would probably be too large to be acceptable. Even with this size, the oil can be cooled only to a temperature of 25 F above the cooling water temperature. The data collected do not indicate this type of heat exchanger to be practical.

b. Similarly it is impractical to jacket the crankcase or to use water coils in the oil pan because the low heat-transfer factor of such arrangements requires use of about 85 sq ft of heat-transfer surface to reach within 25 F of water temperature.

c. The field tests on a tractor-type cooler (using air to cool the lubricating oil) indicated it to be successful.

Control of Bearing Temperature

Introduction – Engine bearing life as low as 10 miles and as high as 150,000 miles has been experienced by present-day truck operators. The factors causing this range in bearing life are many and varied. These factors change both with respect to the magnitude of their effect and with respect to the ease with which they may be studied and evaluated. Some factors at the present time are, without doubt, unknown.

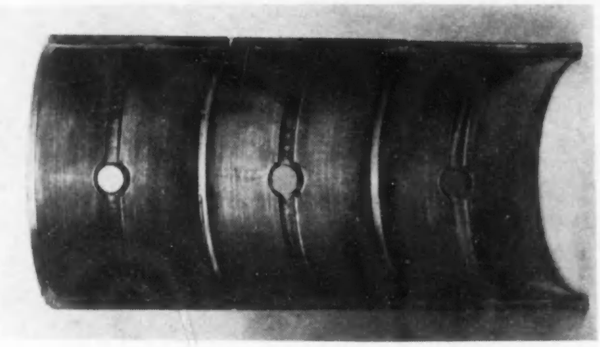


Fig. 3 – Bearing shells from engine equipped with an oil cooler after 500 hr of operation

It is apparent that with an established engine design, with technically controlled maintenance, and with operating conditions accounted for, bearing life depends upon the lubrication practice. It is the purpose of this paper to present the results of a study of those factors related to the lubricating oil that affect bearing life.

In order to limit those variables not directly related to the lubricating oil, it was desirable to study one make of engine which was widely used in the trucking industry in order that the maximum amount of data could be collected. As diesel engines, as a class, are subject to greater shock and maximum loads on the bearings, the diesel type of engine was selected for testing.

With the selection of the engine made, the investigation was divided into field surveys and laboratory experiments. In the field, operators' experience was called upon to direct the investigation to the hardest and most varied routes traveled, and here comprehensive temperature, load, and speed surveys were conducted on the engines under road operation. As evaluation of certain factors under road operation was difficult due to the impossibility of maintaining constant engine conditions and due to the inconvenience caused the operator, it was readily apparent that study on an engine installed in the laboratory was necessary. The laboratory investigation therefore was directed by the findings in the field; that is, the loads, speeds, and temperatures and the manner in which these items varied in hard field operation were reproduced by the engine testing in the laboratory.

Bearing Metal Properties

Before embarking on a program of engine testing, it is important to obtain a clear picture of the properties and behavior of bearing metals, to know how temperature and load affect their life, and why.

The life of a bearing in an engine is dependent upon the following properties of the bearing metal: "fatigue strength; ability to resist seizure of the steel shaft during periods when there is metal-to-metal contact; good bonding characteristics; mechanical compressive strength; conformability; embeddability; and corrosion resistance."

"Fatigue strength usually is the most important property of a bearing material as it determines the allowable load and life in the majority of cases. It is recognized generally that alterations of load will cause fatigue cracks which may surround and undermine areas. Loss of sufficient area so greatly reduces the load-carrying ability that failure occurs."

A fatigue failure is the result of repeated or reversed stresses at intensities of stress much below that required to break the metal in tension under a steady load but above a value termed the "endurance limit." Many metals show true endurance limits, that is, limiting stresses below which the metal will escape breakage indefinitely. The endurance limits of common non-ferrous metals range from about 20% to 50% of their tensile strength. As the magnitude of the stress increases beyond the endurance limit, the number of repetitions of the stress (cycles) to cause failure decreases very rapidly as an exponential of the stress.

The tensile strength of a mteal is dependent upon its temperature. Fig. 1 presents the variation of tensile strength and Brinell hardness with temperature of cadmium-silver and SAE 11 babbitt bearing metals. Referring to this

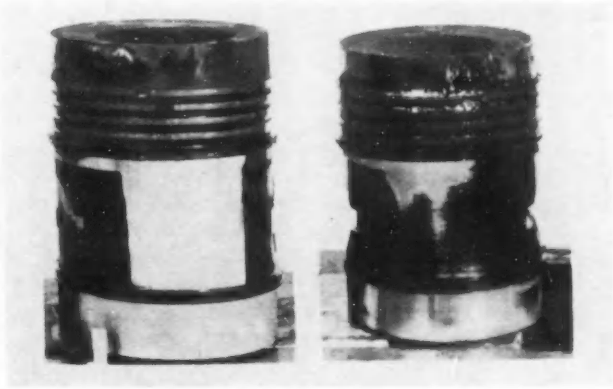


Fig. 4 – Effect of use of oil cooler on engine operating under severe temperature conditions – Piston at left from engine operated with oil cooler; piston at right from engine operated without oil cooler

Table 2 - Diesel Engine Main Bearing Temperatures

Gross Load, 36,400 lb - Lubricant, SAE 30 Grade - Atmospheric Temperature, 35-40 F

Place	Oil Sump, F	Oil Pressure, lb per sq in.	Center Main Brg., F	Front Main Brg., F	RPM
Checking Station	185	50	181	160	1600
Inkom	200	50	188	160	1650
McCammon	215	40	198	160	1650
McCammon	220	30	200	160	1600
Virginia	240	25	225	160	1600
Logan	250	25	210	150	1600
Start Sardine Canyon	220	30	200	160	1650
Top Sardine Canyon	260	15	225	180	(152 H.P.C. 1650 Wide Open)

drawing it is seen that, at a temperature of 250 F, the tensile strength of the stronger metal, cadmium-silver is 10,000 lb per sq in., at 200 F, the tensile strength is 12,000 lb per sq in., or a 20% increase for a 50 F reduction in bearing temperature. With an increase in tensile strength, the load on the bearing remaining the same, the resistance to fatigue failure increases; that is, the useful life of the bearing increases. If the strength becomes great enough that the endurance limit is not reached, the bearing will never fail from fatigue.

■ Field Investigation

As pointed out previously, the majority of bearing failures in the field are the result of fatigue failure of the bearing metal. The subsequent discussion pointed out that the fatigue resistance of the bearing metal was dependent upon the ratio of the tensile strength to the stress to which the metal is subjected. The tensile strength was shown to be dependent on bearing temperature. And finally, it was concluded that bearing life was dependent upon the temperature of the bearing metal. The primary object, therefore, of the field investigation was to determine whether bearing temperatures became high enough to cause a serious reduction in bearing metal strength. Simultaneously with the primary investigation, the effect of other factors related to bearing failure were to be studied.

Bearing temperatures were measured by means of thermocouples soldered in the lower halves of the main crankshaft bearings, the thermocouple junctions being within 1/64 in. of the crankshaft journal, and the thermocouples were located one-quarter of the bearing width from the edge of the bearing.

Table 2 presents the maximum main bearing temperature obtained on a run from Pocatello, Idaho to Ogden, Utah.

It should be pointed out that some of the figures in this chart may not be entirely correct in so far as actual maximum temperatures are concerned. It was not the purpose of this test to show actual maximum temperatures, but instead to show actual changes in temperatures. It is generally accepted that bearing temperatures are 35 to 50 F higher than oil sump temperatures but, due to the method of assembling the thermocouple and its location, some of the bearing temperatures shown are lower than the oil temperatures.

It is seen that a maximum temperature of 225 F was reached in the center main bearing. From Fig. 1 it is found that the bearing metal strength at that temperature was 10,900 lb per sq in. This is about a 41% reduction in bearing strength from that existing at room temperature. This means that the maximum load to which the bearing can be subjected repeatedly without a fatigue failure has been reduced by about 41%. Usually the bearings are subjected to the greatest stress when the bearing temperature is the highest. It therefore appears entirely reasonable that the combination of the decreasing magnitude of the bearing metal endurance limit and the increasing stress to which the bearing is subjected as the engine load increases, results in the operating stress of the bearing exceeding the endurance limit.

A temperature survey made on a diesel-engine powered truck running between San Francisco, Calif., and Pocatello, Idaho, showed a maximum temperature of 240 F, representing a reduction in bearing strength of about 40%. During these tests other factors were observed which affected bearing life. Of particular importance is the condition of the big end of the connecting rod. The bore should be perfectly round and free from taper, and the bolts tight to prevent shifting of the connecting-rod cap. In the case of rods which are slightly out of round, undersized bearing shells should be fitted and fly cut in place to the desired size, locating the center of the hole from the center of the wristpin bearing. Also of importance is the speed at which the engine is operated. This particular engine should be driven within a speed range of about 1500 to 1800 rpm with a maximum speed of 1600 rpm when the engine is used as a brake.

■ Laboratory Investigation

In the field tests it was established that bearing temperatures of sufficient magnitude were reached to cause serious reduction in bearing strength. The next phase of the investigation was to determine the factors governing bearing temperature and the methods of controlling these factors. Such an investigation requires careful control of the engine conditions in order to obtain significant results. This control could not be obtained in the field without seriously affecting operating schedules and, in general, inconveniencing the operator. A diesel engine therefore was installed in the laboratory to perform the second phase of this investigation.

The possible factors affecting bearing temperature are: engine load, engine speed, lubricating-oil temperature, cooling-water temperature, lubricating-oil viscosity, and lubricating-oil flow rate.

Lubricating-Oil Temperature

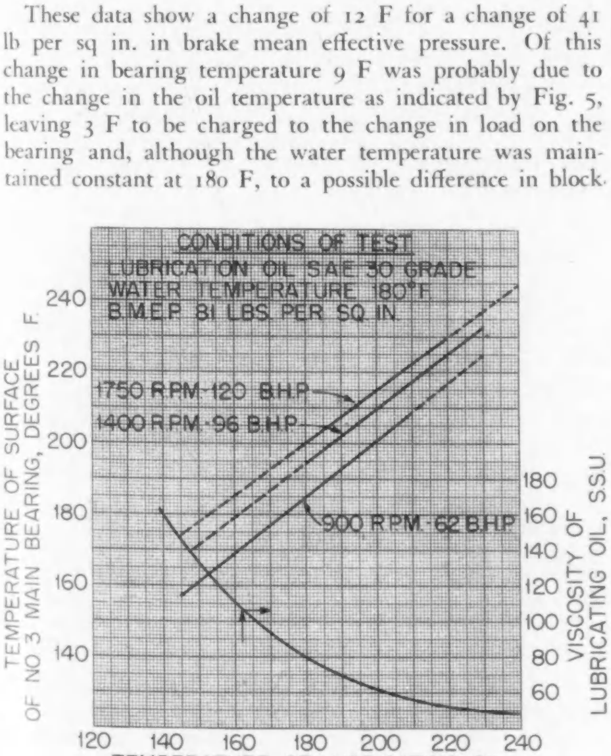
Of the factors mentioned in the preceding paragraph, the effect of the temperature of the lubricating oil was found to be the greatest. This is reasonable when it is considered that the oil is in direct contact with the bearing surface and in the best position to extract heat from the bearing surface, providing the temperature of the oil is below the bearing-surface temperature and the rate of flow of the lubricant is great enough (which it is proved to be by the data obtained). On Fig. 5 is plotted the temperature of No. 3 main bearing against the temperature of the lubricating oil. (The temperature of the bearing was measured in the same manner as in the field investigation.) From this drawing it is seen that, with the engine operating at 1750 rpm and 120 bhp, a reduction in lubricating oil temperature from 240 F to 180 F caused a 44 F reduction in bearing temperature. This reduction in bearing temperature represents a 1700 lb per sq in. increase in tensile strength of the bearing metal.

■ Engine Power

With increased power output from an engine, the bearings are subjected to greater loads. With greater loads, the frictional energy generated by the bearings is greater and bearing temperatures should be treated at the increased power. To check on this a test was made at 1400 rpm to determine the effect of load on bearing temperature with the following results:

Rpm	Bmep, lb per sq in.	Bhp	Oil Temperature, F	Bearing Temperature, F
1400	81	96.2	223	216
1400	40	47.5	211	204

These data show a change of 12 F for a change of 41 lb per sq in. in brake mean effective pressure. Of this change in bearing temperature 9 F was probably due to the change in the oil temperature as indicated by Fig. 5, leaving 3 F to be charged to the change in load on the bearing and, although the water temperature was maintained constant at 180 F, to a possible difference in block temperature as a result of the change in the combustion-chamber temperature. It therefore appears that load at constant speed has small effect on bearing temperature.

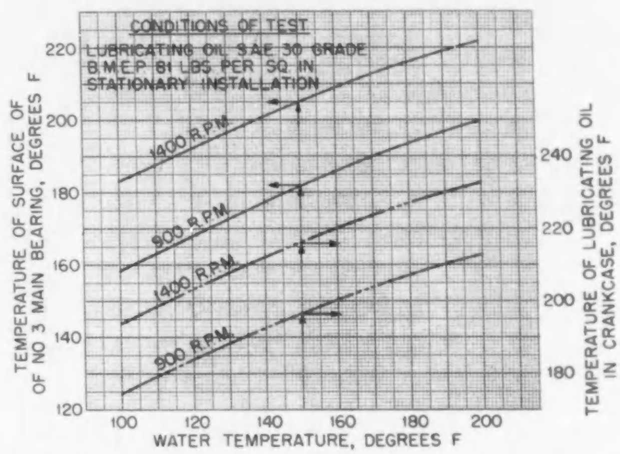


■ Fig. 5 – Effect of oil temperature on bearing temperature (diesel engine)

This observation in no way decries the importance of load in its effect on bearing life, but shows simply that a change in load alone will not change bearing *temperature* to any important extent.

■ Engine Speed

As a journal rotates in a bearing, a wedge-shaped film of oil is formed between the journal and the bearing. One side of the film adheres to the bearing and is therefore stationary. The other side of the film adheres to the journal and has a velocity equal to the surface speed of the journal. There is therefore a shearing action in the oil film supporting the journal in the bearing. This shearing action generates heat. The greater the shaft speed, the greater the quantity of the heat generated and, consequently, the greater is the temperature of the bearing. To determine the magnitude of this effect, bearing temperatures were determined at engine speeds of 1400 and 1700 rpm with a constant bmep of 81 lb per sq in.



■ Fig. 6 – Effect of water temperature on bearing temperature (diesel engine)

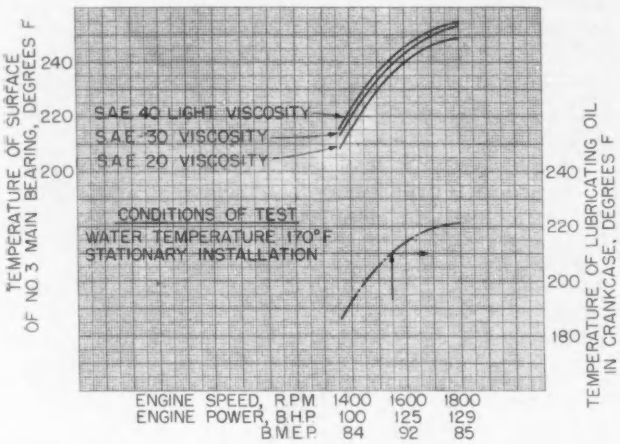
Engine Speed, rpm	Oil Temperature, F	Bearing Temperature, F
1400	223	216
1700	237	236

In this run the oil temperature was not held constant although all other variables were. The bearing temperature change must therefore be corrected for the change in oil temperature. This correction is 10 F which is deducted from the data on Fig. 5. After applying the correction the change in bearing temperature for a 300-rpm change in engine speed is 10 F or about 3 F per 100 rpm.

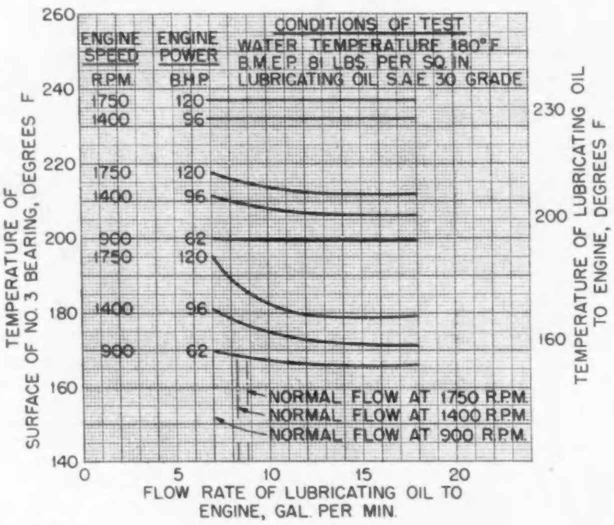
It therefore appears that the principal effect of a change in power output and speed is to change the temperature of the lubricating oil which, in turn, determines the temperature of the bearing surface.

■ Cooling-Water Temperature

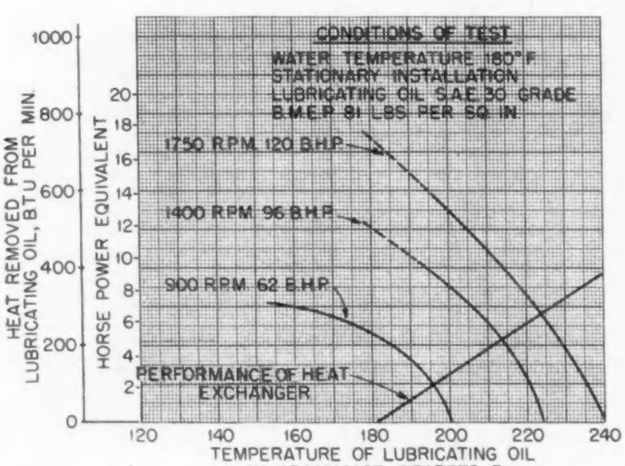
The temperature of the engine block is largely dependent upon the temperature of the cooling water. The temperature of the upper portion of the crankcase and the amount



■ Fig. 7 – Effect of oil viscosity on bearing temperature (diesel engine)



■ Fig. 8 – Effect of oil flow on bearing temperature (diesel engine)



■ Fig. 9 – Relation of oil temperature to heat removed from oil (diesel engine)

of heat conducted from the main bearings are therefore affected by the water temperature.

Fig. 6 presents the measured effect on bearing temperature and oil temperature of a change in water temperature. It is apparent that the effect is that of a built-in oil cooler. Lowering the water temperature from 200 F to 160 F reduces the oil temperature 13 F. This decrease in lubricating oil temperature will produce an 8.5 F reduction in bearing temperature according to the relation shown on Fig. 5. Subtracting this value from the 13 F gross change in bearing temperature leaves 4.5 F chargeable to the direct heat transfer from the bearing to the cooler engine block.

■ Viscosity of Lubricating Oil

As previously stated, the film of oil between the bearing and the shell is in a state of rapid shearing action which produces heat. As the viscosity of the film increases, the shearing speed remaining constant, the amount of heat generated increases and, conversely, a reduction in viscosity reduces the generation of heat.

The data of Fig. 7 indicate the effect on bearing temperature of a change to one lower SAE viscosity grade to be a reduction in temperature of about 4 F. This change in bearing temperature indicated as the result of using a lighter viscosity grade lubricant helps but is far from sufficient to control bearing temperature adequately.

■ Rate of Flow of Oil

The rate of heat removal from a bearing by the lubricating oil depends upon the temperature difference between the oil and the bearing and the rate of flow of the lubricating oil. With the lubricating oil at the same temperature or at a greater temperature than the bearing, no benefit results from increased flow of oil through the bearing. Under certain conditions of operation it has been found that the lubricating-oil temperature equals or exceeds the bearing temperature. This condition is especially true under hill-climbing conditions. It is therefore apparent that an increased lubricating-oil flow rate would be of little value unless the lubricating oil is cooled.

With the lubricating oil controlled at lower temperatures, an increased oil flow rate results in decreased bearing temperature, principally at the higher crankshaft speeds. Fig. 8 with a lubricating-oil temperature of 160 F and 1750 rpm of the crankshaft, an increase in the flow rate from the normal value of 8.7 gpm to 14 gpm results in a reduction of bearing temperature of 7 F. This reduction is small in comparison to that attainable by control of the lubricating oil temperature and, therefore, this method is not as attractive as is lubricating-oil temperature control.

■ Controlling Oil Temperature

It is recognized by all authorities on automotive lubrication that the lubricating oil temperature giving optimum overall engine performance is 180 F because, at this temperature, water and light hydrocarbon vapors can be removed effectively; the oil is not unreasonably subjected to oxidizing conditions; and bearings of economical design can be used. The heat removal from the lubricant necessary to maintain the oil at this temperature is dependent upon the engine load as shown in Fig. 9.

At 120 bhp and 1750 rpm the cooling needed to maintain 180 F oil temperature is 750 Btu per min. The estimated heat removal at 150 bhp and 1750 rpm is 950 Btu

per min, equivalent to 22.8 hp. This is the maximum dissipation of heat that would be necessary in operation.

The following presents the results of a study of the various types of heat exchangers and their suitability for controlling the lubricating oil temperature:

Coil of Copper Tubing

A simple heat exchanger most suitable for stationary installations where a source of cooling water at a lower temperature than the engine cooling water is available, consists of a copper coil immersed in a bath of water. A test made on such a unit using ½-in. copper tubing revealed a heat-transfer rate as high as 90 Btu per sq ft per hr per deg F, with a value of 60 being a safe one to use in designing such a unit.

Heat Exchanger

A cooler designed for heat exchange between the engine cooling water of a marine installation and the sea water was tested to determine its effectiveness when using the jacket water of a diesel engine to cool the lubricating oil. The results of tests on a cooler having an estimated area of 3.88 sq ft, are shown on Fig. 9, together with data indicating the amount of heat removal from the lubricating oil necessary to attain any given temperature. The results of tests on this unit indicated it to have a heat-transfer factor of 110 Btu per sq ft per hr per deg F. As indicated on Fig. 9 this exchanger was capable of reducing the lubricating-oil temperature from 240 F to 224 F, with the engine operating at 1750 rpm and 120 bhp. This is equivalent to a reduction in bearing temperature from 244 F to 232 F or 12 F. This is a material reduction in bearing temperature, but it may not be large enough to insure adequate control of bearing temperature under all conditions of operation. It is estimated that, by the use of a similar exchanger having four times the area of the one tested, the bearing temperature under the same operating conditions would be reduced to 218 F, or a total reduction of 26 F. This amount of cooling would no doubt be manifested in greater bearing life, but the cost of such an installation and the space required for such a unit might be prohibitive.

Also, with this exchanger of 15.52 sq ft area, the lubricating oil cannot be cooled lower than about 25 F above the cooling water temperatures under conditions of maximum engine loading. This means that, under warm-weather operation, the controlled temperature of the lubricating oil could be as high as 230 F, or 50 F above the desirable operation temperature of 180 F.

From the data obtained with this type of cooler it is estimated that the minimum size for an oil cooler from which some benefit could be obtained (using the engine cooling water) is one in which the product of the area and the heat-transfer factor is 1700 Btu per hr per deg F. This value precludes the use of a water-jacketed crankcase because, in such a case, the low velocity of the lubricating oil in the crankcase would cause a low heat-transfer factor. Assuming a value of 20, which may be high, an area of 85 sq ft is indicated, which is about ten times the area available for water jacketing.

Tractor-Type Cooler

An oil cooler of the type commonly used on tractors mounted in front of the cooling water radiator on diesel-engine-powered trucks, as shown in Fig. 10, has been

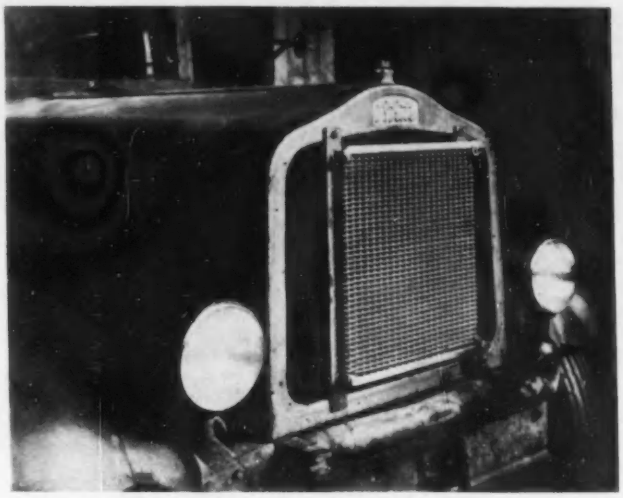


Fig. 10 – Oil cooler mounted in front of the cooling-water radiator on diesel-powered truck

found to be an effective controller of lubricating oil temperature. As high as 40 F reduction in maximum lubricating-oil temperature is found normally on the road. This brings about a reduction in bearing temperature of 30 F. Such an installation appears promising because the cooling medium, air, is at a relatively low temperature, which means that the cooler can be small, light, and free of complicated water plumbing.

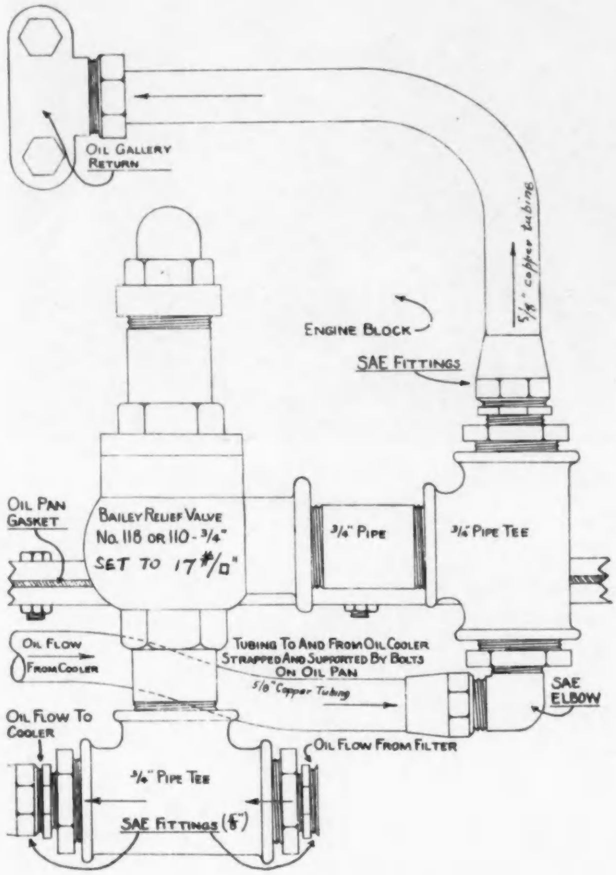
Fig. 11 shows details of the pipe fitting necessary to install the cooler with bypass, on a 6-cyl diesel engine. The mounting of the cooler is illustrated in three photographs showing respectively the cooler mounted in a 16-gage sheet-steel channel frame, Fig. 12; then the assembly front, Fig. 13; and quarter view as mounted on a truck with temperature control shutters, Fig. 14. In building this assembly, one change has been made in the cooler, namely: SAE tubing elbows are brazed into the cooler inlet and outlet looking downward on the left side of the truck and horizontal openings having been plugged.

In the head-on view, it will be noted that the copper-tube oil inlet to the cooler has been covered with a protective loom, and this construction is recommended. It is also necessary to provide a flexible section of tubing between the engine and the radiator. An inexpensive and satisfactory method is to use aircraft-grade gasoline or oil hose, ⅝-in. internal diameter, with hose clamps and an Army-Navy standard hose liner.

Oil Bypass

Fig. 11 illustrates the installation of a bypass around the cooler. While the "Bailey No. 118" unit is shown, other types may be used, the only stipulation being that a ¾-in. or larger unit be chosen. The reasons for the use of a bypass, and the adjustment of it are brought out in the following complete discussion which applies in principle to any full-flow cooler which is to be used over a range of temperatures.

At low atmospheric temperature the lubricating oil is of very high viscosity, 20,000 sec Saybolt Universal and higher. With viscosities of this order, practically no flow through the oil cooler exists, see Fig. 15, and all the oil must pass through the relief valve. The oil pressure that



■ Fig. 11 – Pipe fitting necessary to install oil cooler with bypass on 6-cyl diesel engine

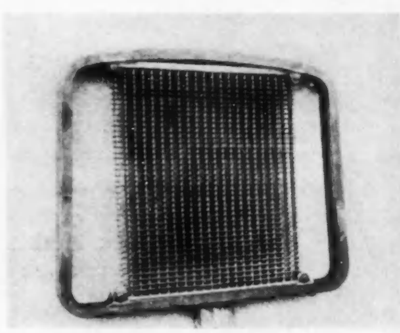
the pump must provide is equal to the pressure drop across the relief valve, plus the pressure at the oil gallery fitting. At low temperatures this latter pressure can be quite high. It is evident that, at low atmospheric temperatures and under starting conditions, the pressure at the pump is high and is increased by the amount of the opening pressure of the relief valve. This high pressure puts a greater load on the pump and causes objectionable "squealing." For this reason the relief-valve pressure should be set as low as is consistent with the function of the system. It is also necessary that the piping in the relief-valve line be of sufficient size not to introduce material resistance under conditions of high oil viscosity.

In order to specify the minimum setting for the relief valve, it was necessary to determine the flow characteristics of the coolers. Fig. 15 presents the data obtained on an air-cooled radiator type of cooler. Plotted are the flow rates through the cooler against the lubricating oil viscosity for pressure drops of 10, 20, 30 and 40 lb per sq in. across the cooler. With these data and the temperature-viscosity relation of the lubricating oil (here considered is an SAE 30 viscosity grade lubricant), a series of flow rate vs lubricating oil temperature curves can be plotted as shown on Fig. 16.

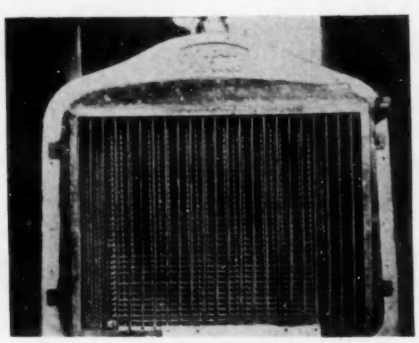
Also plotted is the capacity range of the lubricating oil pump over the normal speed range of the diesel engine. From this curve it is possible to determine the relief-valve setting required for the desired operating condition.

A lubricating oil operating temperature of 180 F is ideal from all considerations including bearing life. Under conditions of low atmospheric temperature the temperature of the lubricating oil can be controlled by means of shutters or a curtain installed in front of the cooler. In order to maintain the oil at 180 F at as high an atmospheric temperature as possible, it is necessary that full flow through the cooler exist when the lubricating oil reaches 180 F. As the lubricating oil pump capacity in the normal operating range is about 8.6 gpm the pressure drop across the cooler must be sufficient to cause this flow rate when the lubricant reaches 180 F. It is seen from Fig. 16 that at 10 lb per sq in. and 180 F. oil temperature the flow rate through the cooler is 5.7 gpm. At 20 lb per sq in. and 180 F the flow rate is 9.9 gpm. The desired flow rate occurs at a pressure between these two values (10 and 20 lb per sq in.) and, as estimated from the drawing, is 17 lb per sq in. The desired setting of the relief valve using the cooler just studied is therefore 17 lb per sq in.

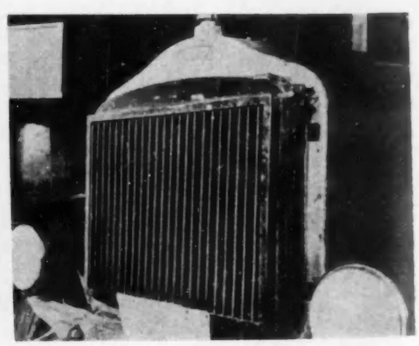
In gathering these data it has been necessary to study fleets in many parts of the country, and under varying conditions. As far as possible the study has been made on fleets using the same type of engine, and both the gasoline



■ Fig. 12 – Oil cooler mounted on 16-gage sheet-steel channel frame

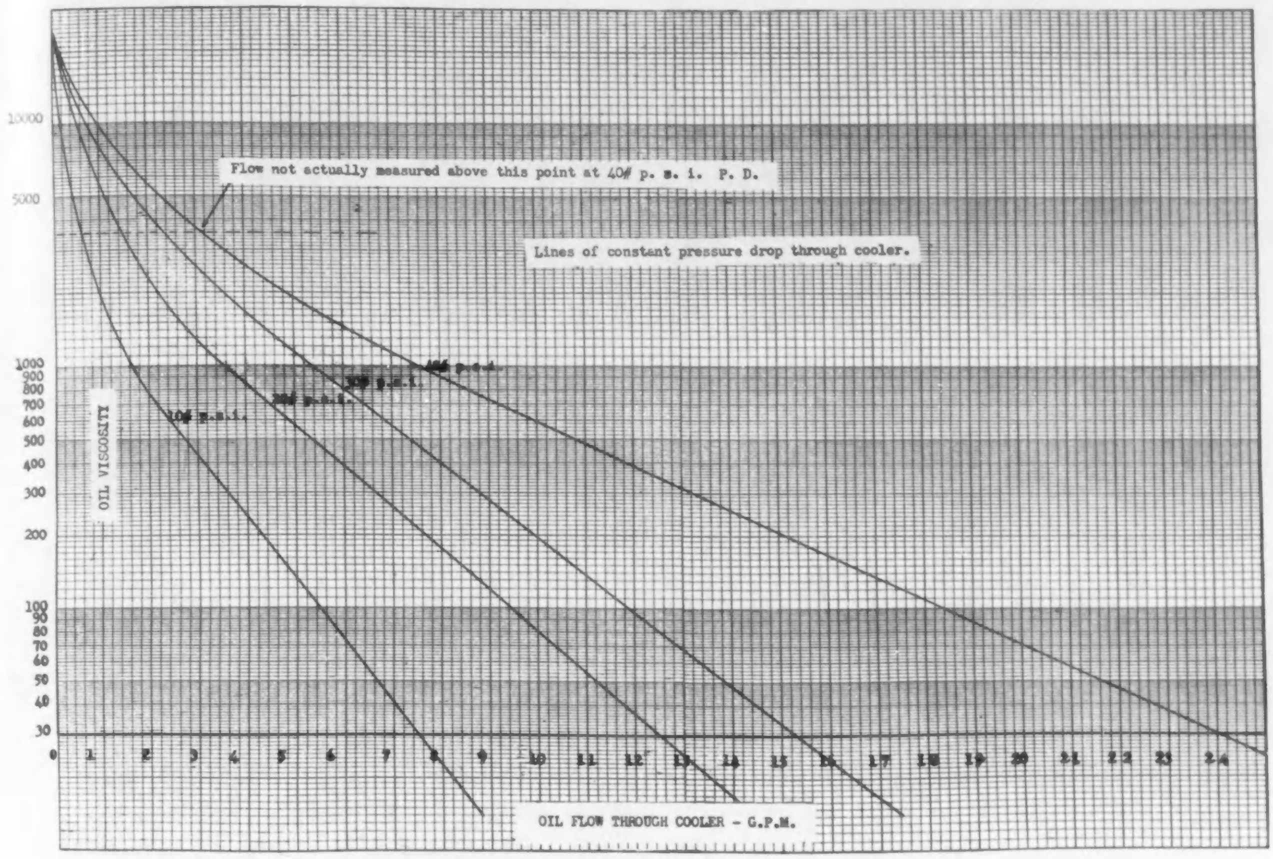


■ Fig. 13 – Front assembly of oil-cooler installation



■ Fig. 14 – Quarter view of oil-cooler installation on truck with temperature-control shutters

Fig. 15 chart:



■ **Fig. 15 – Oil viscosity versus oil flow for air-cooled-radiator type oil cooler**

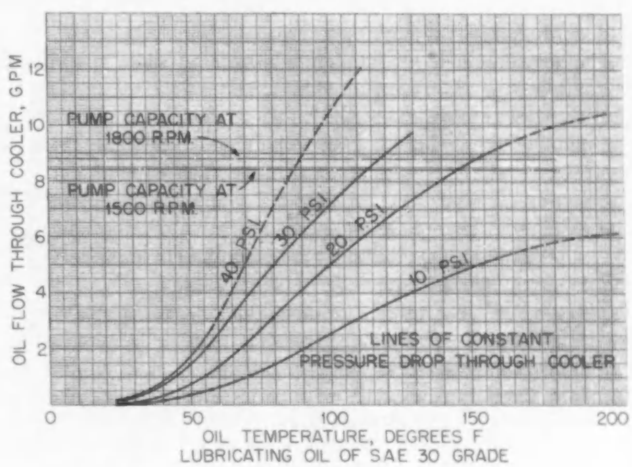
engine and heavy diesel truck types engines have been brought under observation, both in field and laboratory tests.

It was not thought sufficient to study only operations that were giving trouble. Instead, the observations were extended to fleets that were operating practically free from bearing and ring trouble. It is not difficult to find the same type of equipment operating entirely satisfactorily in one territory and giving no end of trouble in another part of the country. When conditions of this kind are encountered, an investigation is made to determine under what conditions the equipment that is operating satisfactorily is working. An investigation is made covering atmospheric temperature, water outlet temperature, water inlet temperature, oil temperature in crankcase sump, bearing temperature, engine compartment temperature, oil pressure at speed and oil pressure at idle, also engine rpm, and the type of roads over which the engine is being operated. The same test is made on equipment that is having bearing failures, and a comparison of the data usually will produce the solution to the trouble.

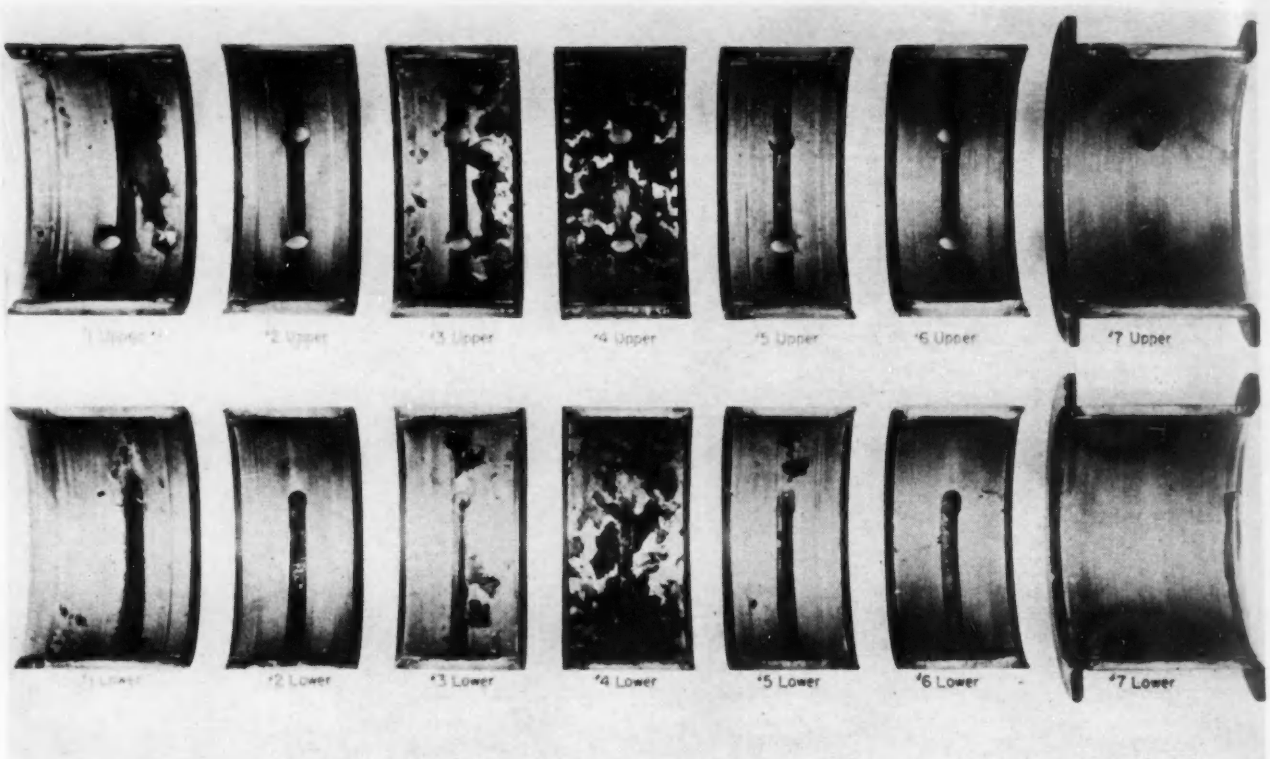
Table 3, showing the field of usefulness of various bearing materials, is based on numerous successful installations, but it is not a positive guarantee of a successful installation. There are too many factors over which the manfacturers of bearings have no control to guarantee 100% bearing life.

Fig. 17 is a good illustration of the failures which might occur, although all engineering calculations show this engine using cadmium-silver bearings to be entirely within the range of the field of usefulness chart. It is quite evident from the condition of the center main bearing in this group that some unusual strains were set up in this engine which caused the bearing to fail. However, when crankcase temperatures are maintained at or below 190 F, failures of this type are very rare in this engine.

If crankcase temperatures of 275 F or over are encountered due either to high atmospheric temperatures or severe



■ **Fig. 16 – Flow rate versus lubricating oil temperature curves for constant pressure drop through oil cooler**



■ Fig. 17 – Example of failure of cadmium-silver bearings in an engine not equipped with an oil cooler

operating conditions, failures of this type are not uncommon in this engine. It is, therefore, concluded that, if proper oil coolers were selected for this engine, oil temperature below 200 F would be maintained, and very little bearing trouble would be encountered.

Referring back to Fig. 3, this engine is being operated successfully with babbitt bearings although the pressures are in excess of 1800 lb per sq in. of projected bearing area. This is made possible through the use of a properly selected oil cooler that keeps the oil temperature at the minimum temperature consistent with economical operation. Where oil temperatures are not high enough to cause serious deterioration of the lubricating oils and where the bearings loads are within the prescribed range, the proper selection of bearing materials may be all that is necessary for normal bearing life. There are some conditions which make the use of higher duty metals impossible for greatest economy.

Again referring to Table 2, it will be noted that shaft hardness is not important with tin and lead base materials. But cadmium-silver must be used on shafts of not less than 250 Brinell, and copper-lead on shafts of 300 Brinell or over. If cadmium-silver or copper-lead bearings are used on shafts of less than the recommended hardness, excessive shaft wear would offset any saving due to longer bearing life.

Where oil sump temperatures are too high for economical operation due to the formation of lacquers and oxidation of the lubricating oil, there seems to be no solution other than to cool the lubricating oil, regardless of whether the bearing material will stand up under these conditions or not.

Table 3 - Load Capacities and Other Characteristics of Bearing Metals

Description of Bearing Metal	Maximum Permissible Unit Pressure, lb per sq in.	Minimum Permissible Zn/P_{max}	Maximum $P_{max} V$	Oil Reservoir Temperature, F	Minimum Crankshaft Hardness	Affected by Corrosion
Tin-Base Babbitt Copper .... 3.50% Antimony .... 7.50% Tin .... 89.00% Lead (max) 0.25%	1000	20	35,000	235	Not important	No
		Standard quality bearings				
Tin-Base Babbitt Same composition as above	1500	15	42,500	235	Not important	No
		Alpha Process quality bearings				
High-Lead Babbitt Tin .... 5 to 7% Antimony ...... 9 to 11% Lead .... 82 to 86% Copper (max) 0.25%	1800	10	40,000	225	No important	No
Cadmium-Silver Silver .... 0.75% Copper .... 0.50% Cadmium 98.75%	Over 1800 and up to 3850	3.75	90,000 and upwards	260	250 Brinell	Not likely if temperature is maintained as specified and proper lubricating oil is used.
Copper-Lead Copper .... 60% Lead 40%	Over 1800	3.75	90,000 and upwards	260	300 Brinell	

FUEL RATING—

Its Relation to Engine Performance

by A. M. ROTHROCK

Senior Physicist, National Advisory Committee for Aeronautics

AN aircraft engine is a means for transferring potential energy within a fuel-air mixture into kinetic energy. In this combination of aircraft engine and aircraft fuel we are primarily interested in three things – the rate at which the energy can be transformed (the power), the efficiency with which the energy can be transformed (the specific fuel consumption), and the reliability of the unit. Of these three factors in aircraft use, reliability is the most important. To control these three factors in the operation of aircraft we must know the manner in which the fuel "fits" the engine. For this reason we must determine those characteristics of the fuel which affect its performance within the engine. There are many of these characteristics which must be measured. We must know the vapor pressure of the fuel, the specific gravity of the fuel, the boiling range of the fuel, and its heat of combustion. These quantities are comparatively easy to measure. They are chiefly important in determining the degree to which the fuel and the air can be mixed successfully and in determining the total energy contained in a given quantity of fuel.

Because the fuel must be burned within the engine, we must know whether or not this burning can be controlled. It was discovered a good many years ago that, under certain operating conditions, this burning cannot be controlled. Sometimes the last part of the fuel which is burned in any one cycle burns with explosive violence at a rate much more rapid than can be permitted safely. If this manner of burning persists, the increased temperatures which accompany it will cause the engine to fail. We term this uncontrolled burning of the last part of the charge to be reached by the flame, knock or detonation. Under other conditions, because of some hot-spot within the cylinder, the charge may be ignited before the spark jumps the gap in the spark plug. This early ignition has the same effect as advancing the spark too far. It may cause the engine to knock; it will probably cause the engine to overheat, and to lose power. It is the evaluation of these two combustion characteristics – preignition and knock – that we attempt to determine when we rate fuels.

The rating of fuels for aircraft engines is a problem in experimental science. For this reason we must rely on our experimental data to supply us with information needed to determine an adequate method of fuel rating. These data are being determined both in service and in the laboratory. But, in spite of the amount of data which are available, we do not yet have sufficient information to permit us to define a method of fuel rating that will rigidly classify a fuel relative to its combustion characteristics. Until such information has been obtained, our ideas on fuel rating must be flexible and we must recognize the necessity

THIS paper presents an analysis of the physical principles involved in knock and preignition as an approach to the solution of the problem of fuel rating. From this examination, the author proceeds to an analysis of the manner in which the different engine operating conditions affect these factors which cause knock and preignition.

Finally, he investigates the extent to which present methods of rating fuels are in accord with the analysis made in order to recommend the lines that future research should take so that knock and preignition can be understood better and so that fuels can be rated more adequately.

Among the conclusions reached are that the knocking characteristics of a fuel cannot be expressed adequately by a single value – that knock depends upon the interrelation of two factors, end-gas density and end-gas temperature; and that, for this reason, variation of actual service values from the laboratory value is unavoidable if a single knock rating is to be used.

The paper emphasizes that preignition and knock must be considered separately and points out the difficulties encountered in attempting to express both characteristics by a single method of fuel rating.

[This paper was presented at a meeting of the Chicago Section of the Society, Chicago, Ill., Feb. 6, 1940.]

of changing these ideas as new facts are discovered. Even though a satisfactory method of fuel rating has not been found, some form of fuel rating is required. For this reason we must at all times use our existing knowledge to the best of our abilities. This use of incomplete knowledge need not lead us astray, provided that we recognize the limitations of our knowledge and are willing to change our ideas as new facts become available.

In rating a fuel according to its combustion characteristics we are limited by our lack of knowledge of the manner in which the conditions to which the fuel is being subjected affects the characteristics we are trying to measure. As we have stated before, in spite of this limitation, it has been necessary to devise some method of fuel rating. The Society of Automotive Engineers deserves credit for the success it has achieved despite this lack of complete explanation of the factors being dealt with. Without some measure of fuel rating the progress which has been made in the development of both aircraft engine and aircraft-engine fuels would have been impossible. Now, in the light of knowledge which has been obtained on combustion phenomena, we are in a position in which we can contemplate rating methods that will be more in accord with the factors controlling the combustion process and which, as a result, will give us rating methods that are more certain of giving us the information we require.

We have stated that in rating fuels we are attempting to evaluate the combustion characteristics of the fuels. To develop a successful method of rating fuels we must examine the manner in which this information is to be used. We must consider fuel rating from the standpoint of the fuel producer and the aircraft operator and from the standpoint of future development in both aircraft fuels and aircraft engines.

In the production of aircraft fuels the manufacturer must know that the fuels which he is producing meet certain standards. Because these fuels are being produced at numerous plants, the test method must be comparatively simple and capable of being conducted by personnel that need not be too highly trained. To a large extent this test requirement is a question of economics.

The aircraft operator must know that the fuels which he purchases will perform satisfactorily in his engines, for the failure of an engine may have disastrous results. The operator requires that the rating of the fuel shall insure safe operation of the fuel, but he is also interested in the economics of the problem. The method of rating a fuel should not, through its inadequacies, place restrictions on the manufacturer that increase the price of the fuel.

From the standpoint of future development of aircraft fuels and of aircraft engines we are interested primarily in determining along what lines engine and fuel development should go. We need not consider the economics of the problem to the same extent which they must be considered in rating current fuels. We need not restrict ourselves in regards to the simplicity of method. The adequacy of the method is of primary importance. In regards to future development we are interested in determining what is the maximum power and what is the minimum fuel consumption that can be gotten from any one fuel and, from this information, in determining the manner in which the engine can be designed to give these results. The problem is here one of fitting the engine to the fuel as much as it is one of fitting the fuel to the engine.

It is probable that fuel rating should be divided into two divisions – first, rating fuels for current engines and, second, rating fuels for future development. The different requirements of these two purposes has led to some misunderstanding. The production engineer in his desire for simplicity has looked with horror on some of the rating methods that have originated in the research laboratory. And the research worker with his desire for extensive facts has at times forgotten that complicated procedures cannot be tolerated in large-scale production. In any case the solution of the problem of fuel rating will be arrived at most successfully through an examination of the physical principles involved in knock and in preignition. It is the purpose of this paper to present a partial analysis of these factors. In this paper few new facts will be presented. The analysis will consist more of a review of the knowledge we already have and an examination of existing methods of fuel rating in the light of this knowledge. We will examine those facts which we have obtained relative to knock and to preignition. From this examination we shall proceed to an analysis of the manner in which the different engine operating conditions affect those factors which control knock and preignition. And finally, we will examine the extent to which present methods of rating fuels are in accord with our analysis. From such a treatment we will be in a position to recommend the lines that future research should take so that knock and preignition can be understood more completely and so that fuels can be rated more adequately.

ANALYSIS

■ Knock

In analyzing those factors which determine whether or not a fuel knocks within an engine cylinder we assume that knock is the sudden burning or completion of burning of the last part of the charge to be reached by the flame front. This assumption is in accord with the experimental evidence which we have on knock, and it is generally accepted as a fact. For reasons which we do not fully understand, the burning up to the time of knock takes place at a comparatively slow rate and then, instead of the last part of the charge burning at the same rate, it goes off with a rapidity that gives the impression of a violent explosion. We know that minute quantities of certain compounds, notably tetraethyl lead, decrease the tendency of a fuel to knock. We do not know why they do so. Because of our lack of knowledge of just what knock is, we are forced to construct certain hypotheses as to the factors which determine whether or not a given fuel will knock in an engine. Having constructed these hypotheses, we must see if they are supported by the experimental facts.

We start with the assumption that, because knock is a chemical reaction, it must be controlled by the temperature and the density of the last part of the charge to burn and by the time which this charge is maintained at this temperature and density. We will term the last portion of the charge to burn, the end gas. Our reasons for assuming that the temperature and density of the end gas determine whether or not the fuel knocks are based on the fact that the temperature determines the velocity and average energy (exclusive of chemical energy) of the molecules within the gas and the density determines the number of molecules per unit volume within the gas and, for this

reason, the distance which any one molecule can travel before striking another. The temperature and the density therefore control the number and the intensity of the molecular impacts within the end gas. We use density in preference to pressure because pressure is a measure of the number and the intensity of the molecular impacts on the wall of the containing vessel and so is a result of both density and temperature. We will assume further that, for any given temperature, there is a given density above which the end gas will knock and below which it will not knock. We are assuming, therefore, that there is a density-temperature relationship for the condition of knock that can be expressed as a single curve. We will expect that, as the temperature of the end gas is increased, the maximum permissible density to which the gas can be raised without knocking will be decreased. In the present analysis we will not consider the factor of time, that is, the time interval during which the end gas is maintained at a given density and temperature. Because of insufficiency of experimental data, we will consider the time factor to be constant. We might express this constancy by saying that we are considering only one engine speed.

If the assumptions which we have just made are correct, we should be able to make an engine knock under a series of operating conditions and express the results as a single curve of end-gas density against end-gas temperature for the condition of knock. The temperature and the density of the last part of the charge to burn have never been measured. In time, we may be able to make such measurements. Until we can do so, we must attempt to estimate the end-gas temperature and density from the engine data which we can record. The end gas is compressed by the piston on its upward stroke in the cylinder. When burning starts at the spark plug, the end gas is compressed further during the burning process. This second stage of compression can be visualized by picturing the flame front as pushing ahead of it the unburned fuel-air mixture. To calculate the temperature and the density of the end gas, we must consider the extent to which the last part of the charge to burn is compressed by the already burned mixture. Following this procedure, we can show that the end-gas temperature and density immediately before its combustion are given approximately by the two expressions[1]

$$K\rho_3 = \frac{RP_1}{T_1}\left(1 + \frac{H}{c_v T_1 R^{\gamma-1}}\right)^{\frac{1}{\gamma}}$$

$$T_3 = T_1 R^{\gamma-1}\left(1 + \frac{H}{c_v T_1 R^{\gamma-1}}\right)^{\frac{\gamma-1}{\gamma}}$$

in which

$K\rho_3$ = a constant times the end-gas density, ρ_3

R = the compression ratio of the engine.

P_1 = the pressure of the charge as it enters the engine.

T_1 = the temperature of the charge as it enters the engine.

H = the heat of combustion of the mixture (1160 Btu per lb of mixture)

c_v = the specific heat of the burned gases at constant volume (0.25 Btu per lb per deg F)

γ = the ratio of the specific heat at constant pressure to the specific heat at constant volume of the burned gases (1.29)

(The numerical values in the parenthesis are those that were used in the present paper.) These equations will not give the true values of either the density or the temperature because we do not know the correct values of all the quantities involved. To list a few, we have not considered the heat-transfer effects that take place during the induction of the charge into the cylinder, or during its compression and combustion. We have not considered the effects of residual gases. The values of the specific heats have never been measured accurately under the conditions with which we are dealing. For these reasons, the results obtained from the equations must be considered as being comparative values.

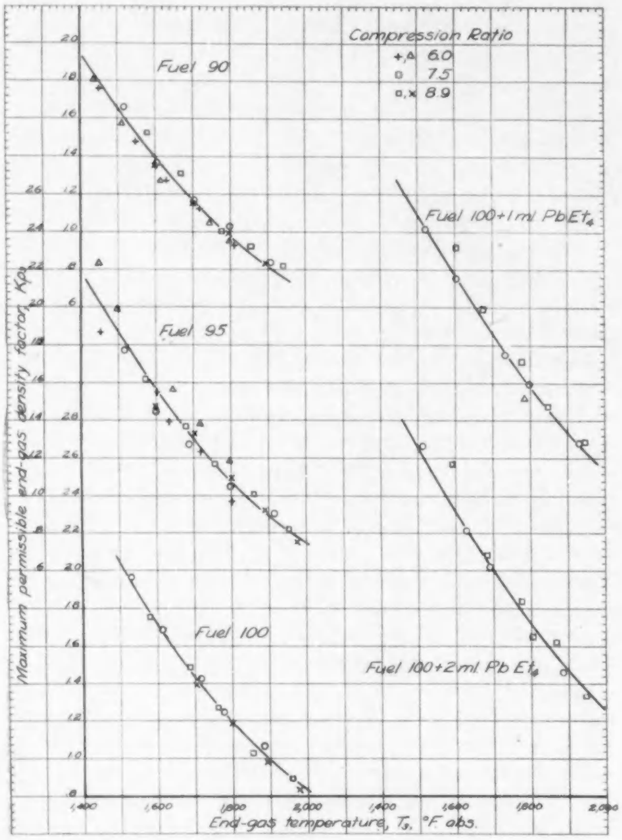
The method used at the NACA laboratories to determine the knocking characteristics of a fuel has been to operate a single-cylinder engine over a series of compression ratios, boost pressures, and inlet temperatures. At each compression ratio the temperature of the incoming air to the carburetor is varied from a temperature of 120 F to a temperature of 300 F in increments of 40 F. At each inlet temperature the pressure of the incoming air is boosted until the engine knocks audibly. The boost pressure is then dropped 2 in. of hg so that conditions can be held stable and the engine data recorded. In the tests for which the results are reported herein a 5¼ by 4¾-in. engine was used. It was operated at a speed of 2500 rpm and was cooled with ethylene glycol at a temperature of 250 F. The spark advance was maintained constant at 29 deg before top center. The results of the tests on five fuels are shown in Table 1. These fuels consisted of mixtures of CFR S-1 and M-2 reference fuels and of S-1 reference fuel plus tetraethyl lead. S-1 fuel is a commercial grade of iso-octane (more correctly known as 2,2,4-trimethyl pentane). By definition of octane number, this fuel has an octane number of approximately 100. The M-2 reference fuel has an octane number of about 11 by the ASTM method – that is, it would knock far too severely to be used under ordinary running conditions.

If we calculate the end-gas conditions from the data in Table 1 according to the formulas already given, we see that the data for any one fuel can be expressed by a single curve expressing the maximum permissible end-gas density as a function of the end-gas temperature (Fig. 1). The curves indicate that our original assumptions are correct. What is the main significance of these curves? This significance is that the knocking characteristics of a given fuel are not constant, but on an absolute basis these characteristics must be expressed as a curve of the maximum permissible end-gas density plotted against the end-gas temperature for the condition of knock.

The fuel producer or the engine operator is not interested in curves such as we have presented here. He is interested in whether or not the knock characteristics of a given fuel can be expressed relative to a standard fuel. Furthermore, in the routine testing of aircraft fuels at the refinery or in the engine laboratory it is not possible to go through as complete a procedure as that followed in determining the curves presented in Fig. 1. These data are not then presented as a method of making routine knock tests. They are presented as a means of understanding more clearly what we are doing when we rate fuels by any method.

In general the procedure by which a fuel is rated consists of varying one or more engine operating conditions and so causing the fuel to knock. Because the knock is controlled by the temperature and the density of the last

[1] See NACA Technical Report No. 655, 1939: "The Knocking Characteristics of Fuels in Relation to Maximum Permissible Performance of Aircraft Engines," by A. M. Rothrock and Arnold E. Biermann.

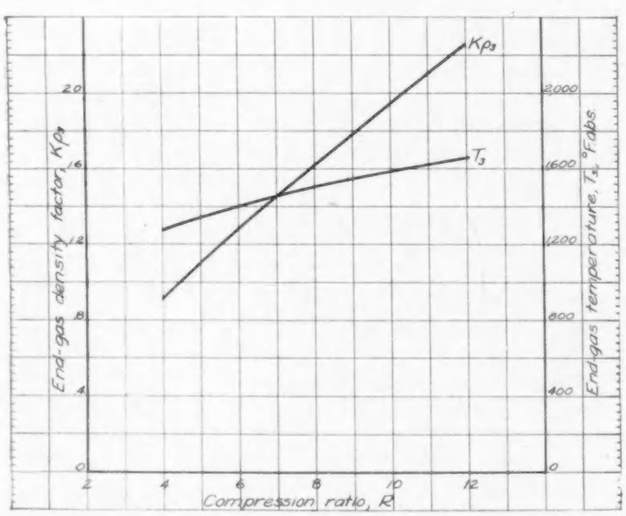


■ **Fig. 1 – Relationship between end-gas temperatures T_3 and maximum permissible end-gas density factor $K\rho_3$, calculated according to Equations (1) and (2) from engine data presented in Table 1**

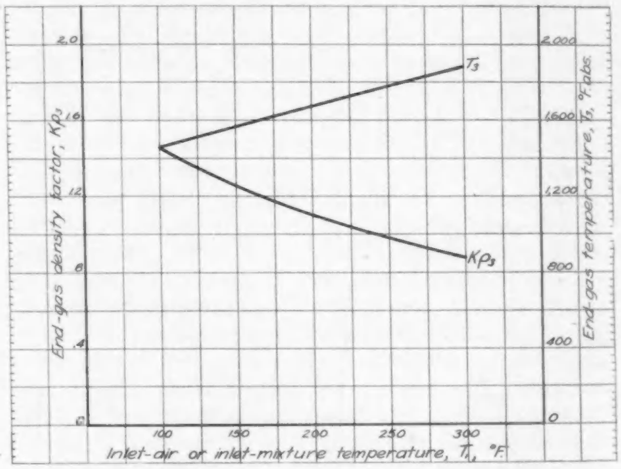
part of the charge to burn, we will examine the manner in which the end-gas temperature and density are varied through variations in the engine operating conditions.

The factors that can be varied most easily are spark advance, compression ratio, inlet air temperature, coolant temperature, engine speed, and inlet air pressure. Fuels could be rated by varying any one of the foregoing variables and maintaining the others constant. Varying the spark advance has the same effective result as varying the compression ratio; varying the coolant temperature apparently has the same effect relative to knock as varying the inlet air temperature. Varying the engine speed varies the time of burning and also varies the temperature and density of the inlet charge because of the effect of speed on volumetric efficiency and on heat transfer. Because we do not have sufficient data to separate the time and temperature effects we will not consider the results obtained by varying the engine speed. We will therefore consider the three variables: compression ratio, inlet air or mixture temperature, and inlet air or mixture pressure.

Fig. 2 shows the effect of the compression ratio on the end-gas conditions. Both the end-gas temperature and density increase as the compression ratio is increased but, for the range of compression ratios shown, the density increases 2.5 times whereas the temperature increases 1.3 times. Fig. 2 also can be assumed to represent the effect of spark advance on the end-gas conditions.

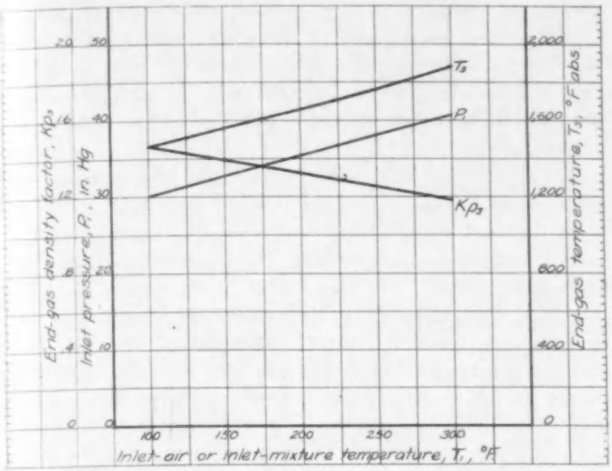


■ **Fig. 2 – Relationship between compression ratio and calculated end-gas conditions – Inlet pressure, 30 in. hg; inlet temperature, 100 F**



■ **Fig. 3 – Relationship between inlet-air or inlet-mixture temperature and calculated end-gas conditions – Inlet pressure, 30 in. hg; compression ratio, 7:1**

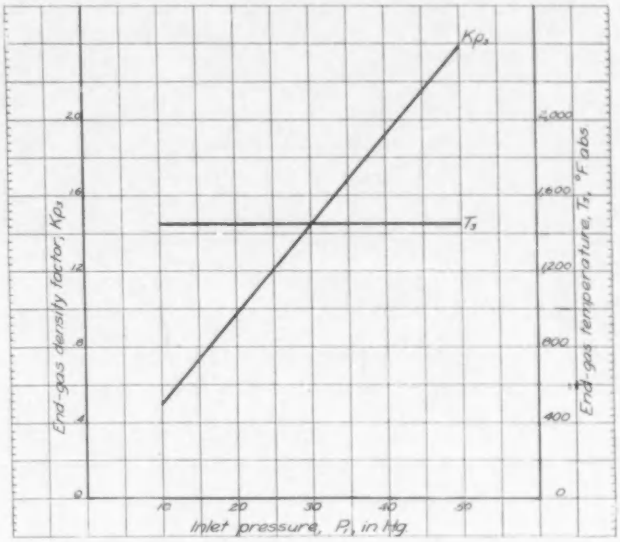
Fig. 3 shows the comparative change in end-gas temperature and end-gas density for different inlet air temperatures. In this case, as the inlet temperature is increased, the end-gas temperature increases but the density decreases. No method of fuel rating is used in which the inlet mixture temperature is varied. But different rating procedures specify different inlet mixture temperatures and, for this reason, the data are of value in analyzing methods of fuel rating. Increasing the inlet temperature and maintaining the inlet pressure constant result in a decrease in the inlet density and a consequent loss in engine power. If the inlet density is maintained constant by increasing the inlet pressure sufficiently to compensate for the increase in the inlet temperature, the results shown in Fig. 4 are obtained. For the condition represented in Fig. 4 in which the compression ratio is constant, the condition of constant inlet density results in constant engine power. These curves represent, therefore, conditions for an engine delivering approximately constant power at different inlet air temperatures. This constant-power requirement is of considerable interest in aircraft work.



■ **Fig. 4 – Relationship between inlet-air or inlet-mixture temperature and calculated end-gas conditions for a constant inlet density (P_1/T_1 = constant) – Compression ratio, 7:1**

Fig. 5 shows the variations in the end-gas temperature and density when the inlet temperature and the compression ratio are maintained constant, but the inlet pressure is varied. These curves represent the condition of the boosted or the throttled engine. The end-gas density increases at a rate directly proportional to the increase in the inlet pressure. The end-gas temperature is independent of the initial pressure and is therefore represented by a straight line.

These preceding figures show the effect on the end-gas temperature and density of the conditions in which we are interested in aircraft operation: (1) variable compression ratio (which has the same effect as varying the spark advance); (2) variable mixture temperature both with decreasing and with constant power output; and (3) variable inlet (supercharging) pressure. Now we must ex-

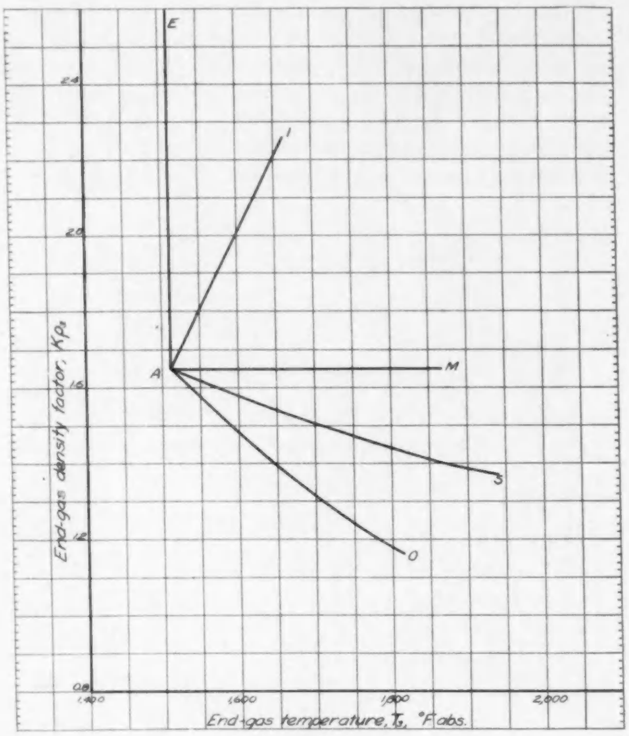


■ **Fig. 5 – Relationship between inlet-air or inlet-mixture pressure and calculated end-gas conditions – Compression ratio, 7:1; inlet temperature, 100 F**

Table 1 – Effect of Inlet-Air Temperature and of Compression Ratio on Maximum Permissible Inlet-Air Pressure (Boost Pressure)

Recorded inlet pressures are 2 in. hg less than pressure for audible knock. Engine speed, 2500 rpm; spark advance, 29 deg BTC; engine coolant temperature, 250 F; mixture ratio for maximum knock.

											Fuel
	90		95		100		100		100		% S-1
	10		5		0		0		0		% M-2
	0		0		0		1		2		ml tetraethyl lead
R	T_1, F	P_1, in. hg	T_1, F	P_1, in. hg	T_1, F	P_1, in. hg	T_1, F	P_1, in. hg	T_1, F	P_1, in. hg	
6.0	125	44.4	125	47.2	...	....	...	....	...	....	
	173	42.5	173	45.9	...	....	...	....	...	....	
	212	40.3	215	44.4	...	....	...	....	...	....	
	257	39.4	256	43.2	...	....	...	....	...	....	
	302	36.2	297	37.3	...	....	...	....	...	....	
	117	44.5	121	55.6	...	....	...	....	...	....	
	153	43.7	146	53.7	...	....	...	....	...	....	
	204	39.7	219	50.3	...	....	...	....	...	....	
	271	37.8	255	48.3	...	....	...	....	...	....	
	297	36.4	296	45.2	...	....	292	57.5	...	....	
7.5	118	34.2	116	36.4	123	41.0	119	49.9	115	54.3	
	158	31.4	156	33.1	161	39.0	158	49.5	170	52.2	
	204	30.4	197	32.3	209	37.1	220	46.8	199	51.5	
	250	29.5	249	30.2	239	34.9	250	45.9	252	47.6	
	300	26.8	305	29.3	292	33.5	313	42.1	291	46.0	
8.9	127	24.9	131	27.1	128	31.6	...	....	...	....	
	170	23.9	168	27.5	171	28.8	...	....	...	....	
	212	22.7	215	25.3	216	27.3	...	....	...	....	
	258	21.1	253	23.7	259	25.1	...	....	...	....	
	117	27.3	303	21.1	297	23.1	...	....	...	....	
	157	26.0	126	28.6	116	31.7	128	43.0	122	46.6	
	206	24.2	171	27.8	164	30.2	162	40.2	163	42.1	
	242	22.6	215	25.8	200	28.4	205	38.5	204	41.0	
	279	21.8	255	24.6	241	25.1	238	36.0	246	40.1	
	...	....	300	22.3	290	24.2	282	34.6	283	35.7	



■ Fig. 6 – Relationship between end-gas temperature and end-gas density for conditions shown in Figs. 2 to 5 – *AE*, variable inlet pressure (boost pressure); *AI*, variable compression ratio; *AM*, constant end-gas density; *AS*, variable inlet temperature with inlet density constant; *AO*, variable inlet temperature with inlet pressure constant

amine the manner in which these variations are related to the end-gas conditions which cause the fuels to knock.

First, we plot curves representing the relation of end-gas density to end-gas temperatures for the conditions that have been discussed. The results are shown in Fig. 6, the data being calculated from Equations (1) and (2). So that the curves can be compared later with the fuel rating curves which already have been determined, a point common to all the curves has been chosen. This point labeled *A* represents the following conditions:

Compression ratio 8.2:1
Inlet pressure 30 in. hg
Inlet temperature 100 F

Curve *AE* represents the relation between the end-gas temperature and the end-gas density for a variable inlet pressure with the compression ratio and inlet temperature constant. Curve *AI* represents a variable compression ratio with the inlet temperature and the inlet pressure constant. Curve *AS* represents a variable inlet temperature with a constant compression ratio and constant inlet density. Curve *AO* represents a variable inlet temperature with a constant compression ratio and a constant inlet pressure. A constant end-gas density curve *AM* also has been drawn for comparison with the constant end-gas temperature curve *AE*.

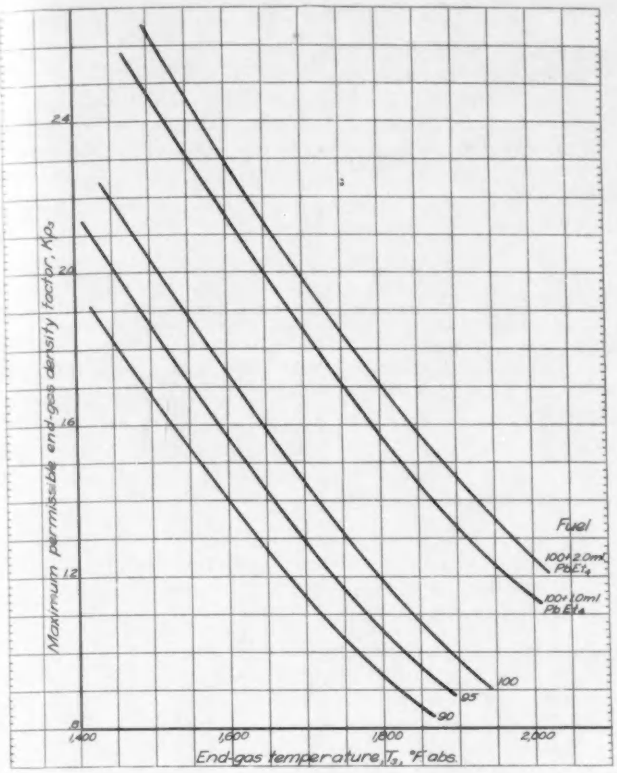
The curves shown in Fig. 1, which represent the end-gas conditions for knock with the five fuels tested, are plotted as shown in Fig. 7. These curves show the maximum permissible end-gas density plotted as a function of the end-gas temperature for the blends of S-1 and M-2 reference fuels and S-1 plus tetraethyl lead. We note that the curves are parallel. This fact is attributable to the fact that the fuels are all similar in nature. The curves show that the introduction of 1 ml of tetraethyl lead to the S-1 fuel resulted in greater improvement in maximum permissible end-gas density than did the change from 90% to 100% S-1.

The curves shown in Fig. 6 are next superimposed on those of Fig. 7 as shown in Fig. 8. The line *AI* represents the conditions under which these fuels are rated if they are rated by determining the maximum permissible compression ratio. This procedure is followed in the various CFR methods. As the antiknock property of the fuel is increased, the fuel is rated at increasing values of both end-gas temperature and end-gas density. The line *AE* represents the conditions under which the fuels are rated when the maximum permissible boost pressure is determined. This method has not been accepted but has been suggested by various laboratories. Comparing the two lines *AI* and *AE* shows that normally we will expect the percentage increase in permissible boost pressure to be greater than the percentage increase in permissible compression ratio. The boost-pressure methods tend to rate the fuels at a constant end-gas temperature. The line *AM* represents the conditions of the end gas under which the fuels are compared if they are rated at a constant end-gas density. Such a method has not been considered in practice. The curves *AS* and *AO* represent the end-gas conditions at which the fuels would be rated if the maximum permissible inlet temperature were the determining criterion. In this case, as the antiknock value of the fuel is increased, it is rated at a successively higher end-gas temperature but at a successively lower end-gas density. This method of rating is of interest only in comparing rating methods to aircraft-engine operating conditions.

Table 2 lists the operating conditions which result in the end-gas conditions shown in the lettered points in Fig. 8. Some of these conditions do not represent practical methods of testing fuels. For instance, if the end-gas density were maintained constant, two of the three variables, compression ratio, inlet temperature, and inlet pressure, would have to be varied simultaneously. The data are presented to show different combinations of the foregoing three variables that can be used to obtain the different end-gas conditions.

If all fuels had knock-rating curves that were parallel, as is the case for those shown in Fig. 8, the problem of knock rating would be comparatively simple and the methods which are now in use would probably prove adequate because, regardless of engine conditions, the fuels would be rated in the same order. Experience with different fuels has shown us that the knock characteristic curves are not all parallel. Fuels of differing chemical constituents will give curves of different slopes. For this reason fuels which give one octane number at one condition of engine coolant temperature, inlet temperature, or engine speed will give another octane number at another value of any one of these three variables. This variation of the octane number for any one fuel with engine operating conditions has been the problem that has caused the greatest amount of trouble in the different methods of fuel rating. We can explain this variation in octane number with engine operating conditions from the data presented. First we had better review briefly just what is meant by octane number.

The efficiency of an internal-combustion engine is a

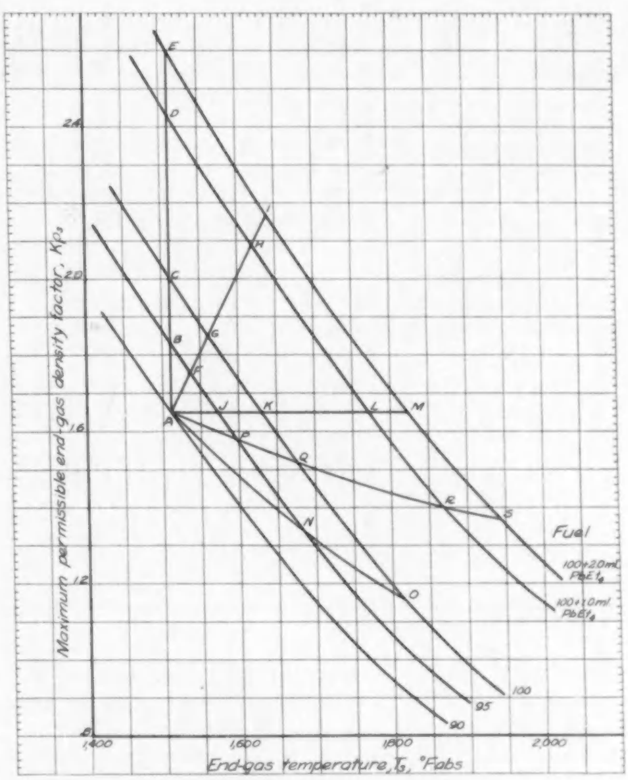


■ Fig. 7 – Fuel rating curves presented in Fig. 1

function of the compression ratio of the engine. For this reason, methods for permitting the compression ratio of an engine to be increased have been the subject of research since the first internal-combustion engine was built. For any one fuel in a given engine, one factor limiting the compression ratio is the knocking characteristics of the fuel. For this reason when the question of rating fuels became important, Ricardo suggested that each fuel be rated according to the maximum permissible compression ratio for that fuel. Unfortunately, this value of compression ratio for any one fuel was different in different engines or even in the same engine if some other engine factor were varied. In an attempt to overcome this difficulty, it was decided to compare each unknown fuel to some standard fuel to which some antiknock compound was added or with which some antiknock fuel had been blended. This method of fuel rating had two possible advantages. In the first place, it expressed the knock rating of a fuel as the amount of antiknock compound or antiknock fuel which added to the standard fuel gave the same knock characteristics as the unknown fuel and did not express the knock rating as a function of engine operating conditions. Second, it was believed that, by comparing the unknown fuel to a standard fuel, there would be less variation in the knock rating value determined by different laboratories than would be the case if some engine function such as highest useful compression ratio were used as the knock characteristic designation.

The choice of the fuel or fuels to be used as a standard was not simple. Early suggestions were benzene and toluene to be blended with the standard fuel or tetraethyl lead or aniline to be used as antiknock compounds in the standard fuel. In 1927 Edgar[2] suggested that each unknown fuel be compared to a blend of normal heptane (a fuel of low antiknock characteristics) in one of the octanes, 2,2,4 trimethyl pentane (a fuel of high antiknock characteristics). Both these fuels could be prepared to a high degree of purity. These fuels were adopted, and from them, the designation of "octane number" as the antiknock value of a fuel. We can define octane number as follows: The octane number of a fuel is that percentage of iso-octane in a mixture of iso-octane and normal heptane which has the same knock characteristics as does the fuel being rated under a given set of engine conditions. Or, to be more specific – According to the ASTM procedure of knock testing, the octane number of any fuel is equal to the percentage of iso-octane in a mixture of iso-octane and normal heptane which has the same highest useful compression ratio as does the fuel being rated.

Returning to the question of the variation in octane number with engine operating conditions, in Fig. 9 are drawn knocking characteristic curves for fuels of 90 to 100 octane number in increments of 2 octane numbers. These curves have been derived from the 90, 95, and 100% S-1 curves shown in the previous figure, it being assumed that the percentage of S-1 in M-2 is equivalent to the octane number of the fuel. On these curves have been superimposed assumed knocking characteristic curves for two unknown fuels, *X* and *Y*. The curve for fuel *X* has a numerical value of the slope greater than that for the S-1 blends, and the curve for fuel *Y* a value less than that for the S-1 blends. The curve *AI* represents the end-gas con-



■ Fig. 8 – Curves of end-gas condition from Fig. 6 superimposed on fuel-rating curves shown in Fig. 7 – *AE*, variable inlet pressure (boost pressure); *AI*, variable compression ratio; *AM*, constant end-gas density; *AS*, variable inlet temperature with constant inlet density; *AO*, variable inlet temperature with constant inlet pressure

[2] See *Industrial and Engineering Chemistry*, Vol. 19, January, 1927, pp. 145-146: "Measurement of Knock Characteristics of Gasoline in Terms of a Standard Fuel," by Graham Edgar.

Table 2 – Effects of Different Methods of Rating Fuels on the End-Gas Temperature and the End-Gas Density at which the Fuels are Rated

Fuel, % S-1 in M-1	Symbol	Constant	Variable	R	P_1, in. hg	T_1, F	$K\rho_3$	T_3, F absolute
			I – Maximum permissible boost					
90	A	R, T_1, T_3	P_1, $K\rho_3$	8.2	30.0	100	1.65	1510
95	B			8.2	33.5	100	1.84	1510
100	C			8.2	36.5	100	2.01	1510
100 + 1.0 ml TEL	D			8.2	44.2	100	2.43	1510
100 + 2.0 ml TEL	E			8.2	47.3	100	2.60	1510
			II – Maximum permissible compression ratio					
90	A	P_1, T_1	R, $K\rho_3$, T_3	8.2	30.0	100	1.65	1510
95	F			8.7	30.0	100	1.75	1535
100	G			9.3	30.0	100	1.85	1562
100 + 1.0 ml TEL	H			10.9	30.0	100	2.09	1620
100 + 2.0 ml TEL	I			11.5	30.0	100	2.17	1640
			III – Maximum permissible inlet-air temperature, inlet pressure constant					
90	A	R, P_1	T_1, $K\rho_3$, T_3	8.2	30.0	100	1.65	1510
95	N			8.2	30.0	175	1.35	1675
100	O			8.2	30.0	240	1.16	1815
100 + 1.0 ml TEL								
100 + 2.0 ml TEL								
			IV – Maximum permissible inlet-air temperature, inlet density constant					
90	A	R, P_1/T_1	T_1, P_1, $K\rho_3$, T_3	8.2	30.0	100	1.65	1510
95	P			8.2	32.0	138	1.58	1596
100	Q			8.2	34.0	174	1.52	1673
100 + 1.0 ml TEL	R			8.2	38.9	266	1.40	1869
100 + 2.0 ml TEL	S			8.2	40.6	298	1.37	1943
			V – Maximum permissible end-gas density					
90	A	R, T_1, T_3	P_1, $K\rho_3$	8.2	30.0	100	1.65	1510
95	B			8.2	33.5	100	1.84	1510
100	C			8.2	36.5	100	2.01	1510
100 + 1.0 ml TEL	D			8.2	44.2	100	2.43	1510
100 + 2.0 ml TEL	E			8.2	47.3	100	2.60	1510
90	A	P_1, T_3	R, T_1, $K\rho_3$	8.2	30.0	100	1.65	1510
95	B			9.0	30.0	84	1.84	1510
100	C			9.5	30.0	75	2.01	1510
100 + 1.0 ml TEL	D			11.0	30.0	54	2.43	1510
100 + 2.0 ml TEL	E			11.5	30.0	46	2.60	1510
			VI – Maximum permissible end-gas temperature					
90	A	R, $K\rho_3$	P_1, T_1, T_3	8.2	30.0	100	1.65	1510
95	J			8.2	32.4	126	1.65	1570
100	K			8.2	34.8	154	1.65	1630
100 + 1.0 ml TEL	L			8.2	41.4	220	1.65	1770
100 + 2.0 ml TEL	M			8.2	43.3	242	1.65	1820
90	A	T_1, $K\rho_3$	R, P_1, T_3	8.2	30.0	100	1.65	1510
95	J			9.6	26.4	100	1.65	1570
100	K			11.3	23.1	100	1.65	1630
100 + 1.0 ml TEL	L			15.9	17.5	100	1.65	1770
100 + 2.0 ml TEL	M			17.9	15.9	100	1.65	1820
90	A	P_1, $K\rho_3$	R, T_1, T_3	8.2	30.0	100	1.65	1510
95	J			8.7	30.0	116	1.65	1570
100	K			9.2	30.0	134	1.65	1630
100 + 1.0 ml TEL	L			10.4	30.0	172	1.65	1770
100 + 2.0 ml TEL	M			10.9	30.0	186	1.65	1820

ditions for a varying compression ratio with an inlet air temperature of 100 F. This curve is the same as curve *AI* shown in the previous figure. The curve *A'I'* represents the end-gas conditions for a varying compression ratio and an inlet air temperature of 300 F. For each of the curves *AI* and *A'I'* the inlet pressure is 30 in. hg. If the two unknown fuels are rated at an inlet temperature of 100 F, fuel *Y* will have an octane number of 93 and fuel *X* an octane number of 99. If the two fuels are rated at an inlet temperature of 300 F, fuel *Y* will have an octane number of 99 and fuel *X* an octane number of 91. Under these conditions we state that increasing the inlet temperature at which the fuels are rated causes fuel *Y* to appreciate and fuel *X* to depreciate. This statement defines the slope of the knock curves of the unknown fuels relative to the slope of the knock curves of the standard fuels. The term has no other significance.

The variations in octane number of the two assumed

fuels in Fig. 9 are within the variation that is experienced in practice. For this reason, we can accept these curves as probably representing conditions that do exist although experimental verification is needed. If the curves shown in Fig. 9 represent the basic knock characteristics of fuels, we must then ask the question – can a single method of knock rating prove satisfactory? The answer is "yes" if we recognize the cause of variation in octane number of a given fuel under different operating conditions, and if we accept this variation as not being objectionable. If the variation that does exist in the octane number of a given fuel under different operating conditions is objectionable, we must consider means of overcoming this objection. The method that has been used most generally to decrease the variation has been to attempt to find some single set of engine conditions under which a fuel can be rated that will represent an average octane number. There does not seem to be much chance of finding such a solution. Even had we chosen an engine temperature in the preceding figure that represents the intersection of the two unknown fuels, we have no reason to believe that a third unknown fuel would also intersect at the same point. We must recognize the fact that, if for reasons of simplicity we rate a fuel at one and at only one condition, we must expect an appreciable deviation in the octane number as determined in the laboratory and the octane number as determined in service for, in the latter case, the engine operates under a wide variety of conditions. With the increased interest in aromatic fuels and in synthesic fuels in general, it is to be expected that this variation between laboratory octane number and service octane number will increase rather than decrease.

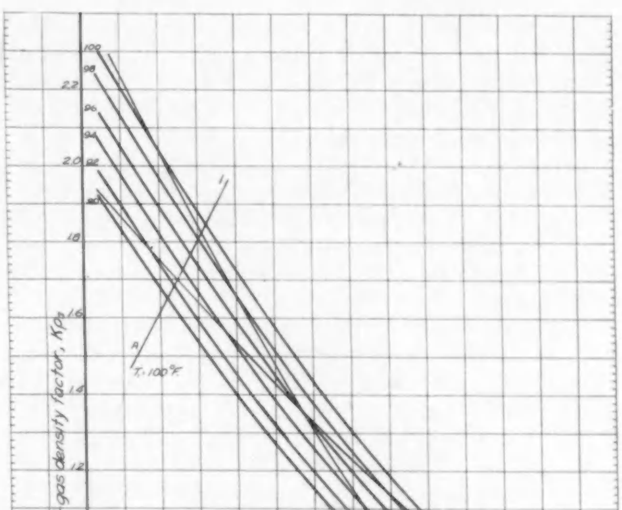
In rating aircraft fuels it has been suggested that the fuels be rated at the "most severe condition." This method of rating might unduly penalize certain fuels, but it can be justified in the interests of safety. It is difficult to justify it in the interests of accuracy. In Fig. 9 what shall we consider to be the most severe condition? As the end-gas temperature is increased, the octane number of fuel Y increases; whereas, with fuel X the octane number decreases with an increase in the end-gas temperature. If fuels were rated at even two sets of conditions, representing the extremes, these two octane numbers of any one fuel would indicate the slope of the fuel curve relative to the slope of the curve for the reference fuels. Such a procedure, which has been suggested by Edgar[3], would remove the main inadequacy of a single octane number and, at the same time, would rate the fuels in relation to a reference fuel and not to maximum permissible engine output.

If we cannot define a most severe condition relative to the octane number of several fuels, can we define a most severe condition relative to the operating conditions experienced in service? We have shown that the knock characteristics of a fuel on an absolute basis must be represented by a curve and not by a single value. An engine in service operates under numerous end-gas conditions. We have no reason to believe that, if two fuels having knock characteristic curves of different slopes are operated in the same engine, the two fuels will be most prone to knock under the same operating condition. Experience has shown that, given two fuels of the same or nearly the same octane number, one will knock under cruising conditions and the other under take-off conditions. To insure safe operation these fuels must be rated under these two conditions.

■ Preignition

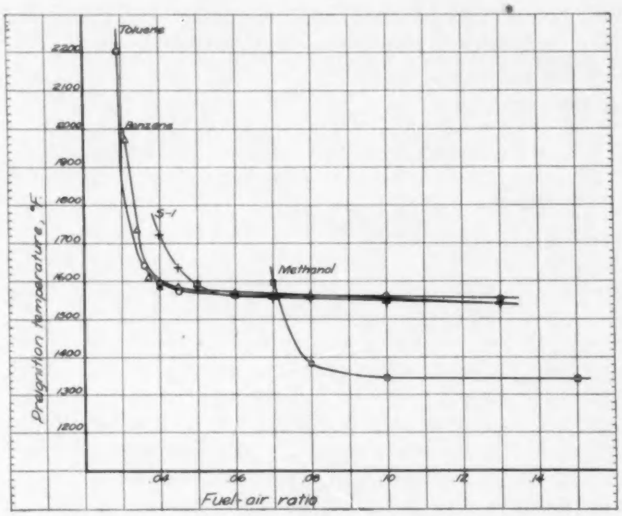
The data presented so far have been limited to knock. Equally important from the standpoint of engine failure is the problem of preignition. Under present fuel rating procedures a fuel that fails because of preignition is given an octane number and the same procedure is used in determining this octane number as is used for fuels which fail through knock. Yet there is little experimental evidence to support the contention that these two phenomena, preignition (including after ignition) and knock, are controlled by the same factors. It is quite true that a failure from one cause can be just as disastrous as a failure from the other and, for this reason, adequate assurance must be given that neither will occur. But it does not follow from this contention that one rating method is adequate for these two forms of combustion.

In the analysis of knock we started with the assumption that whether or not a fuel knocked depended on the density and on the temperature of the end gas within the combustion chamber. In dealing with preignition we are not interested in the end gas but in the first part of the charge that burns. Whereas knock is probably a gaseous phenomenon, that is, it originates within the gas, preignition is probably a surface phenomenon and originates from the contact of the gases with a hot surface within the combustion chamber. For this reason it is possible that we must consider three variables (again excluding time) –



■ **Fig. 9 – Variation of octane number of two fuels, X and Y, with end-gas conditions – Curves for different octane numbers extrapolated between curves shown in Fig. 7 – AI, end-gas conditions for variable compression ratio with inlet temperature of 100 F – $A'I'$, end-gas conditions for variable compression ratio with inlet temperature of 300 F**

[3] See SAE Journal, Vol. 43, No. 3, September, 1938, pp. 7-12, 17-20: "Knock Testing – In the Laboratory and in Service," by Graham Edgar.



■ Fig. 10 – Effect of fuel-air ratio on preignition temperature of four fuels – Engine speed, 500 rpm; compression ratio, 7.0:1; engine coolant temperature, 250 F

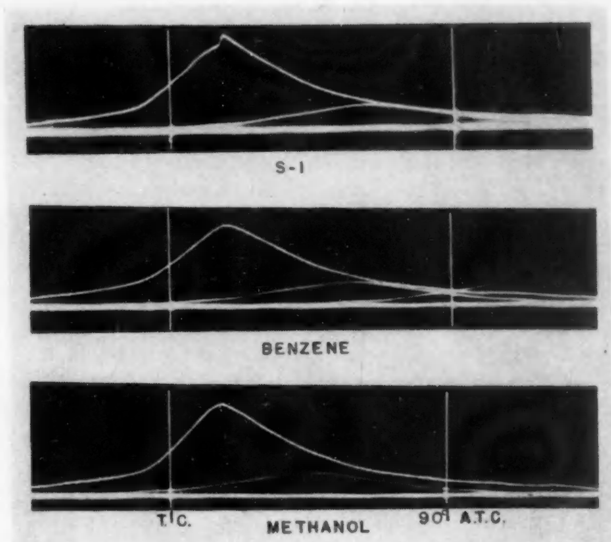
the density of the gas, the temperature of the gas, and the temperature of the wall. The study of the ignition of gases on hot surfaces is not simple. The problem of the adsorption of the gases on the surface may be of importance and if so we will expect that different surface materials will result in different preignition temperatures. This phase of the problem is too complicated to be discussed here and, in any case, more information is needed on it before definite conclusions can be drawn. It may well be that, because of these additional variables, the rating of the preignition characteristics of a fuel will be more complicated than the rating of the knocking characteristics. Constructing a hypothesis for the interrelation of the factors controlling preignition is not as simple as constructing a hypothesis for the factors controlling knock because of the additional variables involved in preignition. Therefore, we will examine the data which we have on preignition to see if these data will suggest certain principles on which we can work. It will be noticed that, in this analysis, we will run into conflicting evidence which must be the subject of future research.

We have recently obtained data at the NACA laboratories on the preignition temperature of several fuels. The procedure consists of placing an electrically heated wire in the combustion chamber of a single-cylinder engine. The engine is driven by an electric motor at the desired test speed. A single charge of fuel is injected into the engine during the intake stroke. By successive tests with the wire at different temperatures, the minimum temperature required to ignite the fuel is determined. Sample results are shown in Fig. 10. The striking factor shown in these curves is that benzene, toluene, and S-1 all "preignited" at about the same temperature of the heated wire. The temperature required to ignite methanol was considerably lower. For each fuel, as the mixture was leaned from a rich value, the preignition temperature remained about constant until the mixture reached a value approximately equal to that of the chemically correct ratio. Further leaning of the mixture resulted in a rapid increase in the preignition temperature.

[4] See *Shell Aviation News*, December, 1937: "Essais de detonation avec suralimentation," by S. D. Heron and E. Gillig, presented at the World Petroleum Congress, Paris, June 14-19, 1937.

When these same three fuels were ignited with the spark plug and time-pressure records taken to determine whether or not the fuels knocked, it was found that conditions which resulted in a fairly severe knock with the S-1 did not result in knock with either the benzene or the methanol (Fig. 11).

Additional information on the characteristics of fuels which are prone to knock in comparison with fuels which are prone to preignite is presented by Heron and Gillig[4]. In these tests a series of fuels was run in a CFR single-cylinder test engine. Two different engine speeds and engine coolant temperatures were used. Under each test condition the engine was boosted until either knock or preignition or after ignition occurred. Fig. 12 shows the maximum permissible imep's recorded with S-1, methanol, and benzene under the four engine conditions. In each case S-1 was limited by knock, and methanol and benzene were limited by preignition or after ignition. With the knocking fuel the quadrangle formed by joining the points



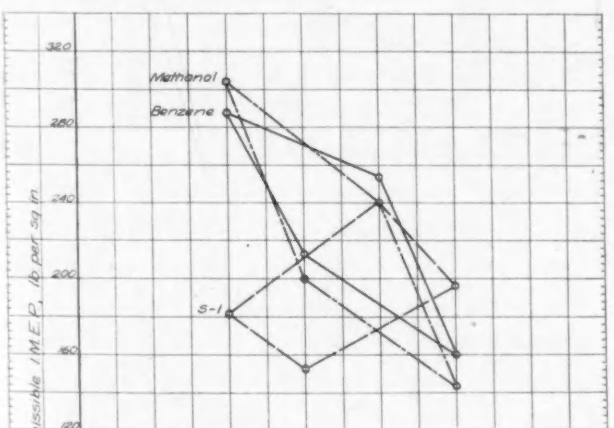
■ Fig. 11 – Indicator cards for three fuels – Compression ratio, 7.0:1; engine coolant temperature, 310 F; engine speed, 500 rpm; spark advance, 30 deg; one spark plug – Mixture ratio for complete combustion

in the manner shown inclined upward to the right whereas, with the preigniting fuels, the quadrangles inclined downward. Increasing the engine coolant temperature increases the temperature of the walls of the engine cylinder and combustion chamber and also increases the temperature of the fuel-air mixture within the cylinder. Increasing the engine speed increases the cylinder and combustion-chamber wall temperatures because of the increased heat flow through the walls. Increasing the engine speed, for those cases in which the incoming mixture is cooler than the cylinder walls, decreases the gas temperatures within the cylinder because there is less time during the intake and early part of the compression stroke for heat to be transferred from the cylinder to the incoming gases. In these tests the incoming mixture was at a temperature of about 80 F. Examination of the data in Fig. 12 shows that increasing the engine coolant temperature decreased the maximum permissible imep with either the knocking or

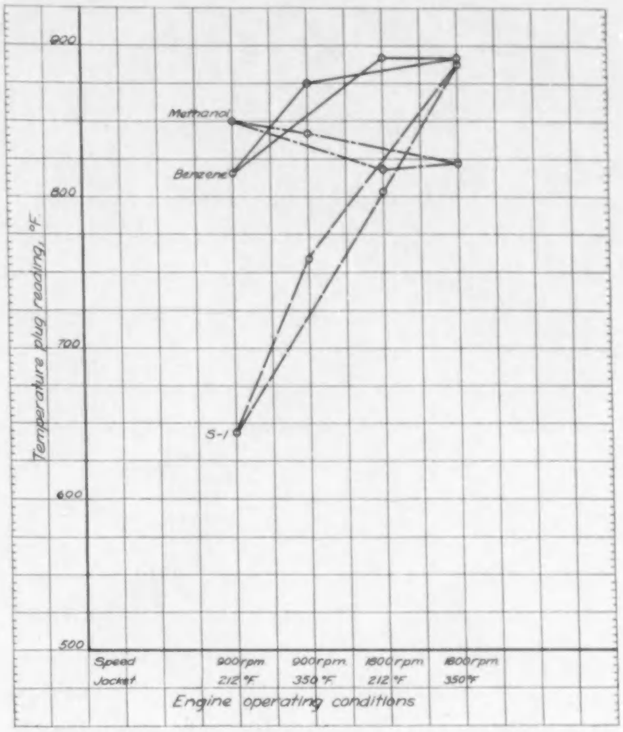
the preigniting fuels. Increasing the engine speed increased the maximum permissible imep with the knocking fuel, but decreased the imep with the preigniting fuels. The data indicate that, whereas knock is a function of the gas temperature within the cylinder, preignition is chiefly a function of the cylinder-wall temperature. Or, worded in another way and drawing on the previous analysis of knock: The fuel knocks when the end-gas density exceeds the maximum permissible value for the particular end-gas temperature within the cylinder; and the fuel preignites when the hottest surface within the cylinder exceeds the preignition temperature of the fuel. Which of the phenomenon happens first depends on which condition is reached first.

There is no reason to believe that all fuels cannot be made to preignite, providing some surface within the combustion chamber reaches the requisite temperature. In regard to knock, we do not know whether or not a similar statement can be made. I do not believe that data have been published showing any of the preigniting fuels considered here, benzene, toluene, and methanol, to knock. If it is true that certain hydrocarbons will not knock, the proof of the statement would be of considerable assistance in the investigation of those factors which control knock.

The investigations on preignition conducted by the NACA have indicated that the effects of compression ratio and of heating the engine cylinder walls are only effective in regards to preignition to the extent to which these variables affect the temperature of the hot-spot. These data indicate that the hot-spot temperature required to preignite any given fuel does not vary much with different engine operating conditions. Further support of this contention is presented in the thermal-plug temperatures recorded by



■ Fig. 12 – Effect of engine operating conditions on maximum permissible imep for S-1, benzene, and methanol, CFR engine (data from Heron and Gillig[4]) – Inlet mixture, 80-95 F; spark advance 30 deg; compression ratio, 5.5:1

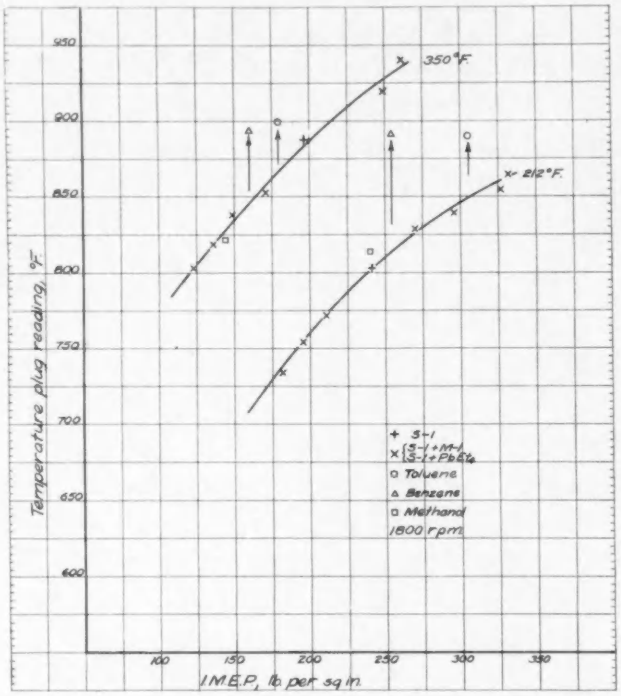


■ Fig. 13 – Temperature plug readings for maximum permissible imep's recorded in Fig. 11 (data from Heron and Gillig[4])

Heron and Gillig[4] in their tests. The thermal plug consists of a metal disc mounted in a spark-plug hole. In the face of the disc a thermocouple is mounted. The temperature of this thermocouple does not necessarily represent the temperature of the hottest surface within the engine-combustion chamber, but it is probably an indication of the surface temperatures. Fig. 13 shows the thermal-plug temperatures recorded for S-1, methanol, and benzene at the same time the maximum permissible imep's shown in Fig. 12 were recorded. With S-1 the thermal-plug temperature reading showed considerable variation for the boost pressures causing the fuel to knock at the four different engine conditions. With the two preigniting fuels, methanol and benzene, the thermal-plug temperatures remained more nearly constant. The data present some justification for the statement that the preignition temperature of a given fuel under different engine conditions is nearly constant. But, as was stated previously, more data are needed to determine whether or not this statement is generally justified.

In Fig. 10 it was indicated that the preignition temperatures of benzene, toluene, and S-1 are nearly the same. Yet, in Fig. 12, it was shown that, at an engine speed of 1800 rpm and a jacket temperature of 350 F, S-1 had a maximum permissible imep of 197 lb per sq in. and benzene, a maximum permissible imep of only 161 lb per sq in. Furthermore, the S-1 did not preignite, but was limited by knock. Fig. 13 shows that, despite the fact that S-1 permitted a higher imep, the thermal-plug temperature reading was slightly higher for benzene than it was for S-1. The data, together with those for toluene and methanol, are tabulated in Table 3.

In general, these data are in accord with those determined in the NACA laboratories. The data show that, from the standpoint of preignition, fuels cannot be compared on the



■ Fig. 14 – Effect of imep on temperature plug reading for different fuels (data from Heron and Gillig[4]) – Engine conditions same as in Fig. 12

basis of the preignition temperatures alone but that the effect of the fuels on the engine temperatures must also be considered. An indication of the effect of the fuel on the engine temperature is shown in Fig. 14, in which the imep's for several fuels are plotted against the corresponding thermal-plug temperature, the data again being taken from Reference 4. The blends of S-1 and M-1 and of S-1 plus tetraethyl lead show a steady increase in thermal-plug temperature with imep, the points lying along a smooth curve. For a given imep, either benzene or toluene had a thermal-plug temperature considerably in excess of that for the S-1 blends (as indicated by the arrows). This difference in engine temperature for the same imep is a result of the different thermal characteristics of the different fuels and is therefore an inherent characteristic of the fuels. This temperature difference is shown in the calculated peak combustion temperatures for the fuels, Fig. 15 (from Reference 5) and also in the effect of variable compression ratio on the thermal-plug temperature, Fig. 16. (The data presented in Figs. 16 and 17 were obtained by Harold M. Trimble of the Phillips Petroleum Co. The author expresses his appreciation to Mr. Trimble for permission to publish these data.) The thermal-plug temperatures for the three fuels shown in Fig. 16 are in the same order as those for the same three fuels in Fig. 14. Fig. 16 indicates that the mixture of iso-octane and 6 ml of tetraethyl lead started to knock at a compression ratio of 7.9:1 and that the toluene preignited (probably) at a compression ratio of 6.9:1. Benzene shows no indication of either knock or preignition as indicated by the fact that the temperature curve remains a straight line. The fact that benzene shows no signs of fuel failure at a compression ratio considerably above that at which toluene failed is not in agreement with the data presented in Figs. 10 and 14. It is apparent that disagreements of this sort indicate the need of much additional data on preignition.

[5] See University of Illinois Bulletin No. 139, March 17, 1934: "An Investigation of the Maximum Temperatures Attainable in the Combustion of Gaseous and Liquid Fuels," by G. A. Goodenough and G. T. Felbeck.

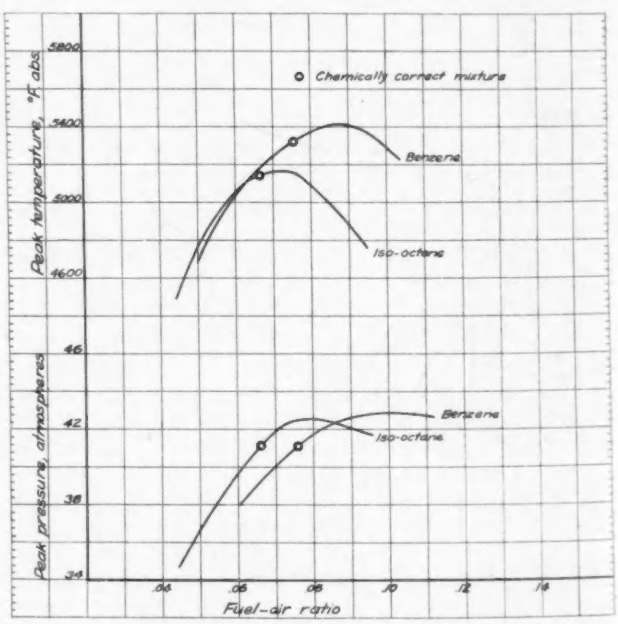
Table 3 – Data on Knocking and Preigniting Fuels

Fuel	Maximum permissible boost pressure, in. hg	Imep, lb per sq in.	Thermal plug Temperature, F	Cause of fuel failure
Toluene	45.8	180	898	after ignition
Benzene	40.3	161	892	after ignition
S-1	47.9	197	888	light knock
Methanol	35.8	144	822	after ignition

We have shown in this analysis on preignition that, in rating a fuel for this characteristic, the temperature at which the fuel preignites does not seem to vary much with the engine operating conditions. We have shown that the engine factors which cause failure with a preigniting fuel are those factors which tend to make the engine run hotter, that is, which tend to make the cylinder and combustion-chamber surfaces run hotter. We have shown that, because different fuels result in different engine temperatures for the same engine output, this effect of fuel on engine temperatures must be considered in rating the preignition characteristics of the fuels. We have also shown that we do not really know much about preignition and that there is a great deal of work to be done on this subject.

If preignition can be avoided, there is reason to believe, based on present data, that considerable improvements can be obtained by using those fuels which are prone to preignite in our present engines, but which have shown remarkably high antiknock characteristics.

Preignition can be more dangerous than knock. With

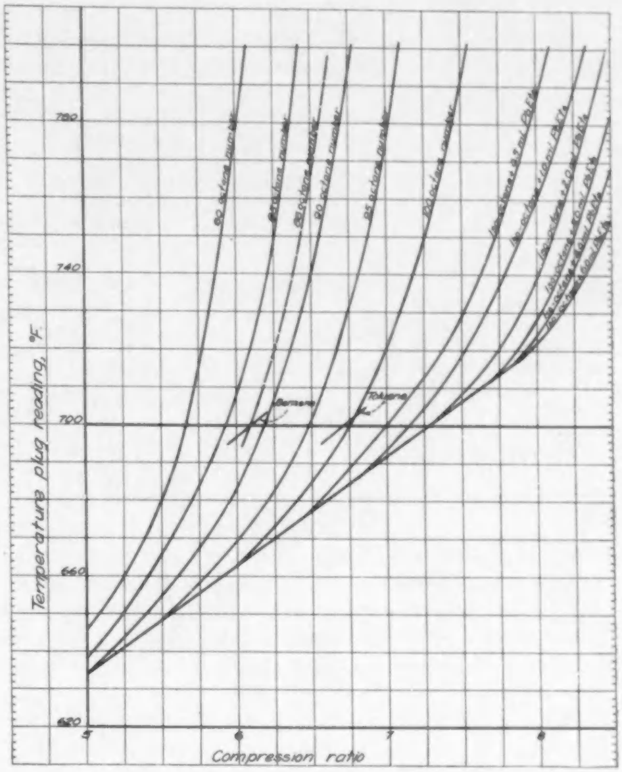


■ Fig. 15 – Effect of fuel-air ratio on calculated peak combustion temperatures and pressures for iso-octane and for benzene (data from Goodenough[5])

some fuels, notably benzene, when preignition starts, the charge ignites earlier on each cycle with such rapidity that engine failure can result before the fuel supply is cut off. This possibility of a critical condition arising with little or no warning justifies such fuels being ruled out until we know how to eliminate or at least control preignition. It also justifies fuel specifications which give preigniting fuels a low "octane number" even though the method of rating has little physical basis. It is hoped that through a more thorough understanding of preignition we will be able to prepare fuel specifications that will be more certain of success and more in accord with the facts which control preignition.

▪ Maximum Permissible Temperature Method

In the section on knock, the method of determining the octane number by means of the maximum permissible compression ratio has been discussed. In this case the fuel is rated at the compression ratio which causes the fuel to knock or to preignite or after-ignite. The system that has been used more generally to rate aircraft-engine fuels (although not officially recognized by the Society of Automotive Engineers) has been a modification of the maximum permissible compression ratio determination. In this method of fuel rating (known generally as the Army method) the fuels are rated at some predetermined temperature within the engine cylinder. This temperature is indicated by the thermocouple in the thermal plug already referred to. The procedure is briefly as follows: Each fuel is utilized in a variable-compression-ratio engine. The compression ratio of the engine is increased until the temperature reading of the thermal plug reaches a certain predetermined value. The octane number of the fuel is defined as the percentage of iso-octane in normal heptane

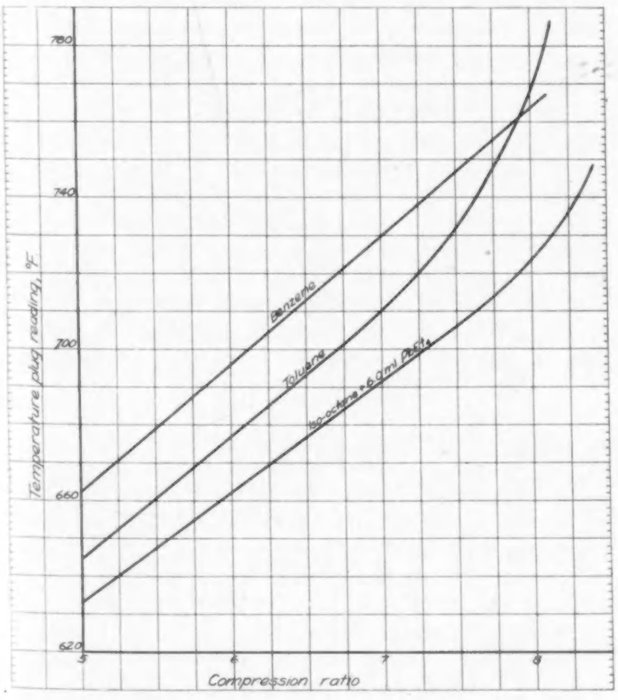


▪ Fig. 17 – Effect of compression ratio on temperature plug readings of several fuels – Army antiknock test engine (data from Trimble)

which, at this same compression ratio, gives the same standard temperature.

In Fig. 17 are presented the thermal-plug reading curves shown in Fig. 16 for iso-octane plus 6.0 ml of tetraethyl lead, the curves for different octane number reference fuels, the curves for different lead additions to iso-octane, and short sections of the curves for benzene and toluene. The curves for the reference fuels and for the iso-octane plus different lead concentrations all join the curve for iso-octane plus 6.0 ml of tetraethyl lead. The point at which the curve for each fuel joins the curve for iso-octane plus 6.0 ml of tetraethyl lead is presumably the compression ratio at which that particular fuel started to knock. These compression ratios are tabulated in the second column of Table 4. Neither toluene nor benzene have thermal-plug temperature curves which join the curve for the reference fuels as was shown in Fig. 16. The compression ratio of 6.9 given for toluene is that ratio at which the temperature curve for toluene ceases to be a straight line. (See Fig. 16.)

In rating fuels by the maximum permissible temperature method, the standard thermal-plug temperature first has to be determined. In the Army method, this temperature is defined as the thermal-plug reading at that compression ratio at which an 88-octane reference fuel gives the same thermal-plug reading as does benzene. The intersection of the curves for these two fuels determines the "standard temperature." For these particular data, this temperature is 700 F. It is noted that, by this method of fuel rating, benzene is arbitrarily assigned an octane rating of 88. In assigning this value to benzene, the fuel is neither knocking nor preigniting. The value is based presumably on service use of benzene. That is, in service operation it has



▪ Fig. 16 – Effect of compression ratio on temperature plug reading for benzene, toluene, and iso-octane plus 6.0 ml tetraethyl lead, (data from Trimble) – Army antiknock test engine

Table 4 – Sample Data for Rating Fuels by Maximum Permissible Temperature Method

1	2	3	4	5	6	7
Fuel	Compression ratio at which fuel first knocks	Temperature-plug reading at compression ratio at which fuel first knocks, F	Compression ratio at which fuel is rated, i. e., at which thermal-plug reading is 700 F	Non-knocking thermal-plug temperature for compression ratio at which fuel is rated, F	Thermal-plug temperature at which fuel is rated, F	Temperature increase caused by knock, F
90 octane number	5.0	634	6.2	668	700	32
95 octane number	5.5	647	6.5	677	700	23
100 octane number	6.0	663	6.8	684	700	16
Iso-octane + 0.5 ml $PbEt_4$	6.5	676	7.0	692	700	8
Iso-octane + 1.0 ml $PbEt_4$	6.9	688	7.2	696	700	4
Iso-octane + 2.0 ml $PbEt_4$	7.3	700	7.3	700	700	0
Iso-octane + 3.0 ml $PbEt_4$	7.1	712	.....	.....	.....	....
Toluene	6.9	706*	6.7*	700	700	....

* Probably preignition.

been found that it is not safe to use benzene in those engines which require a fuel of greater than 88 octane number. Each fuel is now rated at that compression ratio at which the fuel gives a thermal-plug reading of 700 F. In the fourth column of Table 4 are tabulated the compression ratios at which the thermal-plug readings for each of the fuels cross the 700 F line. In the fifth column are given the thermal-plug readings for the non-knocking S-M blends at the same compression ratios. The seventh column shows the increase in thermal-plug temperature caused by the knock at the rating compression ratio. The values in this column are equal to 700 F minus the temperature values in column 5.

The curve for toluene crosses the 700 F line at a compression ratio of 6.7, giving it an octane number of 99. As was the case with benzene, the toluene was neither knocking nor preigniting at this compression ratio. The complete curve for toluene in Fig. 16 indicates, as has been stated, that it started to fail at a compression ratio of 6.9. The curve for the non-knocking iso-octane fuels crosses the standard temperature line at a compression ratio of 7.3 which is also the ratio at which the iso-octane plus 2.0 ml of tetraethyl lead leaves the non-knocking base curve. Any fuel that can be operated at a compression ratio of 7.3 or greater without knocking or preigniting will be rated as equivalent to iso-octane plus 2.0 ml of tetraethyl lead. For this reason the rating of such fuels will not be an indication of their combustion characteristics. As a matter of fact, the particular conditions of the Army method are recognized to be limited to fuels of lower antiknock value than about iso-octane plus 0.5 ml of tetraethyl lead, the accuracy of the method being poor in the range from 0.5 ml to 2.0 ml of tetraethyl lead.

Using this method rates the fuels at successively higher degrees of knock or of preignition as the octane number is decreased. This fact is shown in the last column of Table 4, in which it is seen that with 1.0 ml of tetraethyl lead in the fuel the temperature increase caused by knock was 4 F whereas, with 90-octane fuel the increase was 32 F. In addition, fuels that give a higher engine temperature than do the iso-octane blends because of differences in thermal characteristics, such as benzene and toluene, are penalized because of these higher temperatures. If it is desired to have this property of the fuels included in a method of knock rating, the maximum permissible temperature method is an effective way of doing it. The chief objection to the method is that it is limited in the maximum rating which it can give to any fuel and the fact that, as this limit is approached, the accuracy of the method decreases. In the range from 100 octane number to 100 plus 1.0 ml of tetraethyl lead it is difficult to determine the fuel rating to a closer limit than ±0.5 ml of tetraethyl lead. Fig. 7 has shown that this variation permits the same variation in boost pressure as does ±7 octane numbers in the range from 90 to 100 octane number.

The maximum rating limit of this method can be increased by increasing the standard temperature. This increase is apt to be accompanied by an increase in the rating of benzene, toluene, and similar fuels. This difficulty can be partly overcome by operating the engine at conditions which tend to increase the wall temperatures of the combustion chamber without increasing the gas temperature or at least increasing the gas temperature to a lesser extent than the wall temperature. These conditions mean a high engine speed, a high cylinder temperature, and a low inlet air temperature. By so increasing the wall temperatures, the preignition characteristics of the fuels are accentuated. If experience has shown that these preigniting fuels fail in service operation, the accentuation of this characteristic in the rating method is justified. If a rating method that accurately evaluates preignition can be devised, the problem of fuel rating will be considerably clarified. Until that time, we must depend on existing rating methods and be guided by the primary requisite that no fuel will be permitted to be used in an engine which may result in failure of that engine.

Service Fuel-Rating Tests

We have discussed the methods by which fuels are rated in the laboratory. In service tests, that is, in the full-scale engine, fuels are rated by another method[6]. In this case, the engine is run under specified conditions of speed, temperature, and inlet pressure. The engine is operated on the fuel to be rated. Starting with a rich mixture, the mixture ratio is leaned until the engine shows signs of abnormal combustion. These signs consist generally of a rapid increase in the cylinder temperatures, a marked change in the appearance of the exhaust gas, or continued firing when the ignition switch is cut. The abnormal

[6] See SAE Transactions, May, 1936, pp. 161-175: "Rating Aviation Fuels in Full-Scale Aircraft Engines," by C. B. Veal.

combustion may be either preignition or knock. Having determined the fuel flow in gallons per hour at which the fuel failed, the blend of reference fuels is determined which fails at this same rate of fuel flow. The octane number of the fuel being tested is the octane number of the reference blends which showed indications of failure at the same rate of fuel flow. The chief difference between the laboratory rating method and the service rating method is the fact that, in the former, the rating is done by determining the maximum permissible compression ratio of the engine whereas, in the latter, the rating is done by determining the minimum permissible mixture ratio. Unfortunately, sufficient data have not been published to permit us to analyze the effects of fuel-air ratio on the conditions of the end gas that cause knock. We do know that knock occurs first at a fuel-air ratio between that for maximum economy and that for maximum power. The same statement is probably also true of preignition. In the service operation of an aircraft engine we are interested in the maximum permissible boost pressure under take-off conditions (that is, with a rich mixture), in the minimum permissible fuel consumption under cruising conditions, and in the effect of inlet mixture temperature, and of engine temperature under both these conditions. The interest in using a wider speed range between take-off and cruising conditions has further complicated the problem. Information on the combustion characteristics of representative fuels under these different operating conditions would be of considerable help in determining the possibility of specifying whether or not in relation to these fuels a "most severe condition" can be specified. In analyzing fuel rating methods and in proposing fuel rating methods we need data on the effect of these different variables on those factors which control knock and which control preignition. It seems probable that, as far as knock is concerned, the effects of engine temperature and of inlet mixture temperature can be expressed as a single variable. It may be that engine speed can also be included with these two temperatures. We know that one effect of fuel-air ratio is its effect on the end-gas conditions. But we do not know whether or not the effect of fuel-air ratio for all fuels can be predicted from the effect of temperature on the knock characteristics of the fuels. It is interesting to note that, because in service operation of high-powered aircraft engines neither the spark advance nor the compression ratio is changed, the fuels cannot be rated under the same conditions that are used in the laboratory when the maximum permissible compression ratio is determined.

■ Conclusion

In this paper it has been stressed that the knocking characteristics of a fuel cannot be adequately expressed by a single value because whether or not a fuel knocks depends on the interrelation of two factors, the end-gas density and the end-gas temperature. For this reason, if a single knock rating is to be used, a variation of actual service values from this laboratory value is unavoidable. It then becomes necessary to determine whether or not this variation is permissible. Edgar[3] has discussed the advantages and disadvantages of an average knock rating; consequently this phase of the problem need not be gone into here. There does not seem to be much hope of decreasing the variation between service values and a laboratory value through choice of a single set of operating conditions for the laboratory method. For research work on future fuel and engine development, it is necessary that a method of fuel rating be devised which expresses the basic knock characteristics of each fuel. In developing such a method, it is probable that some such procedure as that now used in the NACA laboratories will be required. It is necessary that, in developing such a method, one variable and only one be varied at a time. If this practice is not followed, the interpretation of the data becomes extremely difficult. There do not seem to be data available to show that a single set of engine conditions can be used in aircraft fuel rating which will represent the most severe condition for all fuels. An investigation to determine whether or not such a condition exists is needed.

In regard to preignition, the paper has emphasized the necessity of considering this phenomenon to be separate from that of knock and emphasized the difficulties to be encountered if a single method of fuel rating is to express both the knocking and the preignition characteristics of a fuel. Unfortunately, there are many more data needed on preignition before an adequate method of rating this characteristic can be determined. Until such data are available, present rating methods must suffice, but it is important that, in using these methods, we realize fully the relation or lack of relation between the factors which we are measuring.

Heron has stated[7] that "If cracked gasolines, branched-chain olefines, aromatics, alcohols, ethers, and ketones with or without tetraethyl lead or aniline additions were all permitted components of aviation fuels of 83 octane number or more, the situation with respect to full-scale engine knocking behavior would probably become chaotic." In present fuel research there is an increased interest in some of the fuels listed in this quotation. Adequate methods of rating these fuels must be devised. And such rating methods must not only express the combustion characteristics of the fuels, but must also express the conditions of engine operation under which the fuels can be used most advantageously. With the emphasis that is now being placed on both higher specific outputs and lower specific fuel consumptions, it is necessary that our data determine those facts which will lead to this increased engine performance. We must also continually guard ourselves against fuel-rating methods that tend to stifle rather than to foster progress.

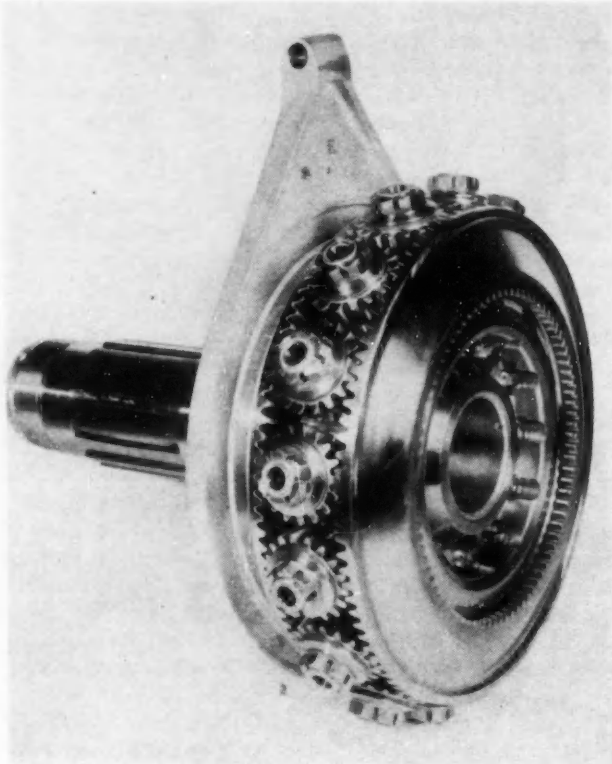
[7] See the *Journal of the Aeronautical Sciences*, Vol. 5, No. 12, pp. 463-479: "Aircraft Fuels," by S. D. Heron and Harold A. Beatty.

Requirements of Earth-Moving Equipment

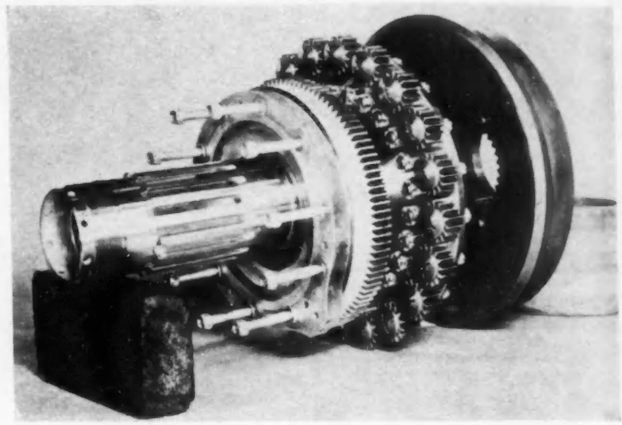
IN earth moving, torque and shock take precedence over horsepower and speed. This brings increased demands upon gears, transmissions, and bearings as well as on frames and other structural members. Earth must be moved up hill or down dale as the case may be, over rough and rocky ground, through swamps and mud, or over the abrasive sands of the desert. Furthermore, construction equipment must be capable of continuous operation at full load for indefinite periods of time, in freezing weather, or in the intense heat of a tropical sun. And, last but not least, it is generally operated by a type of labor that is capable of inflicting an unusual amount of abuse upon it.

Excerpts from the paper: "Earth-Moving Equipment and the Engineer," by F. A. Nikirk, Caterpillar Tractor Co., presented at the National Tractor Meeting of the Society, Milwaukee, Wis., Sept. 25, 1940.

Aircraft-Engine REDUCTION G



■ Fig. 1 – Bevel-gear type of reduction gear developed in multi-pinion form by the Wright Aeronautical Corp.



■ Fig. 2 – Wright 16:7 and 16:9 interchangeable spur-planetary-type reduction gear

IN simple spur planetary reduction gears, ratios near 2:1 give relatively small planet pinion diameters and, accordingly, permit the use of an unusually large number of pinions. The Wright 20-pinion 16:7 and 16:9 gears, now developed to 2000 hp, are described to illustrate the advantages of the multi-pinion design. As engine power outputs, rpm, and propeller diameters increase, greater reduction ratios are required for which the simple spur planetary type is not so suitable.

The design and development of Wright Aeronautical Corp. built-in torque meters is discussed briefly.

■ General Considerations

The design development of all the major units of aeronautical powerplants, including reduction gears, has been influenced largely by the continuing trend towards higher power outputs. The increase in duty on reduction gears is the result of both increased crankshaft torque and increase in gear ratios which accompanies higher engine rpm and the large propeller diameters necessary for the increased horsepower absorption.

For example, in the Wright Duplex engine rated at 2000 hp, take-off, the crankshaft torque is in the 60,000 in-lb range and the gear ratio multiplies this torque at the propeller by $\frac{16}{7}$. It appears that the demand for reduction gears of 250,000 in-lb propeller torque is not far off.

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Nov. 1, 1940.]

Reduction-gear development has followed two broad lines:

(a) For V engines, where it is desired to set the propeller axis eccentrically to the crankshaft to bring the propeller near the center of the frontal area, single-contact spur-type gears are still in use. The general European practice is to use external gears whereas the best-known U. S. example has the crankshaft pinion meshed with an annular gear on the propeller shaft.

(b) For radial engines, planetary gears, giving a concentric drive, are almost universal. These units started with the bevel-gear type developed abroad by Waseige and, in this country, at The Aeromarine Plane & Motor Co. in 1925. This type has been developed in multi-pinion form by the Wright Aeronautical Corp. for reverse rotation and 2:1 gearing (Fig. 1).

■ Multi-Pinion Gears

The simple planetary spur gear comprising planet pinions engaged between a sun and a ring gear is impossible for 2:1 ratio of forward rotation whereat the pinions have zero size. For ratios near 2:1 as, for example, in the Wright interchangeable 16:7 and 16:9 gears (Fig. 2), the pinions become of very small diameter and rotate at proportionately high speed. These particular pinions have a pitch diameter of 1 3/8 in. and run at over 8000 rpm on floating bushings of approximately 3/4 in. inside diameter. At first glance, such small pinions look extremely frail (they are smaller than any ball or roller bearing that could be considered for

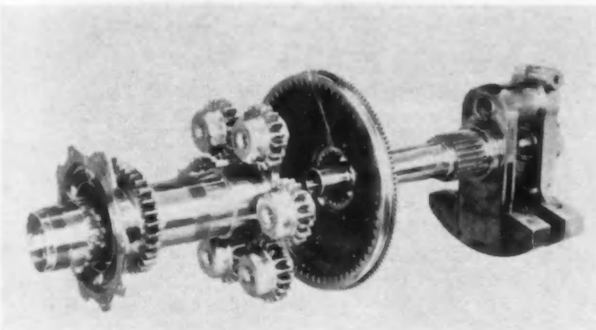
GEARS and TORQUE METERS

by ROLAND CHILTON
Consulting Engineer, Wright Aeronautical Corp.

AS power outputs, crankshaft speeds, and propeller diameters increase, greater reduction ratios are required for which the simple spur planetary type of reduction gear is not so suitable, the author points out. Some of the indicated future requirements for propeller drive discussed in this paper are increased reduction ratios; compound reduction gears interchangeable for change of speed without change of gear housing; provision of two-speed gears; reverse rotating coaxial propellers; alternate right- and left-hand propeller rotation; remote drives permitting buried engines; and torque meters.

In simple spur planetary reduction gears, Mr. Chilton explains, ratios near 2:1 give relatively small planet pinion diameters and permit the use of an unusually large number of pinions. Design and development of the Wright 20-pinion reduction gears, now rated at 2000 hp, are described to illustrate the advantages of the multi-pinion design.

Evolution of the torque meter is traced, and a modern successful built-in design is described.



▪ Fig. 3 – Earlier 16:11 six-pinion type of reduction gear

their mounting) and hence simple planetary gears have, in the past, been regarded as unsuitable for such ratios as 16:7 and 16:9. However, immediately when we accept pinions spaced as closely as their outside diameter permits, these ratios afford 20 pinions, giving a more compact and lighter design than do the ratios previously regarded as most suitable for simple planetary gears. The load per tooth contact is obviously inversely proportional to the number of pinions, and in this case, at 2000 hp, each pinion is transmitting only 100 hp, assuming even tooth load distribution between the pinions. This design gives conservative tooth and bearing loadings on these small parts.

When the outer ring gear is secured to the crankshaft and the inner gear fixed to the nose, the ratio is 16:9. By transposing the parts and driving from the inner gear and anchoring the outer or ring gear to the nose, the ratio with the same tooth numbers is 16:7. The latter set-up increases the tooth loading because the smaller member is the driver, but this gear has been endurance-tested at 2000 hp. Fortunately, these ratios are in demand for current production engines but, with increasing crankshaft speeds and higher power outputs, higher ratios than 16:7 will be required. The higher ratios increase pinion size, reduce the number of pinions and the diameter of the driving gear so that we soon reach ratios for which the simple spur-type planetary gear is unsuitable.

Most of the development work required to get durability under continuous operation at take-off power (2000 hp) was in the pinion bushings, seizure of which may cause heat cracks in the pinions and complete gear failure. Until this bearing unreliability was overcome, it was difficult to refute the contention that unequal load distribution was the cause of the failures, in spite of other evidence to the contrary as follows:

▪ Tooth Load Distribution

This argument is a good example of the difficulty of assigning valid reasons to justify conventional practice. It happens that the first successful planetary reduction gears had three pinions only, thus developing the theory that the tooth load could not be distributed adequately if more pinions were used. This question resolves itself into:

(a) The tolerance in tooth and pinion-arm spacings that can be realized in production, and

(b) The load increment due to these errors, which is a function of the elastic deflection rate of the assembly.

The multi-pinion design so reduces the individual tooth loads as to give relatively flexible rings and pinions when the rim thicknesses are cut as far as is consistent with conservative stresses. In current practice, cumulative tooth spacing errors should not exceed a few ten thousandths and

our tolerances on the pinion journal spacings is ± 0.0005 in. between adjacent journals and ± 0.001 in. between any. The bushing thickness is also held within good manufacturing tolerances. When the first experimental gears were built, a torque rig was made to load the assembly by increments while checking the backlash at the individual pinions to determine at what percentage of operating torque the "loosest" pinion came to zero backlash, which would be an index of the load increment due to spacing errors. This device was discarded when it was found in the first sets of gears that all of the pinions could be brought to zero backlash by the muscular effort of one man on a 3-ft bar.

To facilitate accurate spacing, the trunnions are made integral with the carrier ring and they comprise cantilevers subject to bending deflections, in this case calculated at 0.0002 in. in the length of the tooth. This deflection would tend to overload the inboard end of the teeth if the sun and ring gear rims had uniform stiffness. However, to compensate for this, the thin ring gear has a heavy flange at its outboard end, stiffening that end against radial deflections and the disc of the sun gear is also offset to stiffen the outboard end of the teeth. These provisions have resulted in a satisfactory tooth contact pattern, the difference between the two ends of the teeth being negligible.

For comparison, the earlier 16:11 6-pinion type of gear is shown in Fig. 3. Here the individual tooth loads are much higher and the spider arms more flexible, to compensate for which the pinions have the peculiar re-entrant journal construction, the proportions for which were determined from load tests on sample assemblies, the stiffness relationship being changed until the mating teeth deflected in parallelism.

■ Mechanical Efficiency and Testing

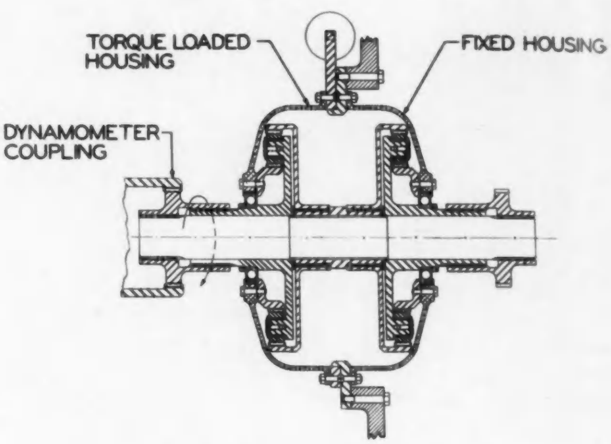
In development testing of the 20 pinion gears, two sets are coupled back to back (Hopkinson coupling) (Fig. 4). The propeller shafts of the two gear sets are connected rigidly by a shaft which passes through a tube similarly connecting the crankshaft gears. The other gears are rigid with respective gear housings, one of which is fixed to the test stand and the other loaded to the desired test torque through a torque arm and hydraulic jack. The propeller shafts are driven through a small dynamometer which has only to supply the friction torque in spite of the fact that the torque times rpm may be at the value for 2000 hp. In this arrangement friction losses are measured directly by the dynamometer torque thus avoiding the serious errors involved in measuring input and output horsepower for subtraction to ascertain the friction loss. Actually, these gears run about 99.1% efficient (0.9% friction loss) at rated power. It is obvious that a ½% error in reading input and output torques by the old method could give results indicating more than 100% efficiency. While friction losses of around 1% may sound small, it is remarked, that in the case of a 2000-hp engine, this figure requires 20 hp heat dissipation in the oil cooler to look after the propeller gears alone.

■ Weights

The compactness of this gear results in a light gear housing; in fact, much of the bulk of this member is due to the propeller thrust bearing, the length occupied by the hydro-propeller oil seals and, in some cases, due to the inclusion of the tappet section in the nose piece. The weight of the gearing, per se for 2000 hp is 51 lb, neglecting the weight of the propeller shaft since this element is necessary in a direct-drive engine (although it may there be of smaller size).

■ Lubrication

The floating pinion bushings are pressure-fed and current 20 pinion gears have an oil flow of around 20 lb per min. In the early tests, pitting of the teeth, which could



■ Fig. 4 – Diagram of gear test rig

not be correlated with power factor or time on test, was experienced. Stress pick-ups applied to the stationary gear rim showed very large stress variation at tooth frequency. This was attributed to oil trapping within the ring gear and was reduced to about 1/10 by merely drilling large drainage holes in the cylindrical part of the gear. Each tooth enters and leaves the mating tooth space in less than $\frac{1}{2000}$ sec for which condition good scavenging is essential to avoid overloading due to hydrodynamic action. The phenomenon of tooth pitting due to excess oil has been found and reported previously; in fact, the use of oil squirts on the incoming side of high-speed gears appears usually to produce this trouble.

During this development, difficulty was experienced with the floating pinion bushings and a wide variety of detail designs were tried. The current bushings are in two pieces providing a flange at each end of the pinion and a circumferential oil passage at the middle. These bushings are good for over 75 hr of continuous testing at take-off hp (2000) both full-scale or on the test rig, and complete gears have passed type tests on Wright 9-, 14-, and 18-cyl engines.

■ New Requirements

It is evident that the propeller drive is due to become more elaborate and diversified. Some of the indicated new requirements are:

(a) Increased Reduction Ratios – These requests are covering a very wide range and, unfortunately, by very small increments. Each airplane design agency calculates for each airplane type a theoretical optimum gear ratio. These

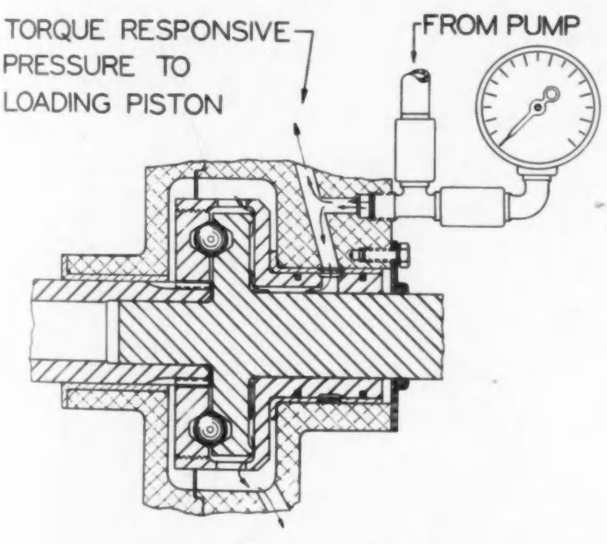
calculations indicate a wide diversity of opinion for apparently similar conditions and to meet all initial requests would require a special gear ratio for almost every new installation. This means endurance-testing in each ratio with each type of propeller which may be specified. There are so many regimes of flight that the figures arrived at by one calculator for optimum gear ratio on a given airplane are apt to vary widely from the recommendations of another. If engine manufacturers had to supply gear ratios only in increments as wide as this variation in opinion, the problem would be greatly simplified.

This brings up one of the disadvantages of the simple planetary type of gear, that is, a small change in ratio requires wide changes in the size of the sun gear, ring gear, pinions and/or carriers. Current W.A.C. ratios include the following:

16:11	– 1.4545	– 0.6875
3:2	– 1.5000	– 0.666
16:9	– 1.7777	– 0.5625
16:7	– 2.2857	– 0.4375

The first two ratios may be bracketed as equivalent gears wherein the choice is dictated by mechanical considerations not concerned with propeller efficiency.

(b) In connection with new engine designs, ratios from 2:1 to over 4:1 have been requested and an attempt is being made to stabilize the requirements in the maximum acceptable increments. Present indications are, however, that these requirements will force a compound type of gear wherein the ratio can be changed without changing the bearing housings or shafts. The serious factor of the number of engine and propeller proof tests required still remains.



■ Fig. 5 – Torque responsive coupling

(c) Over-speed for Take-off and Climb – There is a demand for two-speed gears for high-speed airplanes in order to maintain sufficient tip speed along the helix at low forward velocity. When we are dealing with 2000 hp and upwards, this provision will neither simplify nor lighten the gear.

(d) Reverse rotating coaxial propellers also must be provided for in the future.

(e) Alternate right and left-hand propeller rotation, to eliminate torque effects in multi-engine planes, has been provided for by interchangeable reverse rotation gears in certain models.

(f) Remote drives permitting buried engines have been urged from many sources. This is probably the most difficult requirement of all as it usually involves two right-angle drives on each side. Most layouts indicate two sets of bevel gears in series on each side. These single-contact gears are of relatively large diameter and great face width which, with the necessary housings and shafts, will make the gearing weigh several times as much as the current multiple-contact gears. Neglecting added propeller weights, present multiple-contact gears cost about 1/40 lb/hp. The question is, how much is it worth while to pay for remote drives to get engines in the hull? A consensus of opinion on this would be of great value; for example, does anyone think that remote drive would be worth 1/3 lb/hp added weight?

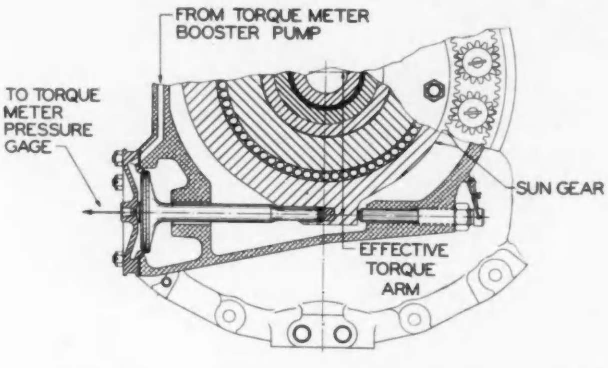
Summarizing, future requirements are visualized as follows: Reduction-gear design should be so flexible as to provide the following range of features by selection of the appropriate interchangeable gears:

(a) Ratio range 2:1 – 5:1
(b) Right-hand rotating propeller
or
(c) Left-hand rotating propeller
or
(d) Dual coaxial reverse-rotating propellers
with
(e) Torque meter
and
(f) Two-speed gear affording from 20 to 80% over-speed for take-off.

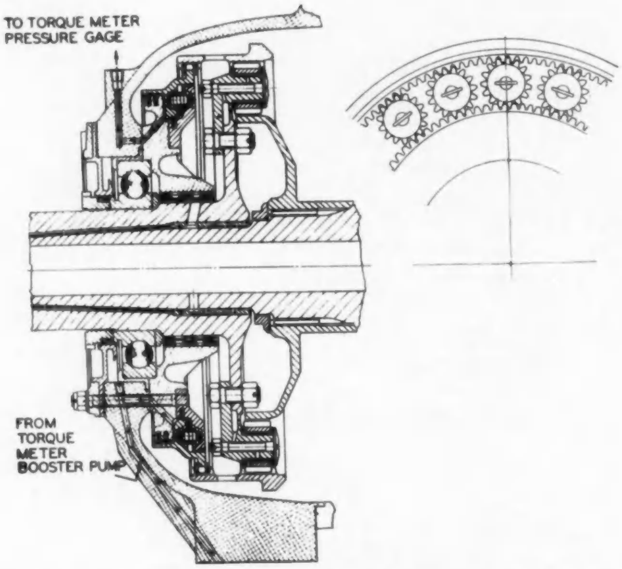
This is a formidable list of requirements, especially considering that the horsepower capacity already has reached 2000. However, as in other cases of insistent demand in aeronautics, the requirements will be met in spite of the technical difficulties. The airplane designer must resign himself to a weight penalty.

■ Torque Meters

The fixed sun gear of the planetary type gear affords a convenient basis for a torque meter, and the first widely used device was so applied by Pratt & Whitney. The genesis of the Wright "flow-type" torque meter dates back to designs aimed at the reverse problem in connection with variable-speed transmissions. In this case, it was desired to load the power-transmitting roller contacts proportionately to the torque transmitted by a shaft rotating relatively to the loading piston. One early design (1934) is shown in Fig. 5. In this construction a "torque-responsive" coupling is included in the shaft. This coupling consists of discs secured to respective shaft elements and connected by a number of balls engaging conical pockets in each disc, thus imposing an axial separating component between the discs proportional to the torque. This coupling is given slight backlash against a cut-off edge or valve seat, the area within the seat comprising a "piston" to which is fed the entire delivery of a very small oil pump. In effect, the torque-responsive coupling becomes the relief valve of the pump and, accordingly, controls the pump pressure to be proportional to the torque which may, accordingly, be read on



■ Fig. 6 – Wright Aeronautical Corp. tangential-type torque meter



■ Fig. 7 – Wright Aeronautical Corp. 20-pinion reduction gear with concentric torque meter

a conventional pressure gage. One advantage of this system is its easy applicability to a rotating part since, by applying oil seals to a shaft bearing, the pressure may be transmitted from a stationary housing to a rotating torque metering coupling.

Experience with this "butt" type cell, wherein the metering or cut-off edges may actually contact each other under conditions of torsional vibration, showed it to give low values for torque reading at torsional resonance in aircraft engines and the cell was, accordingly, modified so as to comprise an actual piston free to over-run the cut-off slot. An example is shown in Fig. 6 as applied to Cyclone engines. This early design is not concentric but has a torque arm on the normally fixed sun gear engaging the thrust rod of a tangentially disposed hydraulic torque-meter cell. It will be noted that the sun gear is supported on an anti-friction bearing which minimizes errors from friction due to the lateral torque arm reactions on the sun-gear mounting.

Fig. 7 illustrates the concentric type such as used in the Wright Duplex engine. The principle is identical with the transmission loading device first described although, in this case, the assembly is non-rotating. This concentric type has the advantage of eliminating local lateral loads on the gear housings and permits the use of large piston areas giving moderate hydraulic pressure and, by appropriate calculation for the slope of the ball pockets, the mep constant may conveniently be made unity for the particular engine displacement.

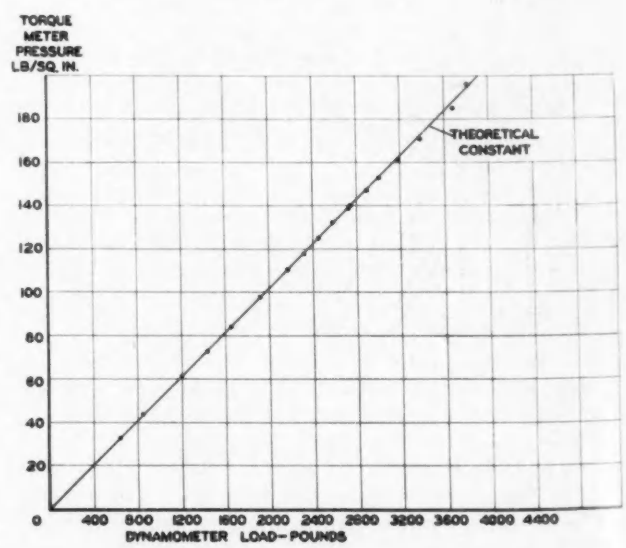
These torque meters introduce about 0.040 in. backlash into the driving system in the event of pump failure, but no trouble has been experienced from this source to date. In the concentric type any slight rotational yield involves fore-and-aft movement, and it is interesting to note that this axial "float" is taken at the planetary pinion tooth contacts. In static rig tests, it was necessary to permit the propeller shaft to float axially on anti-friction means to eliminate the frictional restraint at the (stationary) teeth. A typical calibration curve, Fig. 8, falls within ½ of 1% of the calculated torque/gage-pressure relationship. It is interesting to note that this accuracy is obtained in the running engine with a standard thrust bearing, indicating that the axial restraint at the gear teeth is very small when these are running.

A further advantage of the "flow" type cell, wherein the entire oil quantity from a small pump issues from the metering edge at all times, is that incidental leaks do not affect the system which may, accordingly, be connected to operate controls where torque-responsive action is desired. For example, a manifold pressure regulator giving constant bmep would be a simple adjunct to this type of torque meter.

A sealed volume type of diaphragm cell was abandoned early in this development because minute leakage affected the calibration and mechanical failure of the diaphragms was found to be a major difficulty under the vibration conditions in aircraft engines.

■ Operating Experience

The traditional practice in rating engines has been to base the power curves on dynamometer calibrations. Endurance testing with flight propellers or test clubs was on



■ Fig. 8 – Torque-meter calibration curve

the assumption that the dynamometer calibration would be reproduced at the same rpm and manifold pressure. It is obvious that this standard practice involved several errors as engine horsepower varies with mixture-ratio and cylinder temperatures for which no corrections have been applied, and it is known that the carburetor air temperature correction is approximate. The practice of basing dynamometer calibrations on best-power mixture strength and running the high power part of endurance test with a substantially enriched mixture has been misleading and, for some months after torque meters were first applied to propeller runs, the test reports often stated that the torque meter was reading low. However, by controlling the mixture strength, cylinder temperatures, and so on, to correspond with the conditions under which the dynamometer calibration was made, these power curves are reproduced on the propeller stands and, as a result of this indoctrination, any variations from the official dynamometer power curves are now dealt with by reproducing the actual dynamometer conditions, the torque meter being regarded as accurate.

These considerations apply with equal force to flight testing. The old practice when a prototype airplane failed to meet its speed guarantees was to change engines on the theory that they must be low in power. The tying up of two additional sets of specially calibrated engines on this account has not been uncommon but, when torque meters are used, the question as to whether the horses are only ponies, or headwinds have been built into the airplane is now promptly settled. When thrust meters are available, propeller efficiencies will be directly measurable in flight and the corresponding debate between the propeller, plane, and engine manufacturers will be composed.

Problems of National Defense

ON the basis of headlines coming out of Washington, one would think that the United States does not have at this moment *any* plants or machinery capable of turning out defense equipment. One would think that we had to start from scratch and build from the very bottom a vast system of enormous new plants, filled with brand-new equipment.

Now I have no illusions as to the total production which may be required for national defense. For the entire program, of course, hundreds of new plants will be required. Of course, vast amounts of new machinery will be needed.

But this does not mean that every national defense plant must be enormous or new. Neither does it mean that all of the equipment needed to produce national defense products has to be new.

The danger lies right now that, in the anxiety to develop adequate defense facilities, we will build up an unnecessarily large supply of mammoth plants, and pile up an unnecessarily large inventory of new equipment – which, when the emergency is over, will represent a terrific load of excess plant and equipment.

In our desire to do this job in a big way, we may be overlooking the fact that a great deal of the job can, and should, be done by existing facilities.

So much for manufacturing facilities. Now let us take up the question of men.

We certainly do have a shortage of trained men. But we have no shortage of *men*. We have plenty of men – our shortage is in training.

What's the answer? Inventory and coordinate our training personnel. Marshal our training facilities in exactly the same way that we marshal our machinery and equipment facilities.

Every plant has some men who in a pinch can be spared for training. Let's put those men to work right now. It might mean a shade less production next week, but it will mean a great deal more production next month and next year.

Jobs in many a plant can be re-shuffled in such a way that the men who are more experienced can be freed for training. Many more who know how to teach machine operations are now working on automatic or semi-automatic jobs. Let's put younger and less experienced men on these automatic and semi-automatic jobs, and thereby release more fully skilled men for training activities.

Furthermore, all over this country there are older men, skilled mechanics, who during the depression years drifted out of industry into other occupations. These men should be located and brought back into industry. They should be put to work training the younger men.

Remember, real training can't be done in class rooms. It must be done on the floors of our industrial plants. The major responsibility for training lies upon industry, because the actual proof of a man's intelligence and capacity can only be determined on the factory floor.

But at the same time it seems to me that in connection with this question of man-power, the government, with the cooperation of industry, might take right now an extremely constructive step. I think the government might help to push men out of WPA into industry, while industry endeavors in so far as possible to train men who have been on WPA.

It is certainly doubtful as to whether industry will succeed in getting all the men it needs for mounting production from the annual output of high schools and colleges – especially in view of the conscription act. It is more than probable that industry must of necessity tap the reservoir represented by men who have for some years been on WPA or unemployed, or on relief.

It is true that on the basis of practical experience these men have proved slower to train than is the case with the boys just out of school. It is true that fewer of these men show an aptitude for machine operation than is the case with many younger men. Furthermore, it is true that during past months men on WPA or unemployed have to a certain extent already been surveyed by industry, and those fitted for factory jobs selected and put to work.

Nevertheless the fact remains that, just as we must put all our existing plants and equipment to work for national defense, we must likewise put all our existing man-power to work.

Excerpts from the paper: "Basic Industrial Problems of National Defense," by George T. Trundle, Jr., president, The Trundle Engineering Co., presented at a meeting of the Philadelphia Section of the Society, Philadelphia, Pa., Sept. 18, 1940.

RATING A
in F

THE Automotive Diesel Fuels Division is working, first, to clarify the issue of automotive diesel fuel specifications, and second, to determine those properties of analyses which influence engine operation and engine deposits. Primary membership on the committee is made up of engine men representing the four primary types of automotive diesel engines, namely: direct-injection or open-chamber; swirl-chamber; energy-cell; and pre-combustion chamber. Likewise represented are the petroleum industry, colleges and Government agencies participating in the tests; petroleum men to obtain and offer for sale the four selected reference fuels; and an instrumentation and procedure group set up to find ways and means of standardizing procedure and instrumentation.

Results reported in this paper include the influence of cetane, volatility, viscosity, and gravity of diesel fuels as they affect cold starting, smoothness, low-temperature starting, smoke, power output, fuel consumption, smell, and engine deposits. Four reference fuels were used in these studies.

Automotive Diesel Fuels Division Organization

The CFR Full-Scale Engine Group was organized for the purpose of:

1. Clarifying the issue of automotive diesel fuel specifications,
2. Determining those properties of analyses which influence:
   (a) Engine performance;
   (b) Engine deposits.

The group was formed with the primary membership consisting of engine men who represent the four essential classifications of automotive-type diesel engines:

A. Direct injection of open chamber;
B. Swirl chamber;
C. Energy cell;
D. Precombustion chamber.

It was so conceived that at least two engine men were selected to represent each engine type together with a representative from the U. S. Naval Experiment Station at Annapolis. These engine men were to conduct suggested and prescribed tests on multicylinder diesel engines of their own manufacture selected in characteristics within the following limits: bore $3\frac{1}{2}$ to $4\frac{1}{2}$ in.; number of cylinders, 4 to 6; speed, 1500 to 2000 rpm. These engines were to be operated on four fuels selected as reference fuels.

A secondary group was invited to consist of the representatives from the petroleum industry who have engines in their laboratories of the types of manufacture as represented by the engine men of the primary group, but these engines were acceptable in either single or multicylinder structures. The petroleum men were to run tests as prescribed for and by the engine men, to contribute supporting data to the respective representatives of the engines manufactured by the Engine Men Group.

A third group also was invited consisting of representatives from colleges and government agencies willing to test the reference fuels according to the prescribed tests and to contribute data to the Engine Men Group.

A Fuels Group was selected consisting of petroleum men to find ways and means for obtaining the four selected reference fuels. These fuels were to be made available and offered for sale to all the participants referred to in the foregoing.

An additional committee also was selected for the purpose of setting up instrumentation and procedure in order to standardize the methods of test employed by the various laboratories.

■ Meetings

The Peoria meeting, July 26, 1939, was called for the purpose of organizing the Full-Scale Engine Group. The factors selected by the Full-Scale Engine Group as essential to the investigation of the reference fuels were considered: (1) cetane, (2) volatility, (3) viscosity, (4) gravity; as to their effect on (1) starting, (2) smoothness, (3) low-temperature starting, (4) smoke, (5) power output, (6) fuel consumption, (7) smell, (8) engine deposits. The group decided on a suggested procedure of test for investigating the influences just enumerated on engine performance and on engine-deposit-forming characteristics.

A set of four reference fuels also was established and specifications subsequently formulated for obtaining these fuels. As an interim activity the details of the test procedure were clarified further by correspondence, and the reference fuels were made available by the Fuels Group and were stored and distributed by the Sinclair Refining Co., East Chicago, Ind. It is expected that these fuels will be available for quite some time in the future.

At the Detroit meeting on Jan. 24, 1940, of the Engine Group was indicated the continued desire for obtaining fuels and conducting tests. Many engine manufacturers and petroleum refineries had placed orders and started work on these reference fuels.

The Full-Scale Engine Procedure and Instrumentation Group was formed for the purpose of standardizing instruments and procedure and of assisting cooperating engine men in clearing up questions which may arise regarding matters of procedure and instrumentation.

The Chicago meeting on June 4, 1940, was held jointly by representatives of the engine men and petroleum men and the College Group. In all 20 representatives were present to support and submit data. A motion was made and agreed upon unanimously "that in the committee discussion the names of the various test engines be revealed where desirable and that in the report of the committee to the White Sulphur Springs meeting the types of engines be quoted." An admirable attitude of cooperation was

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 12, 1940.]

Automotive Diesel Fuels Full-Scale Engines

Report of Cooperative Fuel Research Committee

by C. G. A. ROSEN
Chairman, Automotive Diesel Fuels Division

shown by engine men and oil men in the free and open interchange and discussion of data which were pooled for the purpose of this report. Some of the data submitted were quite complete, whereas some are still in the formative stages of the program. All data apparently indicate a close correlation of the general tendencies exhibited by comparable tests on similar engines. The work has been started satisfactorily and, by the January meeting, a comprehensive report is to be expected. It is to be noted that the program, as organized at present, sets up a structure of agreement upon which to build further cooperation and testing.

■ Reports

To date reports have been received on engine testing conducted in 9 different laboratories employing 12 individual engines. Referring to the original engine classification these reports are divided as follows:

1. Direct-injection engine – four reports.
   Three reports on General Motors, Model 3-71, 3-cyl, 2-cycle, 4¼-in. bore, 5-in. stroke.
   One report, General Motors, Model 1-71, single-cylinder, 2-cycle, 4¼-in. bore, 5-in. stroke.
2. Swirl Chamber – four reports.
   One report, Hercules Model DKXB, 6-cyl, 4-cycle, 3½-in. bore, 4½-in. stroke.
   Three reports, Fairbanks, Morse Model 36-A, single-cylinder, 4-cycle, 4¼-in. bore, 6-in. stroke.
3. Energy Cell – three reports.
   Two reports, Dodge Model TKD-TR, 6-cyl, 4-cycle, 3¾-in. bore, 5-in. stroke.
   One report, Mack Lanova type, 6-cyl, 4-cycle, 4-in. bore, 5⅜-in. stroke.
4. Precombustion Chamber – one report.
   Caterpillar Model E48, 4-cyl, 4-cycle, 3¾-in. bore, 5-in. stroke.

In evaluating these data as submitted, an attempt will be made to exhibit the general tendencies which are revealed. It is to be considered dangerous to accept this valuation as final or in any way sufficiently conclusive to warrant considering tentative specifications. As the matter stands, it must be emphasized that certain engine discrepancies which appeared under some one particular condition or setting requires further test work on the part of the engine men to verify or disprove the few out-of-line findings. Going back to the original purpose of this committee, an attempt will be made to cull out the tendencies of the fuel factors of cetane, volatility, viscosity and gravity on engine performance and engine deposits. The work as it now stands does not permit of the sharp separation of such factors as viscosity and gravity. These two factors will be approximately evaluated as to their individual or joint prominence in affecting performance and deposits. At a later date we hope to have these points clarified.

The fuels themselves bear scrutiny as to their essential characteristics:

Fuel	Cetane No.	50% Distillate Points – Volatility, F	Viscosity, S.S.U.	Gravity, deg API
A	56.1	527	37	37.6
B	40.5	607	51	26.6
C	33.2	502	37.5	29.1
D	41.3	413	31	40.8
B + D	40.8	489	36	33.4
A + C	44.0	515	37	33.2

Some of the fuel factors appear to be difficult to tear apart. For the purposes of this initial program these four basic fuels have proved instructive and interesting. Blending of fuels (B + D) and fuels (A + C) offers further opportunity for study. In the future even addition agents may be of value in furthering the usefulness of these basic fuels.

It can be seen from this table that:

Fuel A – High Cetane, 56.1
Fuel C – Low Cetane, 33.2
Fuel D – High Volatility, 413 F
Fuel B – Low Volatility, 607 F
Fuel B – High Viscosity, 51 sec
Fuel D – Low Viscosity, 31 sec
Fuel D – High Gravity, 40.8
Fuel B – Low Gravity, 26.6

The general tendencies of these fuel factors are exhibited as follows:

Fuels A, Blend (A + C), and C exhibit influence of cetane in approximately the same distillation range.

Fuels B, Blend (B + D), and D exhibit influence of volatility in the same cetane range.

Fuels B, Blend (A + C), and D exhibit general influence of viscosity.

Fuels B and D exhibit approximately influence of gravity.

The reports submitted (written or oral or both) were made by representatives of:

1. Atlantic Refining Co.
2. Esso Laboratories
3. The Texas Co.
4. Mack Truck Co.
5. The Pure Oil Co.
6. University of Wisconsin
7. Shell Oil Co.
8. Caterpillar Tractor Co.
9. General Motors Corp.

The reports as submitted by the individual test laboratories are analyzed with due regard to the basic object of the Full-Scale Engine Group. The pertinent tendencies exhibited in these reports are grouped with respect to their bearing on fuel factors.

■ Influence of Cetane

In the General Motors diesel, fuels above 40 cetane show no apparent differences in fuel consumption at all speeds and loads. At light loads and high speeds 33-cetane fuel shows 20% greater consumption than the remaining three fuels. Under idle conditions 30% more fuel is required to maintain 600 rpm. On the low-cetane fuel engine roughness increased as cetane decreased. Starting quality varied with cetane number. Smoke varied somewhat in the inverse ratio to cetane number, but was more definitely affected by volatility. The high-cetane fuels provided better idling qualities, and high-cetane fuel is sweeter than low-cetane fuels. Higher volatility fuels with low cetane produced the most stink in the exhaust.

The Dodge diesel at full load exhibited only small differences in fuel consumption on fuels above 40 cetane. The 33-cetane fuel, however, showed 15% greater fuel consumption. One report indicates that there is some tendency for power output to vary directly with cetane. At light loads the fuel consumption shows the same tendencies as under full load with respect to cetane number. At high speeds the low-cetane fuel increased fuel consumption 60% at idling operation. Cetane influences starting in that higher cetane fuels are better starters. The smoke is affected inversely by cetane at low speeds whereas, at high speeds, volatility exhibits the more marked effect. Cetane also influences the time required to bring the engine up from a cold start to smooth running. Knock intensity is in inverse ratio to cetane.

The Hercules diesel is insensitive to cetane number below 1200 rpm at light load and below 1800 rpm at full load. Fuels of 40 cetane are 15 to 20% higher in consumption than 55-cetane fuel at idle and, at 2400 rpm, as much as 100% greater. The marked increase in consumption of the lower cetane fuels at high speed idle is accompanied by corresponding increases in smoke. The engine appears to require a 50-cetane fuel for good performance.

In the Mack Lanova diesel the power output and fuel consumption are relatively insensitive to cetane influences at all loads and speeds. Cold starting is related directly to cetane and, the higher the cetane, the better the starting properties.

In the Fairbanks, Morse diesel engine sufficient data were not available to interpret the influence of cetane on performance.

In the Caterpillar diesel no real influence of cetane on performance characteristics in the range of fuels used could be described. Present cold starting tests indicate the possible influence of cetane on low-temperature starting. A tendency is apparent to shorten ignition lag when using higher cetane fuels.

■ General Conclusions on Cetane

When low-temperature combustion-chamber envelopes are approached in engine operating conditions, cetane is a helpful factor. Higher cetane fuels improve idling, light-load operation, and starting. Higher cetane fuels are also of influence in promoting smoothness of combustion.

■ Influence of Volatility

In the General Motors diesel the volatility factor shows no appreciable differences on full-load or part-load operation. There is a marked improvement in smoke at idling, as volatility is increased. On the high-volatility fuel, fuel consumption is about 4 to 6% higher. Volatile fuels show a decrease of 8 to 10% in horsepower output, but viscosity plays a part in this influence. The higher-volatility fuels yield the lowest smoke. At the lower speeds cetane is of influence on smoke but, the higher the speed, the greater the influence of volatility. Some stray results indicate that volatility influences fuel consumption in direct ratio as well as power output.

In the Dodge diesel, volatility exhibited no appreciable influence on full-load or part-load performances. Volatility presented a sensitive factor as regards smoking. Low volatility is conducive to high smoking. At low speeds smoke is influenced by cetane in inverse ratio but, as speeds increase, the volatility factor plays an important role in smoke control. Under accelerating conditions high-volatility fuel is desirable as a means of smoke control.

In the Hercules diesel the power output drops 50% at high speeds on the low-viscosity high-volatility fuel and the engine smoke is heavy. High volatility is a factor in smoke control.

In the Mack Lanova diesel smoke control is affected by volatility, high volatility providing no smoke operation.

In the Fairbanks, Morse diesel at 80% rated load, smoke is inversely related to volatility. High-volatility fuels yield low smoke operation. The total range of smoke readings were within narrow limits.

In the Caterpillar diesel the volatility factor appears of no influence in performance characteristics and no appreciable differences were exhibited in smoke tendencies on any of the fuels.

■ General Conclusions on Volatility

Smoke is the most pertinent characteristic influenced by volatility. High volatility tends to lessen smoke, and high volatility with high cetane eliminates stink.

■ Influence of Viscosity

No attempt was made to separate the viscosity factor. Future tests with various blends undoubtedly will be helpful in interpreting the data by further separation of the viscosity factor from the volatility factor.

■ Influence of Gravity

Generally speaking in all tests (without further isolation of properties) the low-gravity fuels produced the highest horsepower output. The low-gravity fuels developed the

best fuel consumption on a volume basis. In the Caterpillar diesel the low-gravity fuels assisted starting.

■ Influence on Engine Deposits

The test procedure for determining the deposit-forming tendencies of the various reference fuels required that the engine be run at idle and at a simulated altitude of 5000 ft.

	General Motors Diesel	Caterpillar Diesel	
	1-cyl,	4-cyl,	1-cyl,
Fuel	4¼ x 5 in.	4¼ x 5½ in.	4¼ x 5½ in.
A	11.39 gm	9.06 gm	2.26 gm
D	12.56 gm	9.30 gm	2.32 gm
C	24.07 gm	10.94 gm	2.73 gm
B	25.70 gm	15.49 gm	3.84 gm

The deposit tests which were run by the Pure Oil Co. under accelerated conditions were not listed as comparable with the foregoing tests inasmuch as the jacket temperature was maintained at 348 F. It will be noted that the General Motors diesel and the Caterpillar diesel list the four reference fuels as to their deposit-forming tendencies in the same order of rank. No definite correlation to any one factor is apparent, but the deposits appear to be affected by a combination of cetane and volatility influence. High cetane and high volatility gave low deposits in each case. Low gravity and low volatility increased the deposits.

■ Additional Tests

The Standard Oil Co. (Ind.) has attempted to evaluate the four reference fuels in a CFR standard test engine. At a setting of 50% smoke density the amount of fuel injected to hold this smoke density revealed the following order of consumption in unit time:

Fuel A – 28 cc
Fuel B – 25 cc
Fuel C – 26 cc
Fuel D – 26 cc

These tests were set up with the thought in mind of determining the smoke density of the four reference fuels. No correlation appears to exist as related to volatility or cetane. The nearest correlation approaches gravity.

The Pure Oil Co. ran accelerated tests on a Fairbanks, Morse engine using a jacket temperature of 348 F, and determined by thermocouples in liner and piston the "cut-off" point, or place where pronounced rise of the temperature from the normal range is interpreted as the time of removal from operation of a ring due to sticking. A standard lubricant was used throughout the test to limit this variable. These tests must be considered as a high-temperature accelerated run; however, the tests show that the reference fuels were rated in ring-sticking hours in direct relation to cetane. The high-cetane fuel gave longest hours before ring sticking, and the lowest cetane fuel, the shortest hours. The curves indicate definite and reproducible characteristics of the "cut-off" point.

Prof. Wilson of the University of Wisconsin reported a series of tests showing ignition delay, combustion duration, and rate of pressure rise on the four reference fuels at 100% and 75% load factor on a Fairbanks, Morse single-cylinder diesel. The ignition delay in milli-seconds was substantially in the general order of inverse ratio to cetane number, the low cetane number giving the longest ignition delay. Combustion duration did not follow a definite order, but the low-cetane fuel had the shortest duration in milli-seconds, and the highest cetane fuel, long duration. Rate of pressure rise at ¾-load showed highest rates for the low-cetane fuel and the lowest rate for the high-cetane fuel.

In the foregoing paragraphs the various contributors to the CFR Full-Scale Engine Test Group have voiced their thoughts on the initial test runs which have been made. Further testing is expected to verify or disprove the statements made. Suffice it to say that there appears to be a somewhat general agreement on certain tendencies. It is hoped that the final summary of results on this initial series of tests will permit the Committee to project a more extended and definite program of future cooperative activity.

Getting the Most out of Modern Fuels

THE question is often asked: "What about the increases in antiknock value of present-day gasoline? What can we do to take advantage of them?" Let us touch briefly on three points.

First, old equipment. There is no doubt that much of your older equipment has compression ratios not sufficiently high to take full advantage of present-day fuels. However, many of these engines might respond very nicely to increased compression ratios. It is not advisable to undertake any change-over campaign since this involves a large immediate outlay of money. I think it better practice to incorporate a change of compression ratio as engines are overhauled. Consult with your engine supplier. Usually they have optional pistons or cylinder heads which can be used to raise the compression ratio of this old equipment. Sometimes they may tell you that it is not advisable to raise the ratio of some certain piece of equipment because of other factors in the design of this engine.

Second, much of your present new equipment is probably capable of taking advantage of today's fuels by proper spark timing. I would suggest that work be done to determine whether the spark timing you are using is correct. In this part of the country with your many hilly roads, it should not be difficult to determine maximum-power spark setting. Once this is established, see that each flywheel is plainly marked, and provide your maintenance crew with a satisfactory timing light. Remember this one thing – the easier you make it for your maintenance crew to do its job, the more likely it is that the job will be done well.

Third, in future equipment you should think ahead. Most of the experts say that the octane number of gasoline will not stop at its present level which, incidentally, is approximately 74 octane ASTM. Some prophesy that by 1945 we will have 80-octane regular-grade fuel. Therefore, for those of you who buy equipment that is to be operated over a period of five to seven years before being entirely depreciated, there is need to give thought to possible changes in fuel octane number and it might be well to ask your manufacturer whether his engine can have the compression ratio changed and, if so, whether the structural design of the engine is such that it will safely permit increased ratios.

Excerpt from the paper: "A Gasoline Man Looks at Fleet Maintenance," by Errol J. Gay, Ethyl Gasoline Corp., presented at a Baltimore Section Meeting of the Society, Baltimore, Md., Sept. 17, 1940.

MAGNESIUM ALLOYS in t

THE use of magnesium alloys in airplanes and engines has increased to the extent that they have become a significant factor in the rapidly expanding aircraft industry. In view of this condition, a general review of their development, a summary of their present applications, and an indication of how they may be utilized even more widely in the future offers a timely subject. In discussing it, the author has omitted tables giving physical and chemical properties of the various alloys as these are covered adequately by the Army, Navy, or SAE Specifications and because the paper presented at the 1939 SAE National Aircraft Production Meeting by L. B. Grant[1] included tables of such properties as well as information on current shop and fabricating practices in the aircraft industry.

▪ Industry Growing Rapidly

The growth of the magnesium industry in the United States has been rather uniform and steady until the recent aircraft program which has greatly stimulated it. In spite of some recent publicity to the contrary, it should be emphasized that the domestic production of this metal is controlled entirely by American capital and American patents and that the only factors restricting an unlimited supply are the physical requirements of men, equipment, buildings, and power which take time to accumulate and build. Within the past year, we have seen the domestic production doubled and early next year, it will be doubled again so that the supply will have been increased four-fold within slightly over a year. The aircraft industry has been a major factor in stimulating this demand, not only by its greatly increased production, but also because the quantity of magnesium alloy per plane has been increased greatly. In addition to its use in magnesium alloys, increasing quantities of magnesium metal have gone into the production of aluminum alloys as an alloying constituent. This latter requirement has also contributed to the increased demand for base magnesium metal.

▪ Magnesium-Alloy Castings

From a very limited beginning in 1924, when magnesium-alloy castings were first used on American engines and accessories, these alloys have been utilized for increasingly important parts until the modern aircraft engine employs them very widely and for all parts where castings can be used and where they are not subjected to excessive heat. In fact, the very low weight per horsepower which we now have would otherwise be impossible. It is probable that the first magnesium aircraft-engine castings were limited to miscellaneous housings and covers. Satisfactory performance and wider experience, coupled with improved and stronger alloys, have since led to their use for such important parts as crankcases on the smaller and medium-powered engines and as supercharger and accessory housings on even the most powerful engines. In this development it is significant that practically all of the earlier applications have not only proved their value but have resulted in the wider and more important uses which followed. The table below lists some of the typical large magnesium-alloy castings which are used on various types of engines.

The tabulation following is, of course, very brief and does not include the many other major engine parts which are made of magnesium nor the numerous smaller hous-

Typical Magnesium-Alloy Engine Castings

1. **Large Radial Engines**
   - **Front Supercharger**
   - **Front Crankcase Section**
   - **Rear Supercharger Housing**
   - **Rear Supercharger Cover**
   - **Rear Section, or Accessory Housing**
2. **Medium Radial Engines**
   - **Thrust Bearing Housing**
   - **Accessory Housing**
   - **Starter Drive Housing**
   - **Oil Sump**
   - **Carburetor Air Intake**
   - **Rear Section of Split Crankcase**
3. **In-Line Engines**
   - **Supercharger Drive Housing**
   - **Accessory Housing**
   - **Intake Manifolds**
   - **Oil Pump Body**
   - **Camshaft Housing**
   - **Camshaft Cover**
   - **Rear Crankcase**

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 31, 1940.]

[1] See SAE Transactions, August, 1940, pp. 325-331: "Production of Magnesium-Alloy Aircraft Parts," by L. B. Grant.

the Aircraft Industry

by JOHN C. MATHES
The Dow Chemical Co.

AFTER reviewing the growth of the magnesium industry in this country and summarizing present aircraft applications of magnesium alloys, Mr. Mathes indicates how they may be utilized even more widely in the future. He shows that the new magnesium alloys compare favorably in strength with aluminum, making it possible to overcome their lower modulus of elasticity as it affects compression members and still show a considerable saving in weight.

Detailed applications of magnesium alloys discussed in this paper include those in the form of castings, forgings, extrusions, and sheets with respect to both aircraft and aircraft engines; favorable results are shown when compared with similar parts of aluminum even when applied to aircraft structures.

The author suggests that substantial savings may be effected in cost and weight, with some gain in aerodynamic efficiency, through the use of monocoque magnesium sheet structures, such as for fins and stabilizers.

ings, covers, brackets, and so on, all of which contribute to attaining the lightest possible weight. When we consider that some of these castings weigh as much as 60 lb and that most of the ones listed for the larger engines weigh over 20 lb, it is apparent that the weight per horsepower would be substantially increased if aluminum, weighing one and one-half times as much, were used instead.

Somewhat less extensive, to date, has been the use of magnesium-alloy castings in the airplane itself, with the exception of the wheel assemblies where magnesium castings have been standard equipment for several years. The service records of magnesium-alloy wheels have been so satisfactory, and their use so widened, that over 75% of the commercial and military planes now being built are so equipped. Wheel castings as heavy as 160 lb have been made, and wheel castings being produced in large quantities average nearly 55 or 60 lb. The use of magnesium, therefore, represents a weight saving of from 50 to 150 lb per plane for a medium or heavy airplane. Except for engine parts, wheel castings still represent the largest tonnage of cast magnesium for aircraft requirements although other applications are increasing rapidly, both in volume and in importance. In 1938, for instance, we note that the majority of airplane castings were for such secondary items as bell cranks, window frames, camera and equipment mounts, mechanism housings, miscellaneous levers and supports, and air intake scoops, although even then such structural parts as tail wheel forks, control wheels, aileron hinge brackets, or brake pedals, were being used. Today, in addition to an even greater number of miscellaneous parts, we find magnesium castings being used for structural parts of the landing gear and control systems. This development has followed a satisfactory service record for the lesser important parts as well as successful experience with more important parts on experimental airplanes. Together with the recent revision in the "Design Handbook of the Air Corps" which permits a more general use of all cast metals, these factors have contributed to a wider use of magnesium alloys for airplane castings.

■ Reasons for Use in Castings

The reason why magnesium alloys have been used so widely is seen readily when we consider that the sections of many castings are often determined by considerations other than strength alone. The inability to pour castings with less than a nominal wall thickness, the necessity for heavy boss sections, the need for rigidity, and so on, all contribute to the conclusion that, in many cases, castings of any metal would be considerably over strength. If it were not for the fact that such parts can be made of magnesium, the weight penalty of casting them in any other metal might be so prohibitive as to require the much greater cost of fabricating them from wrought material of thinner section. In many cases, the added cost would be appreciable, for only in castings can lever arms, housings, bosses, or hollow sections be combined into one unit so efficiently. Even in highly stressed structural castings, the properties of magnesium are such that they can be utilized to great advantage and a significant weight saving accomplished with no sacrifice in strength. This statement is particularly true of parts where rigidity in bending is required because magnesium sections can be made from two to three times as stiff as aluminum sections of equal weight by virtue of the fact that stiffness increases as the cube of the thickness. Thus, with sections of constant width, this factor will compensate for the differences in modulus of elasticity and provide magnesium-alloy sections 2.55 times as stiff as aluminum sections of equal weight. The following table, giving specified minimum values, indicates the comparison between magnesium-alloy castings and the commonly used aluminum alloy. It shows the close relationship between the strengths of the two mate-

rials and why the magnesium-alloy castings, therefore, offer opportunity for weight economy:

Comparison of Properties, Aluminum and Magnesium Casting Alloys

Alloy	Elongation, %	Ultimate Tensile Strength, lb per sq in.	Tensile Yield Strength, lb per sq in.
Dowmetal HHT	7	32,000*	10,000*
Al. 195T4	6	29,000†	13,000†
Dowmetal HHTA	3	34,000*	16,000*
Al. 195T6	3	32,000†	18,000†

* Specified minimum values, U. S. Army Spec. – No. 57-74-1C.
† Specified minimum values, ANC-5.

Sheets and Extrusions

At the present time, the majority of the magnesium-alloy sheet and extrusions in aircraft are for secondary or comparatively non-stressed applications. Such parts include extruded sections for seats, panel stiffeners, fillers, turret and canopy framing and various cabin parts and fittings. Similar sheet applications include oil tanks, cabin linings, wheel fairings, dust covers and hub caps, partitions, and pressed metal seat parts. In addition, sheet is being used for air ducts and other appurtenances of the cooling system. All of these sheet-metal parts are made from the magnesium-1.5% manganese alloy designated by Army Specification 11317 or Navy Specification M-111d, Alloy No. 11 as it is considered to be the most readily fabricated alloy. A similar alloy is widely used in Germany and England for cowlings, fairings, and fuselage coverings although similar uses have not been adopted to any extent in this country. The reason for this may be that their practice is to form these parts by hand so that the necessity of heating the sheet for forming is not as serious a complication as it would be in our plants where similar parts usually are formed in presses and would require a special set-up in order to supply the necessary heat. However, at least one of the major American companies has equipped a press for hot forming and has found that the process is entirely satisfactory and that the warmed sheet, being dead soft, also has eliminated the spring-back problem. In most cases, magnesium is specified for these parts to save weight as, with strength a secondary consideration, a maximum weight saving is obtained. In some applications, however, the ability to secure additional stiffness at no penalty in weight is utilized to advantage. Typical is the use of magnesium-alloy plate for instrument panels where adequate rigidity is obtained with a single flat sheet of fairly thick gage and where the use of a heavier metal would require the fabrication and attachment of numerous stiffeners if thin enough sheet were used to keep the weight comparable.

Structural Applications

It is comparatively recently that magnesium alloys have been considered actively as a material of construction for the entire airplane structure. This possibility is, in fact, so promising of results that test wing panels are now being built for the military services. As these will be the initial attempts with a new material and with new details of construction, some difficulties undoubtedly will show up as construction progresses and the tests are completed, but there are definite advantages in the utilization of such wings and it is believed that the test results will stimulate a much wider interest. If so, the consideration of this possibility by other engineers with their varying experience and background will be very beneficial.

Such work as has been done to date indicates that magnesium alloys, used structurally, may be utilized to: 1. Save weight; 2. Save cost; and 3. Increase aerodynamic efficiency, particularly in high-speed airplanes.

In the following discussion, a brief summary of some of the experimental and theoretical work leading to these conclusions will be presented. It is, of course, impossible to separate the subjects entirely as some features contributing to one of the foregoing objectives also will figure largely in another.

Weight-Saving Possibilities

The extreme light weight of magnesium alloys immediately suggests weight saving as the primary objective in using them structurally. Early tests, however, showed that a comparatively low compressive yield strength more than overcame their lower specific gravity in semi-monocoque, or stressed-skin, construction and it was not until the development of stronger alloys within the past year that this possibility could be attacked effectively. These new alloys compare favorably in strength with aluminum alloys and make it possible to overcome the lower modulus of elasticity, as it affects compression members, and still show a considerable saving in weight. This comparison in strength is shown by the following table:

Comparative Properties of Wrought Magnesium and Aluminum Alloys

		Typical Properties		
Alloy	Form	Ultimate Strength, lb per sq in.	Tensile Yield Strength, lb per sq in.	Elongation, %
Dowmetal ZHTA	Extruded	55,000*	40,000*	7
24ST	Extruded	57,000†	42,000†	12
Dowmetal J-1h	Sheet	45,000*	35,000*	8
24ST	Sheet	62,000†	40,000†	16

* Tentative specifications, The Dow Chemical Co.
† Specified values, ANC-5.

As this table shows, the 24ST sheet is considerably stronger than the magnesium-alloy sheet but, as in a sheet-stringer combination, the strength of the extrusion is the controlling factor; this is not particularly detrimental.

Perhaps the best method of demonstrating the possibilities of magnesium for semi-monocoque construction is to analyze a typical sheet-stringer combination theoretically and then to show how tests check the results so obtained. In this analysis, it should be kept in mind that there has not been sufficient time since the development of these alloys to work out the many empirical formulas which are in common use for aluminum construction and that it is, therefore, necessary to assume that many of the constants

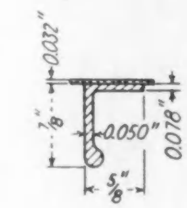
which have been determined for aluminum will apply equally well to magnesium. This is not necessarily the case but, if the conclusions so obtained indicate potential advantage, the desirability of further investigation is apparent. The formulas applying to the immediate problem are the column formula:

$$\frac{P}{A} = f_{yp} - \frac{f^2_{yp}\left(\frac{L}{r}\right)^2}{4c\pi^2E} \qquad (1)$$

and the effective width formula

$$b_e = 1.7t\sqrt{\frac{E}{f_c}} \qquad (2)$$

With the foregoing formulas, it is interesting to obtain the comparative theoretical strengths of equal-weight aluminum and magnesium-alloy sheet stringer panels in flat-end compression tests. Consider, for example, the following combination in aluminum:



Angle = $\frac{5}{8} \times 0.078 - \frac{7}{8} \times 0.050$ in.
Angle Area = 0.111 sq in. (actual by weight)
Effective width of sheet =

$$1.7 \times 0.032\sqrt{10 \times 10^6/36{,}000}$$
$$1.7 \times 0.032\sqrt{10^7/36{,}000} = 0.91 \text{ in.}$$

The section properties of the angle and sheet working together are then:

$A = 0.140$ $\qquad r = 0.36$
$I = 0.0143$ $\qquad l/r = 41.6$ (for 15-in. column length)

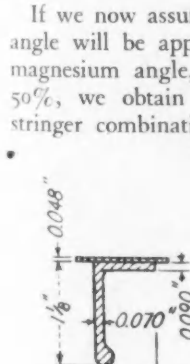
Substituting in Formula (1):

$$P/A = 42{,}000 - \frac{42{,}000^2 \times 41.6^2}{4 \times 2 \times \pi^2 \times E} =$$
$$42{,}000 - 3900 = 38{,}100 \text{ lb per sq in.}$$

The total load carried by each stringer combination is therefore:

$$P = 0.140 \times 38{,}100 = 5340 \text{ lb}$$

If we now assume that the b/t ratio for the aluminum angle will be approximately correct for an equal-weight magnesium angle, and if we increase the sheet gage by 50%, we obtain the following magnesium-alloy sheet-stringer combination of approximately the same weight:



Angle = $0.6 \times 0.090 - 1.125 \times 0.070$ in.
Angle Area = 0.161 sq in.
Effective width of sheet =

$$1.7 \times 0.048\sqrt{6.5 \times 10^6/30{,}000} = 1.2 \text{ in.}$$

The section properties of the angle and sheet working together are then:

$A = 0.224$ $\qquad r = 0.425$
$I = 0.0404$ $\qquad l/r = 35$ (15-in. column length)

Substituting in Formula (1) and using a compressive yield strength of 34,000 lb per sq in., which seems to fit the experimental results fairly well we obtain:

$$P/A = 34{,}000 - \frac{34{,}000^2 \times 35^2}{4 \times 2 \times \pi^2 \times E} =$$
$$34{,}000 - 2750 = 31{,}250 \text{ lb per sq in.}$$

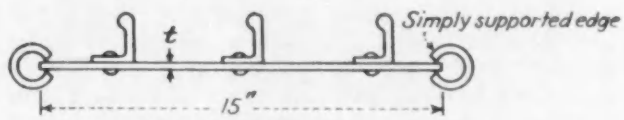
The total load carried by the stringer combinaton is then:

$$P = 0.224 \times 31{,}250 = 7000 \text{ lb}$$

This load represents an increase over the equal-weight aluminum section of

$$\frac{7000-5340}{5340} = 31\%$$

This theoretical conclusion is confirmed by the following actual test results:



TEST SECTION

Panel No.	Alloy	Sheet t, in.	Angle Section	Load at Failure, lb	Panel Wt/Ft, lb	Load Carried per Unit Weight, lb
1	Mg	0.051	(9/16x0.050 – 7/8x0.050)	12,385	0.78	15,800
2	Al	0.032	(13/16x0.040 – 17/32x0.050)	9,380	0.825	11,350
3	Mg	0.064	(7/8x0.090 – 1 1/8x0.080)	24,300	1.15	21,100
4	Al	0.032	(7/8x0.050 – 9/16x0.078)	15,420	0.99	15,500

The last column expresses the load carried in relation to the weight of the section and is used in order that we may make the following direct comparison:

$$\text{Panel 1 vs. Panel 2 – Increase} = \frac{15{,}800\text{-}11{,}350}{11{,}350} = 39\%$$

$$\text{Panel 3 vs. Panel 4 – Increase} = \frac{21{,}100\text{-}15{,}500}{15{,}500} = 36\%$$

These experimental results check the theoretical results previously derived within reasonable limits. Although it would be unwise to generalize too widely from these meager data, it does appear that magnesium used in semi-monocoque construction will effect a substantial saving in weight.

Utilizing Magnesium to Save Cost

In the interest of saving cost, as well as of speeding up production, consideration has been given to effective means for accomplishing this purpose. One very likely method is the development of pure monocoque construction so that the vast number of stringers and stiffeners can be eliminated, or at least reduced appreciably. As it is necessary that this be accomplished without an excessive increase in structural weight, it appears that magnesium alloys warrant serious consideration in the further development of this type of structure. Prof. E. W. Conlon of the University of Michigan, working in conjunction with Wright Field, has made a start along this line by constructing and testing a magnesium tail fin of pure monocoque construction. The results of this test demonstrated the practicability of such construction and are described in his article in the April, 1940, issue of the *Journal of the Aeronautical*

Sciences. A more complete description of the test is given in Air Corps Technical Report No. 4530, entitled "Magnesium Alloy Research" and dated May 16, 1940. It is intended to follow up this work by the design, construction, and test of a complete tail assembly but, as yet, this has not been done. It is anticipated, however, that it will be accomplished at an early date and that the validity of the assumption that magnesium-alloy construction of this type is both practical and desirable will be demonstrated.

For the present, and in lieu of more complete test data, it is necessary to approach the problem from a purely theoretical standpoint. In Prof. Conlon's paper previously referred to, he has demonstrated that, for pure monocoque construction, the use of magnesium-alloy sheet should accomplish a weight saving of 26% for flat sheets and of 20% for curved sheets. This is done, with flat sheets, by assuming equal loads and by equating the formula $P = K_e E\,(t/b)^2\,bt$ for both aluminum and magnesium. Substituting the proper values for E, he derives the theoretical conclusion that magnesium sheet 1.166 times as thick as the aluminum sheet will carry an equal load. This added thickness compensates for the lower modulus of elasticity and, by virtue of the difference in specific gravity, results in a weight saving of 26%. Similarly for curved sheet, he has used the formula $P = \dfrac{KE}{R/T}\,bt$ to determine a theoretical weight saving of 20%.

To illustrate this more specifically, we have made an analysis of a small monocoque section, not only to check the difference in weight between the aluminum and magnesium, but, also to get a rough check on the comparison with semi-monocoque construction. For this purpose, we have arbitrarily taken the cylindrical section shown on page 172 of "Airplane Structure" by Niles and Newell. This cylindrical section has a 20-in. radius, a 0.020-in. thick outer skin and is reinforced by sixteen channel stiffeners formed from 0.032-in. material. The bending moment is given as 330,000 in.-lb and, if we assume that the section is properly proportioned, the semi-monocoque section to resist this moment weighs 4.05 lb per ft of length. The weight of both an aluminum and a magnesium pure monocoque cylinder to resist the same moment may be determined by assuming that the formula:

$$f_{cr} = \frac{0.25\,Et}{R} \qquad (3)$$

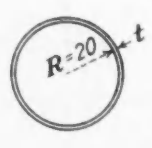
will apply equally well to both materials and by further assuming that the formula

$$f = \frac{Mc}{I} \qquad (4)$$

will hold as long as the stresses do not exceed the buckling stresses as determined by Formula (3). The analysis follows:

(Note: Weight of semi-monocoque section = 4.05 lb per ft)

For monocoque sections:



$$f_{cr} = \frac{0.25 \times 10^7 t}{20} \text{ for Al}$$

$$f_{cr} = \frac{0.25 \times 6.5 \times 10^6 t}{20} \text{ for Mg}$$

M = 330,000 in.-lb

$$S = \frac{\pi}{32(40+2t)}\left[(40+2t)^4 - 40^4\right]$$

Tabulation of Results for Various Thicknesses

Thickness, in.	$S = \pi t R^2$	$f = M/S$	Al $f_{cr} = \dfrac{0.25\times 10^7 \times t}{R}$	Mg $f_{cr} = \dfrac{0.25\times 6.5\times 10^6 \times t}{R}$
0.040	50	6600	5000	3250
0.045	56.6	5840	5625	3660
0.046	58	5700	5750	3740
0.050	65	5070		4060
0.057	71.5	4620		4640

From the foregoing it appears that the monocoque aluminum shell would be 0.046 in. thick and the magnesium 0.057 in. thick. The aluminum shell, 40 in. in diameter, would weigh 7.0 lb per ft, or 73% more than the semi-monocoque construction, while the magnesium shell would weigh 5.6 lb per ft, or 37% more than the semi-monocoque. Comparing the monocoque shells themselves, the magnesium is $\dfrac{7.0\text{-}5.6}{7.0}$ = 20% lighter, as would be expected from the relationship previously shown.

To Increase Aerodynamic Efficiency

There are two major factors which indicate the desirability of heavy skin covering for extremely high-speed airplanes. First, the need of smooth surfaces to increase the aerodynamic efficiency and, second, the need of greater strength to resist skin failure resulting from the high intensity of local pressures which may develop at diving speeds. The use of flush and polished surfaces, extremely smooth, contributes greatly to reducing drag at high speeds because even slight surface irregularities in the forward portion of the wing cause a transition in the boundary layer from the laminar to the turbulent state which increases the drag coefficient. Initial efforts to attain this condition have led to flush riveting, spotwelding, and such means, but the practical difficulties of building an absolutely smooth wing surface with extremely thin gages, and the difficulty of keeping it smooth under load, lead to the consideration of using thicker coverings to obtain the maximum results. Particularly with the high wing loadings that planes will have, the thicker skin also may be required to sustain the excessive normal loads which may be developed over local areas. It has been shown that, even at velocities below the speed of sound, local areas over the wing may develop these velocities and a compressibility shock or burble result. The same irregularities which may contribute to increased drag will also contribute to the development of such shocks and, as noted in the Air Corps Report on this subject, increased skin thickness will be necessary to resist them. This thicker skin will then not only act to reduce the formation of compressibility shocks but also will give a covering better able to resist the excessive local loads imposed. We can only conjecture at the present time, but it seems certain that magnesium, with its extremely low specific gravity, will prove to be a useful tool in overcoming these problems.

CAR CONTROL FACTORS and their MEASUREMENT

by KENNETH A. STONEX
Assistant Mechanical Engineer, General Motors Proving Ground

A DESCRIPTION of two tests for car control, with results showing the effect of such variables as roll rate, load distribution, and tire pressure, is presented in this paper. Car control is defined by the author as the security with which the driver can maneuver his car over the highway, where he wants to and as he wants to, with precision and care. The two tests discussed were designed to find out what actually happens when a car goes around a curve, what factors are variables, and to set up a test procedure whereby these variables can be measured objectively, their relative importance determined, and the laws of their variation found.

In the first of these tests, the skid-pad roadability test, the specific problem is to measure the aspect of the car in relation to the pavement plane and the relative motion of various parts of the car under conditions of constant speed and travel in a circular path.

To study control problems that come during states of transition, when the car is entering or leaving a curve or when it passes from one curve into a reverse curve, or encounters cross winds, varying pavement slopes, or rough road surface, the second test described, a dynamic transition test called the "checkerboard" test, was devised.

BY car control we mean primarily the security with which the driver can maneuver his car over the highway, where he wants to and as he wants to, with precision and ease. In this paper we are concerned with the problem as it pertains to steering and handling, and not to braking. With the advent during the last 10 or 20 years of large traffic volumes, fast roads, and high speeds built into even the low-priced cars, good control has become the most important mechanical element in safety.

Development of the qualities which determine control has been along empirical lines in the main; and, to the best of the author's knowledge, there is no purely objective test of handling. The final check on any design is how it feels, and the immediate problem is solved satisfactorily if the new design feels as much better than the current one as the designer intended. As a part of this empirical system, the objectives of good control are set and the final product checked against them by a small group of experts in each manufacturing organization.

The objection to this system is that control or handling or steering are highly subjective phenomena. No car feels the same to any two people and neither can communicate his feeling intelligently to the other. And further, no two people want quite the same thing in an automobile.

Steady Improvement Obtained

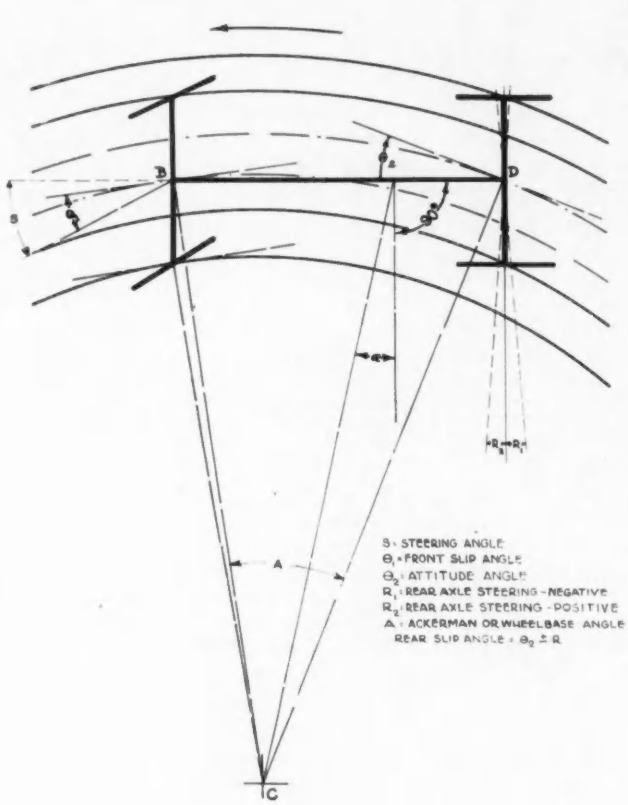
The empirical system has been a practical success, of course, since in general there has been consistent improvement from year to year, 1941 cars being better than 1940, far superior to 1931 models and incomparably better than 1921 cars. The method is slow and laborious, however, and involves a tremendous amount of experiment, so that improvement comes in relatively small steps. What we all desire is some way to lengthen these steps, a method whereby we can build 1950 control, 1950 safety into 1942 cars.

The general problem in car motion is to consider the space relation of the car and the pavement surface. In power and braking problems we are concerned almost exclusively with linear motion on the road; in simple steering most of the problems are two-dimensional in the pavement plane; and in control and handling, the problems are three-dimensional to the extent of the range of spring action. The sprung mass of the car undergoes restrained motion in three dimensions, and the unsprung parts undergo restrained motion, primarily in two dimensions if we assume a smooth surface.

The major part of the control problem is solved when we determine the interaction of these two types of motion and the forces of which they are the resultant, in so far as they affect the driver's ability to put the car where he wants to. Control is a physical problem; handling is the physical problem plus the physiological and psychological effects on the driver.

The first requirement is to find what actually happens when a car goes around a curve, what factors are variables, and then to set up a test procedure whereby we can

[This paper was presented at a meeting of the Detroit Section of the Society, Detroit, Mich., Dec. 2, 1940.]



■ **Fig. 1 – Schematic sketch of car on skid pad**

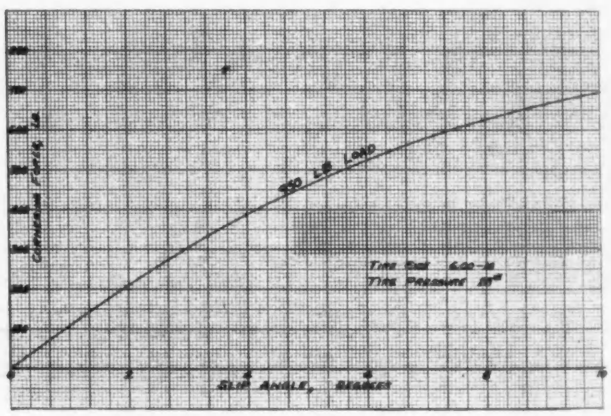
measure these variables objectively, determine their relative importance, and find the laws of their variation. This paper will be limited to the description of two such tests with results showing the effect of a few simple variables such as roll rate, load distribution, and tire pressure.

■ Skid-Pad Roadability Test

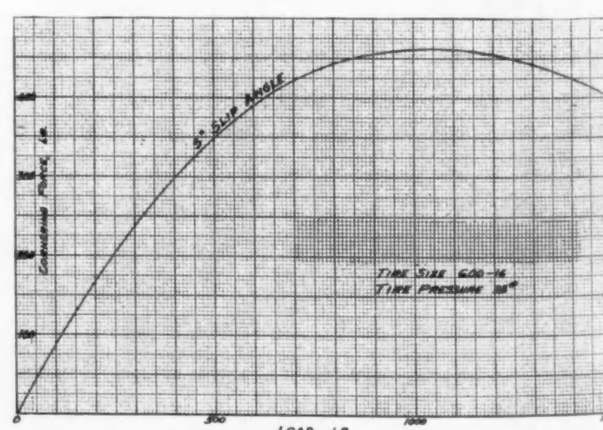
The first test developed is what we call the skid-pad roadability test. The skid pad is a concrete pad, flat and level, except for a slight drainage slope toward the center, with a finely trowelled surface, and large enough so that the car can be driven in a circle of 108-ft radius.

The specific problem on the skid pad is to measure the aspect of the car in relation to the pavement plane and the relative motion of various parts of the car under the conditions of constant speed and travel in a circular path. The aspect of the car relative to the pavement surface is a function of the three angles in space; that is, we may consider a restrained rotation about the three principal axes through some point in the car. For convenience we choose the longitudinal and transverse axes horizontal, and a vertical axis. Rotation about the transverse horizontal axis, that is, fore and aft rotation, or pitch or associated motions, may be assumed to be zero under conditions of constant speed. Since we are running at constant speed and on a circle, rotation about the vertical axis may be measured with reference to the circle. Rotation about the longitudinal axis, or roll, may be measured with respect to the horizontal.

Fig. 1 is a schematic diagram of a car rounding a curve. *B* and *D* are the centers of the front and rear axles, respectively; and the line *BD* we take to be the longitudinal axis of the car. We have found that, as soon as the car moves at an appreciable speed, the rear wheels move out relative to the front. At very low speeds the rear wheels describe a path inside the front and, at high speeds, a path outside the front. This is a rotation about a vertical axis. At very low speeds the axis *BD* coincides with the tangent to the circle, at *D*, which is the direction of travel; and, as the speed is increased, an angle φ_2 is developed between the car axis and the direction of travel. This angle measured with respect to the tangent through the rear axle is called the "Attitude Angle." Since the rear axle is approximately normal to the axis *BD*, the rear-wheel planes are approximately paralleled to *BD*, and the rear wheels, also, operate at some angle to their direction of travel. This angle is called rear slip angle. The importance of slip-angle measurements is pointed out in the papers of Evans, Olley, and Bull.[1, 2, 3]



■ **Fig. 2 – Cornering force-slip angle curve**



■ **Fig. 3 – Cornering force-load curve**

[1] See SAE Transactions, February, 1935, pp. 41-49: "Properties of Tires Affecting Riding, Steering, and Handling," by R. D. Evans.

[2] See Proceedings of the Institution of Automobile Engineers, Vol. XXXII, 1938, pp. 509-572: "National Influences on American Passenger Car Design;" see also SAE Transactions, March, 1934, pp. 73-81: "Independent Wheel Suspension – Its Whys and Wherefores;" both by Maurice Olley.

[3] See SAE Transactions, August, 1939, pp. 344-350: "Tire Behavior in Steering," by A. W. Bull.

Attitude is measured by means of an ordinary surveyor's transit mounted near the car centerline in the rear seat compartment. The effective longitudinal axis is determined by running on the straightaway, and attitude is measured on the circle by sighting continuously on a center pole and reading the average angle between the center pole and the normal to the effective axis. This reading, of course, gives attitude at the transit; to get attitude at the rear axle, the angle between the tangents at the transit and rear axle must be added. This is a function of the distance between the two points and the radius of the circle. Roll angle is also measured with the transit.

On the straightaway, the direction of travel is determined by the position of the rear wheel planes and the rear axle is normal to the direction of travel. If the effective center of vertical motion of the rear axle with spring deflection is above or below the axle center, unequal vertical movement of the two rear wheels, as in roll, will cause slightly unequal horizontal movement and the rear axle will be rotated through some horizontal angle. The amount and direction of this angularity depends upon the position of the effective center. Rear axle angularity is measured by passing a piano wire from an axle extension on one side, around a system of pulleys at the front to a similar extension at the other. Motion of one of these pulleys is transmitted to an indicator in the car by means of Autosyn motors. If the rear axle rotates so that it tends to steer the back end in toward the center, opposing the centrifugal force due to the car's rotation about the circle, we arbitrarily call it a negative steering axle. Similarly, if it rotates in the opposite direction, tending to steer away from the center of the circle and in the same direction as the centrifugal force, we call it a positive steering axle. In Fig. 1, R_2 and R_1 indicate the direction of positive and negative steering, respectively.

Rear-axle steering due to roll angle is proportional to and can be expressed in terms of it; the normal range is from $-1/3$ to $+1/2$ deg, or -4% to $+9\%$ of the roll angle. This rear-axle steering is, of course, the thing which makes the attitude and rear slip angle different; although it is small, measuring it adds to the precision with which rear slip angle can be determined. For a given attitude, the rear slip angle will be greater or smaller as the rear-axle steering is negative or positive. On Fig. 1, the Rear Slip Angle is $\varphi_2 + R_1$, or $\varphi_2 - R_2$ with negative and positive steering, respectively.

There is no direct way of measuring the corresponding front slip angle because the tangent through the front wheels cannot be located in practice. It is computed from the attitude and the steering angle, which is the angle between the front wheel plane and the car axis. The steering angle of each front wheel is measured by means of a universal-joint-hinge system attached to an extension on the spindle and transferring the angular motion to an Autosyn motor clamped on the fender. The individual wheel angles or the average may be used, according to the requirements.

On Fig. 1,

φ_1 = front slip angle (average of 2 wheels)
S = steering angle (average of 2 wheels)
A = angle between tangents = central angle subtended by wheelbase.

This difference between the tangents at B and D must be considered in calculating φ_1.

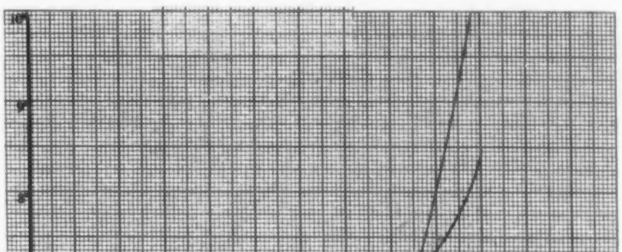
Therefore,

$$\varphi_1 = \varphi_2 + S - A \qquad (1)$$

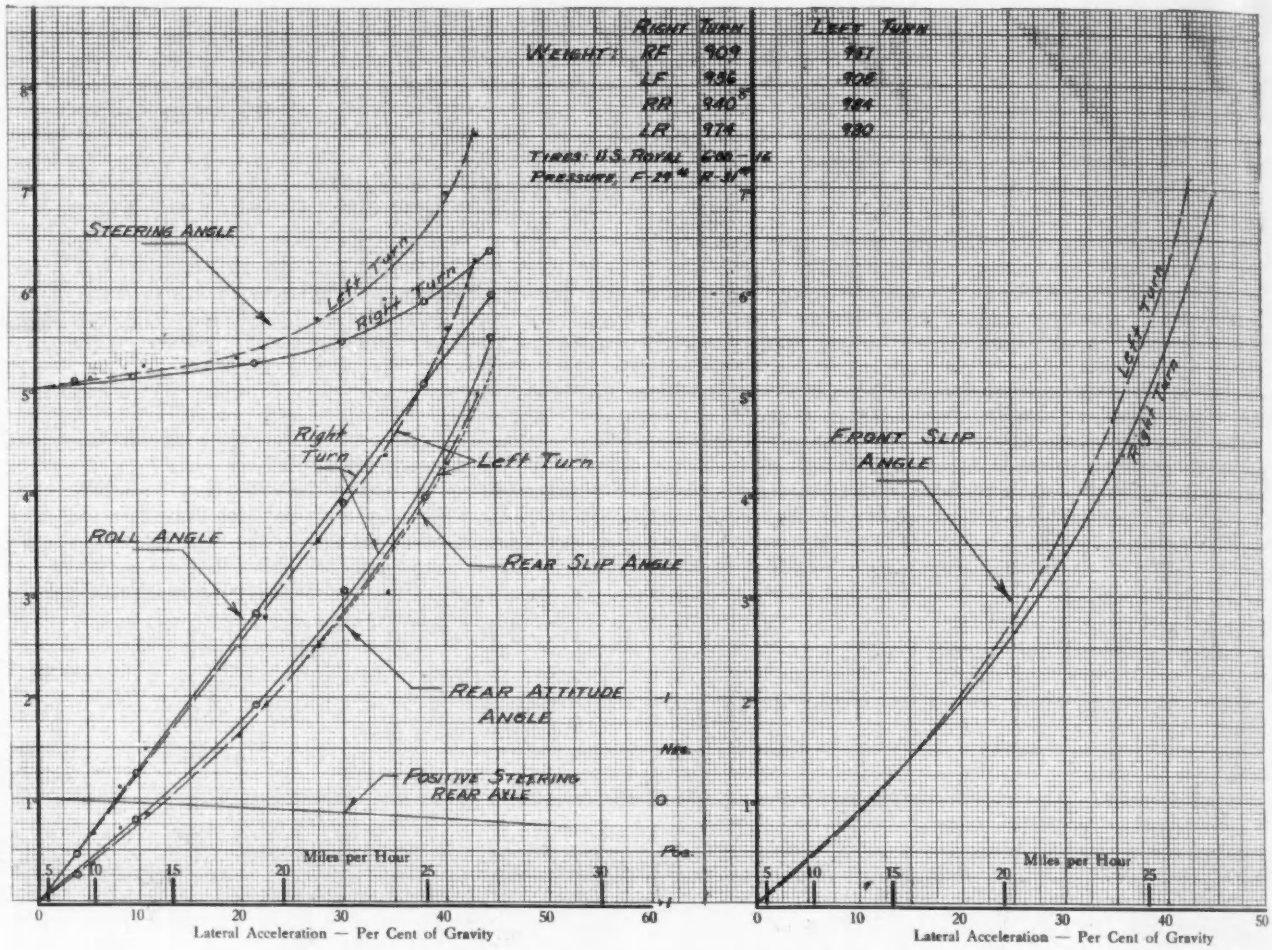
and if $\varphi_1 = \varphi_2$, $S = A$, that is, when the front and rear slip angles are equal, the steering angle equals the wheelbase angle.

Cars fall into two important categories depending upon their steering characteristics. The first type tends to steer toward an external force applied at the center of gravity, and the second type tends to steer away from such a force. Those in the first group tend to head into a curve more and more and have to be steered out of it some, and those in the second group tend to fall away and have to be held into the curve. We call the first group "oversteering" and the second group "understeering." Maurice Olley[2] has shown that oversteering cars are inherently unstable and that understeering cars are inherently stable, even on a straight road for, if an oversteering car comes under the influence of a cross wind or a sloping pavement, it tends to steer away from its course into the exciting force along some arc. The centrifugal force from the curved path adds to the original force, and the car oversteers into an arc of accelerating curvature. The understeering car, on the other hand, tends to steer away from the exciting force so that its centrifugal force opposes and cancels the original force. Understeer is a condition of inherent stability and, therefore, is highly desirable.

The reason for this behavior can be found in the study



■ Fig. 4 – Diagram of steering types



■ Fig. 5 – Skid-pad roadability curves – Car No. 7144 – Standard test, right and left turns

of slip angles. Evans and Bull[1, 3] have shown that a tire operating on a drum at some angle to its direction of travel develops a force having a component normal to the line of travel. They called this cornering force and pointed out that it is a function of slip angle, tire size and pressure, load, and other variables. Figs. 2 and 3 show typical relations between cornering force and slip angle and load. From Fig. 2, it is apparent that, if the cornering requirements are different, or the cornering power, that is, the cornering force per degree is different on the front and rear wheels, equilibrium between cornering force and external force will be reached at different slip-angle values. If the rear slip angle attains a greater size, the rear end will drift farther than the front under the influence of an external force, and the car will head into the force, that is, it will oversteer. If the front slip angle is greater, the front end will drift farther and the car will steer away from the external force, that is, it will understeer. Oversteer is caused by relatively greater rear slip angle and understeer by relatively greater front slip angle.

In making the skid-pad test, the car is run on the 108-ft radius at constant speed in increments of 2½ mph from 5 mph up to about 28 or 30 mph, which is as high as satisfactory readings can be made. Each speed is held for one or more laps of the circle so that good average values can be obtained, and particular care is taken to make readings only when the car is in as stable a condition as possible and when it is actually on the circle. Speed is obtained by measuring the time for a full lap with a stop watch.

$$V(\text{mph}) = \frac{2\pi \times 108}{T(\text{sec})} \times \frac{3600}{5280} = \frac{463}{T(\text{sec})} \tag{2}$$

Data can be plotted against speed, but tests on any other radius are not then directly comparable. It also can be plotted against centrifugal force. Tests on any radius are comparable, but there is no direct comparison between cars of different weight or between the front and rear wheels of the same car. We plot the data against radial or lateral acceleration expressed in percent of gravity; here results on any car, on any radius, and at any speed are directly comparable.

As long as the car follows a level circular path, the cornering force and the centrifugal force are equal for the whole car and at each pair of wheels; the speed, weight, radius of the curve, and the cross gradient of the pavement, Tan φ, if any, completely determine the cornering force.

$$\text{Cornering force} + W \text{ Tan } \phi = \text{centrifugal force} \tag{3}$$

$$\text{Cornering force} = \frac{WV^2}{GR} - W \text{ Tan } \phi$$

$$\frac{\text{Cornering force}}{W} = \frac{V^2}{GR} - \text{Tan } \phi = \frac{\text{Radial acceleration}}{G} \tag{4}$$

The left member of this equation multiplied by 100 is

used by M. L. Fox[4] in his paper before The Highway Research Board describing this test. He calls the quantity "cornering ratio." R. A. Moyer[5] in his Bulletin 120 introduced this quantity into the field of highway engineering under the name "coefficient of friction." This is a misnomer since it is a variable with a certain critical value which, in turn, is a function of the particular automobile as well as of the road surface.

From Fig. 2 it is evident that the slip angle is zero when the cornering force is zero. From Fig. 1 and Equation (1) the value of steering angle at zero side force, or everywhere that $\varphi_1 = \varphi_2$, is equal to the wheelbase angle, or $S = A$. From Equation (1) we get

$$\varphi_1 - \varphi_2 = S - A$$

In terms of oversteer and understeer on the skid-pad charts we have if

$$\varphi_1 > \varphi_2, S > A \text{ – understeer}$$

$$\varphi_1 < \varphi_2, S < A \text{ – oversteer}$$

That is, the car understeers where the steering angle is greater than its value at zero and oversteers where it is smaller than the original value. Most cars are increasingly either understeering or oversteering throughout the range; some are neutral; a few reverse from understeering to oversteering; and, very rarely, there is a reversal from oversteering to understeering. The change in steering angle is a direct measure of the type and amount of the steering characteristic. Fig. 4 is a diagram of steering types.

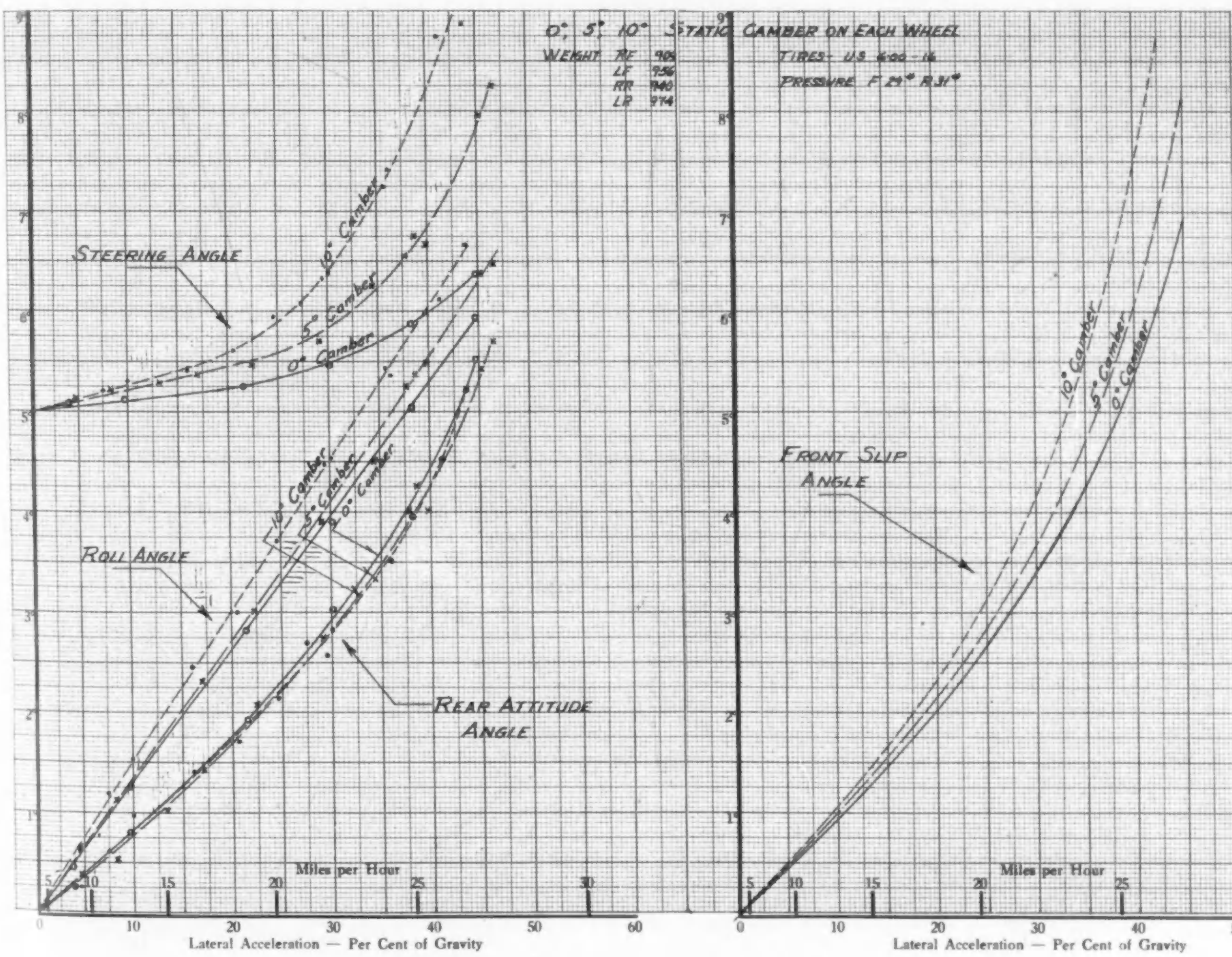
It should be pointed out that the fact that a car oversteers or understeers means that either the rear or the front wheels are doing a relatively poorer job of cornering. For purposes of stability, a certain amount of understeering is desirable, but for efficiency we should have approximately equal slip angles. Relatively high front or rear slip angles are a direct indication of cornering inefficiency, and this inefficiency will show up in terms of high tire wear and high cornering drag. By cornering drag we mean the tangential component of the slip angle thrust, where cornering force is the radial component. With excessive understeer we should expect low front tire mileage and high front wheel drag. Moderate understeer of 2 or 3 deg at $0.4g$ is probably a satisfactory compromise.

Fig. 5 shows the standard skid-pad results on the car used for illustrative purposes in this paper. It has 1 deg understeer at $0.4g$ and conforms to the specifications acceptably.

In the following charts are shown the influences of some simple and convenient variations in suspension characteristics. These variations are used primarily because they were easily done on the particular car, because our experi-

[4] See Proceedings of the 17th Annual Meeting of the Highway Research Board, 1937, p. 202: "Relations between Curvature and Speed," by M. L. Fox.

[5] See Bulletin 120, Iowa Engineering Experiment Station: "Skidding Characteristics of Automobile Tires on Roadway Surfaces and Their Relation to Highway Safety," by R. A. Moyer.



■ Fig. 6 – Skid-pad roadability curves – Car No. 7144 – Camber effect

ence has indicated their importance, and because they correspond to some of the variations reported on tire tests.

On the report of the standard skid-pad test, Fig. 5, the steering angle, roll angle, attitude, rear slip angle and rear-axle steering angle are plotted on one scale and the front slip angle on another for clarity. Results on right and left turns are given. There is characteristically a slight difference in these curves; this difference may be explained by the effect of engine torque reaction and the shift in weight of the transit observer who always rides on the outside. Right turn has been accepted as the standard direction and only right turns were run on the tests following. The average steering angle and front slip angle are plotted because it is simpler and gives a perfectly good indication of the relative balance of cornering capacity. The rear-axle steering is approximately proportional to roll angle, and will not be shown on the succeeding charts.

The curves on Fig. 6 were obtained by running the car with initial static camber settings 0, 5, and 10 deg on each wheel. Assuming that camber change is equal to roll angle, at lateral acceleration of 40% of gravity we have:

Static Camber, deg	Roll Angle, deg	Actual Camber on Outside Wheel, deg	Actual Camber on Inside Wheel, deg	Average Camber, deg +	Total Camber, deg	Change in Average Camber deg	Change in Average Camber %	Change in Roll Angle deg	Change in Roll Angle %
0	5.3	5.3	−5.3	5.3	10.6	...	....	...	0
5	5.5	10.5	−0.5	5.5	11.0	0.2	3.8	0.2	3.8
10	6.1	16.1	3.9	6.1	12.2	0.8	15.1	0.8	15.1

At the same time the front slip angle and steering angle changes are:

Static Camber, deg	Front Slip, deg	Change in Front Slip deg	Change in Front Slip %	Steering Angle deg	Change in Steering Angle deg	Change in Steering Angle %
0	5.35	....	....	6.0	....	....
5	6.10	0.75	14.0	6.90	0.90	15.0
10	7.55	2.20	41.1	8.25	2.25	38.0

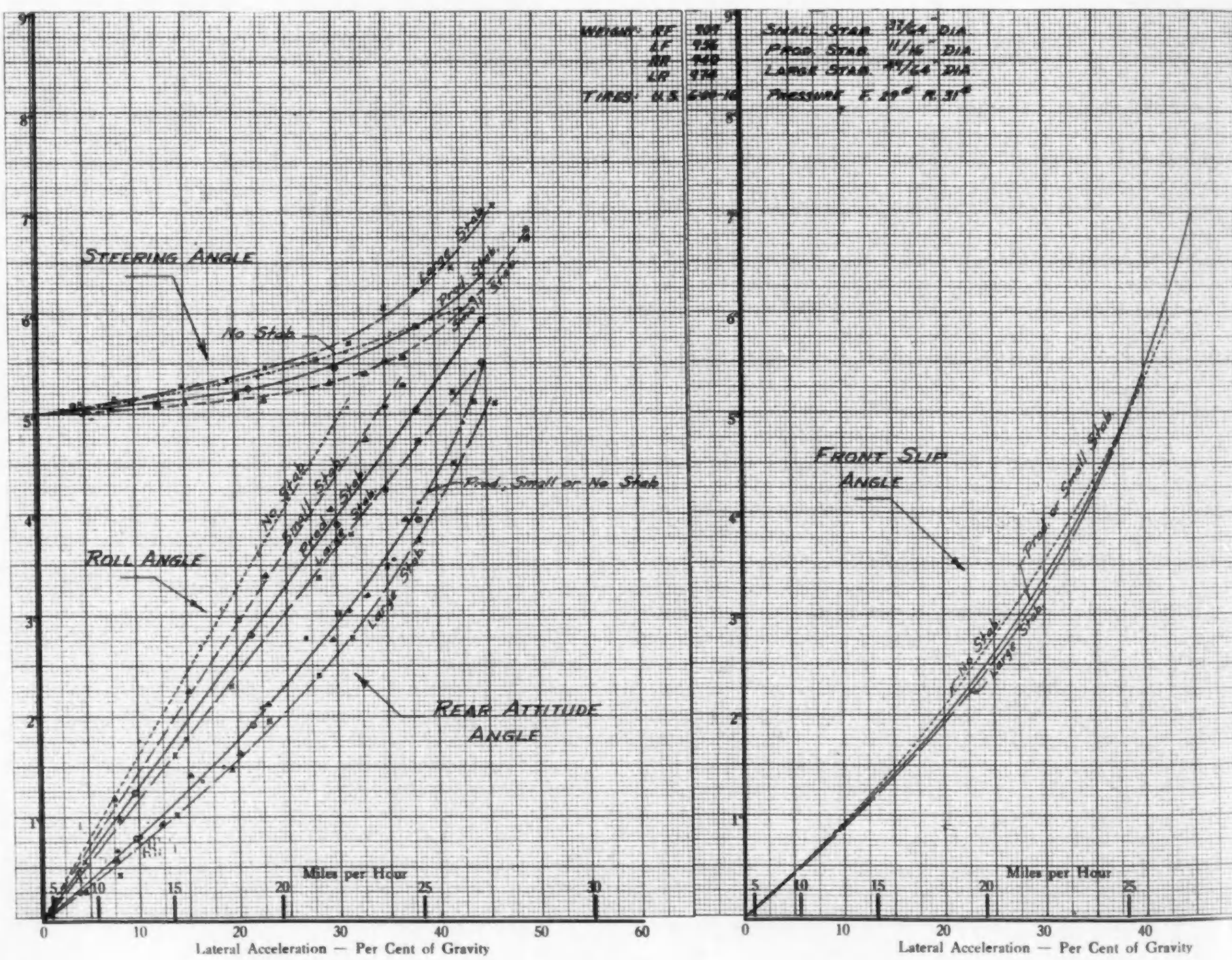
There is no significant change in attitude angle.

These curves indicate that an average camber increase of 15% brings about a slip angle and steering angle increase of approximately 40%.

There are factors involved, such as the increase in roll angle which alters the dynamic load distribution somewhat and the fact that the actual camber was not measured, which suggest that definite conclusions be held in abeyance. The increase in roll angle is probably due to the fact that with camber in these amounts the effective tread is reduced considerably so that the effective roll rate is reduced 3.8 and 15% with 5 and 10-deg Static Camber.

Fig. 7 shows the comparative skid pad curves for several front torsional stabilizer conditions.

Front Stabilizer	Diameter, in.	Calculated Stiffness (Torsion) of Stabilizer	Roll Angle at 0.4g, deg	Roll Stiffness %
Production	11/16	100%	5.30	123.6
Small	37/64	50%	5.85	112.0
Large	49/64	150%	4.95	132.3
None	.. ..	0	6.55	100.0



■ Fig. 7 – Skid-pad roadability curve – Car No. 7144 – Effect of roll rate

The function of a stabilizer is to increase the roll stiffness without affecting the ride rate. In general, a car under the influence of a centrifugal force is subjected to an overturning couple which is resisted by the difference in road reaction on the inner and outer wheels and which, in turn, is shared between the front and rear wheels in proportion to their roll rate. Therefore, with total roll stiffness the same, the load transfer on the front and rear wheels should be, as a first approximation at least, in proportion to the front and rear roll rate. We should expect an increased roll stiffness on either the front or rear wheels to give a larger weight transfer and, from Fig. 3, somewhat lower cornering force. It is clearly evident from Fig. 3 that the maximum cornering force comes with equal load distribution; since the curve in Fig. 3 has an increasingly negative slope beyond a certain value, any general increase in load transfer must result in a loss in total cornering force.

The foregoing tabulation shows a total roll stiffness increase of 32% with the addition of the large stabilizer, all of which is added at the front wheels. From this we would expect a corresponding increase in front slip angle. The failure of this to occur may be explained in part by the fact that there is an increased amount of roll with the lower roll stiffnesses, resulting both in increased camber and in increased weight transfer because the center of gravity is above the roll axis.

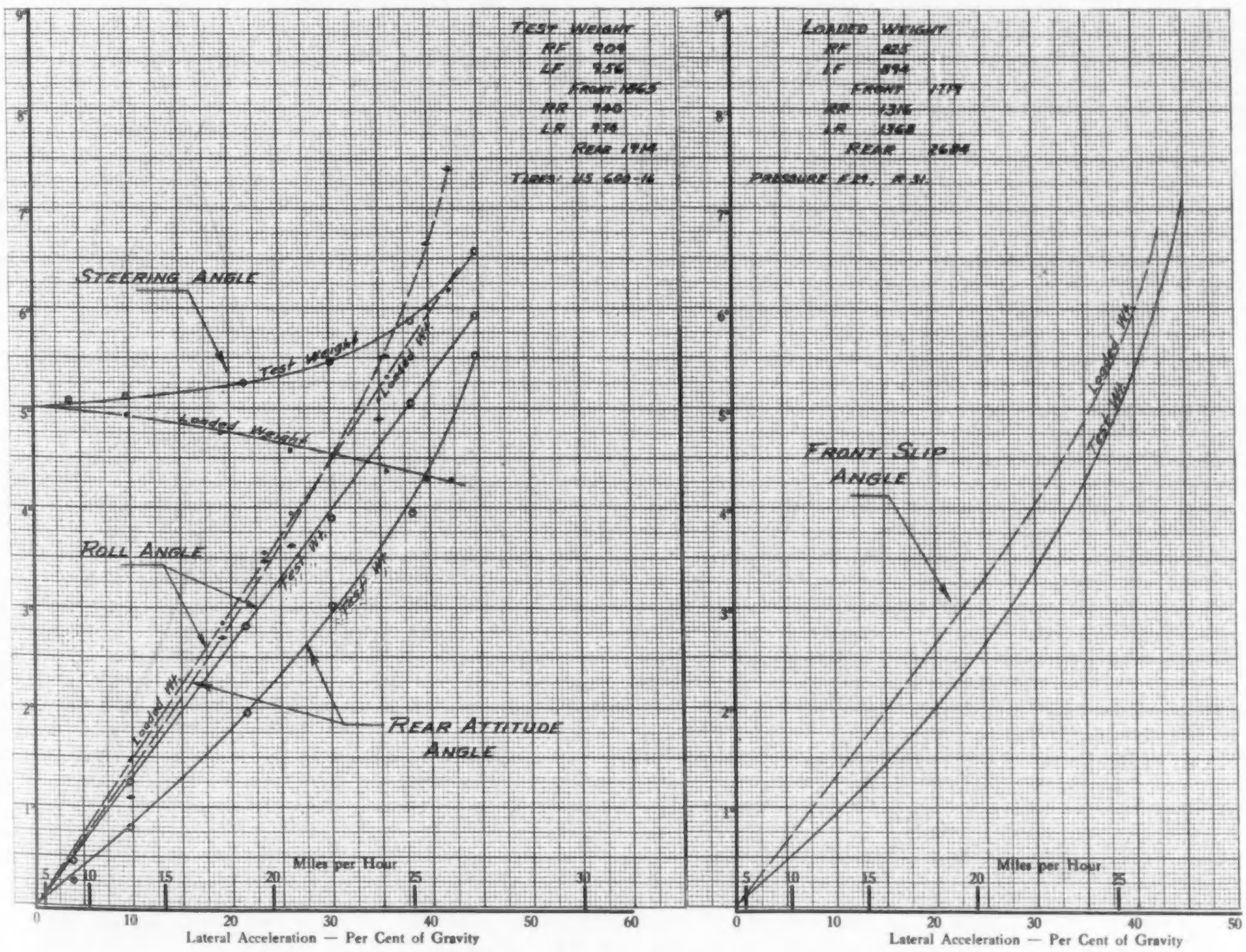
Fig. 8 shows the effect of load distribution. Here the standard-condition curves of Fig. 5 are compared with those on a test with 600-lb load in the trunk. Note that this added load, which is large but not unreasonable, changes the car from understeering to definitely oversteering. Our experience is that similar changes occur when load is added to the front, and that most cars can be made to oversteer or understeer by suitably altering the load. This particular condition is shown because it is easy to accomplish and because it represents a very common practice in private car operation.

Fig. 9 points out the effect of tire pressure on cornering. Four conditions of tire pressure were run:

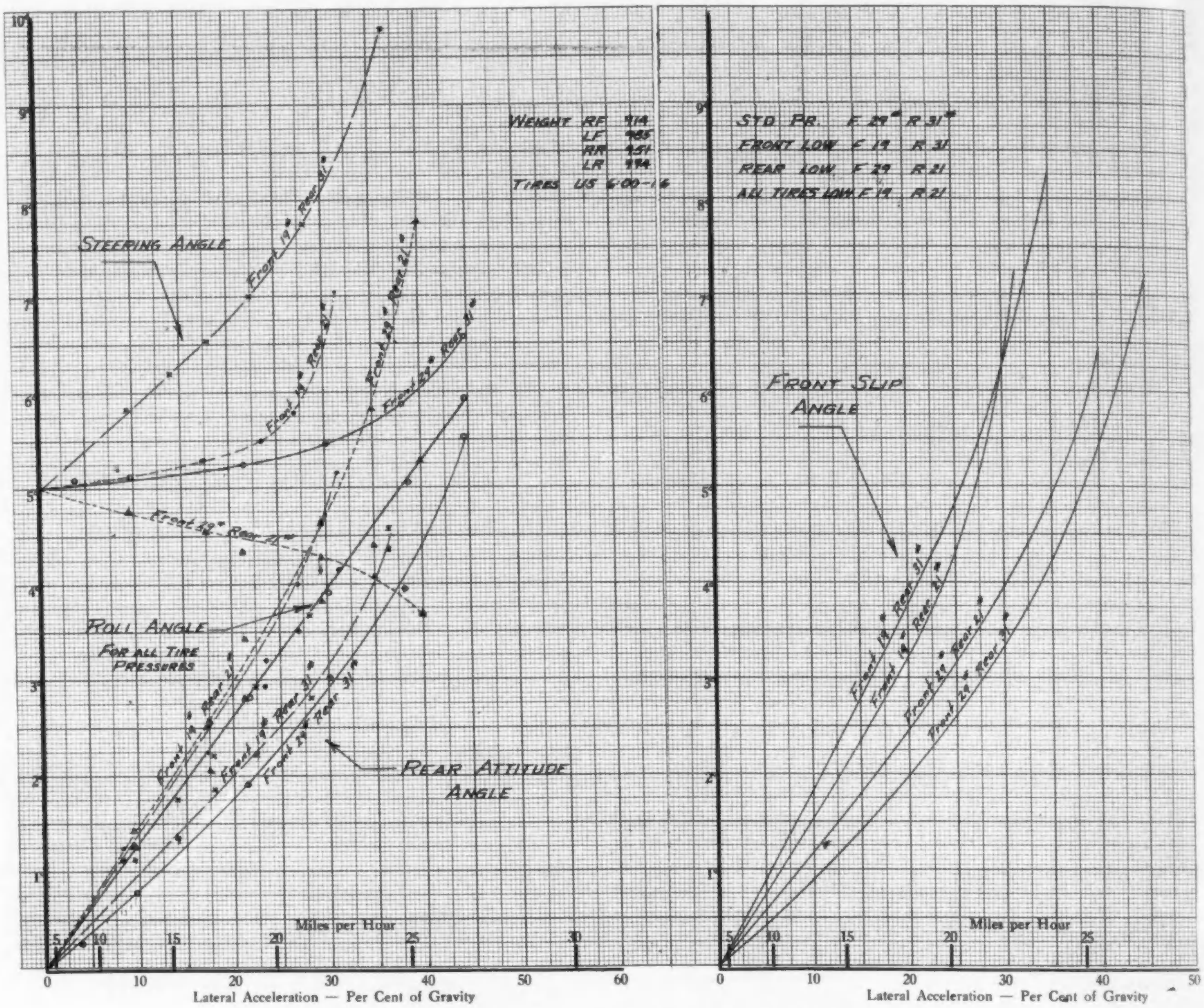
	Pressure, lb per sq in.	
Standard Pressure	F – 29	R – 31
Front Low	F – 19	R – 31
Rear Low	F – 29	R – 21
Both Low	F – 19	R – 21

These conditions occur commonly in private car operation.

The roll angle is not affected by tire pressure in the limits shown. The attitude angle is influenced very slightly or not at all by changes in front tire pressure and the front slip angle is influenced to some extent by changes in rear tire pressure. The car can be made definitely oversteering by dropping the rear tire pressure 10 lb per sq in. or highly understeering by dropping the front tire pressure 10 lb per



■ Fig. 8 – Skid-pad roadability curves – Car No. 7144 – Effect of load



■ Fig. 9 – Skid-pad roadability curves – Car No. 7144 – Effect of tire pressure

sq in. With both tires low, the behavior is very nearly normal at the start but becomes rapidly understeering.

The changes in steering characteristics from load and tire-pressure variation are more or less proportional to the variation.

It is quite clear that cars may not always be either understeering or oversteering, and that the ordinary maintenance and operation habits of the general public may shift the relative steering characteristics materially. The normal losses in tire pressure may make the car more understeering or oversteering with equal probability, and load increases almost invariably will change it in the oversteering direction because a greater portion of the load will be carried on the rear wheels.

The solution to this problem lies in carefully following the tire manufacturer's recommendation as to load and tire size and pressure. Cornering properties seem to vary in the same way as the factors considered in the load-size-pressure tables, so that large and small cars meeting tire manufacturer's recommendations are directly comparable on the skid pad. Large differences in front and rear load on current or projected designs must be balanced by inflation pressure differences or even different tire sizes, according to the tire table recommendations.

The aspect of the car in reference to the pavement plane is completely determined on the skid-pad test, and the significant relative motions of various parts of the car can be measured subject only to the limitations of space and ingenuity.

The test has been invaluable in pointing out what are the significant factors in car control problems and building up a terminology. It has fixed the order and range of the variables and indicated critical values and, by means of it, we have determined approximately the laws of variation. In addition to the important place it holds in the history of car control tests, it is still a fundamental design test of cornering characteristics.

The major part of the control problems, however, comes during states of transition, when the car is entering or leaving a curve or when it passes from one curve into a reverse curve, when the car on the straightaway is under the influence of cross winds or varying pavement slopes, or when the road surface is rough and irregular. In other words, our control problems are most acute when the side forces are variable in intensity and direction, and when the friction between the tires and the road changes abruptly and irregularly. These variations are purposely omitted from the skid-pad tests, and only part of the behavior of

■ Fig. 10 – Car set-up for checkerboard test

the car under the conditions of varying side force can be inferred from the skid-pad test.

It is perfectly obvious that, in order to find general control characteristics objectively, it is necessary to measure simultaneously the rates of change of slip angles, attitude, steering angle, roll and side forces. In other words, a dynamic transition test must be substituted for the essentially equilibrium skid-pad test.

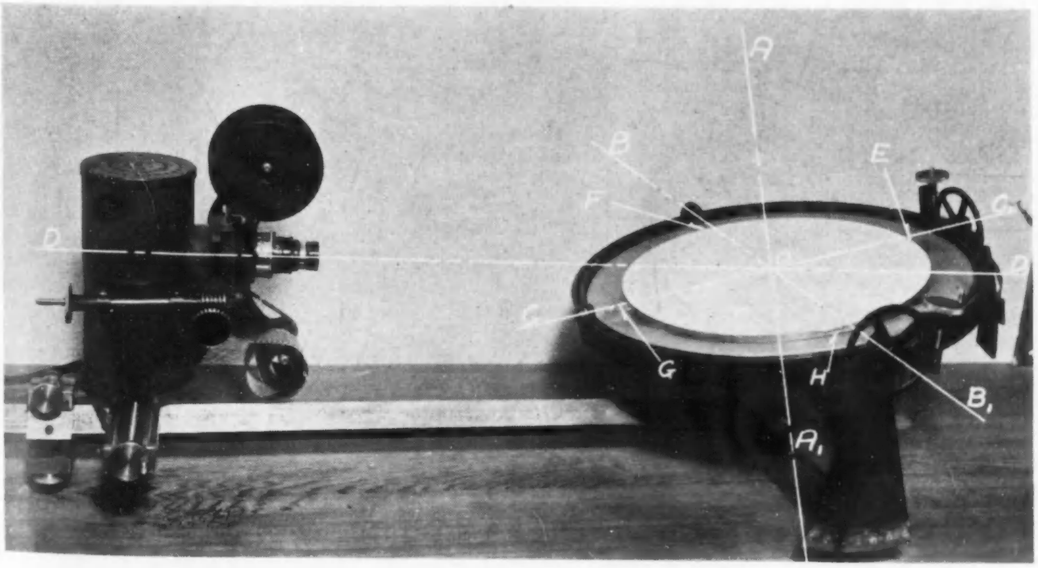
■ The Checkerboard Test

For a small scale transition test, the case of a car turning out to pass another was selected; that is, the car was driven straight in one traffic lane, then turned to drive a parallel course in the adjacent traffic lane, the various angles being measured continuously. The primary test problem, of course, is the method of measurement, which is complicated greatly as soon as the path is different from a circular arc, partly because the center of reference continuously varies and partly because the variables must be recorded continuously. A completely new recording system has to be substituted for the transit for measuring attitude and roll, and the steering angle must be recorded simultaneously.

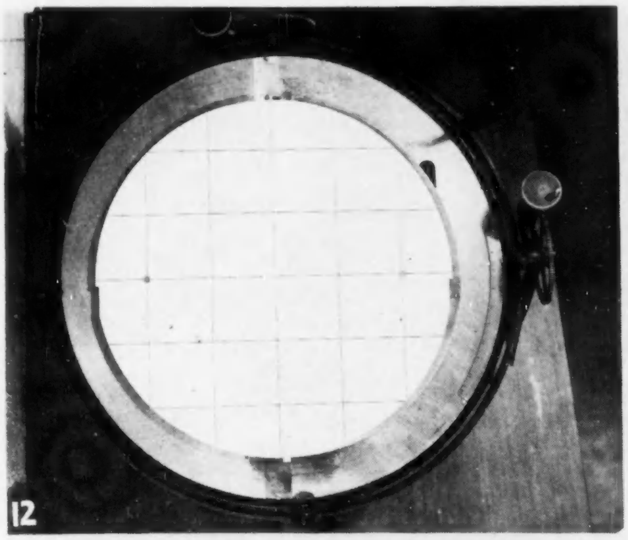
The method finally selected is photographic. The test course has a giant checkerboard with 2-ft squares painted on it, and a 35-mm movie camera is mounted on the car top, inclined so that the optical axis of the lens intersects the ground about 20 ft in front of the car. The camera used at present has a special shutter, with exposure time decreased to about 1/400th sec, and has a cross hair in the aperture plate at the optical axis.

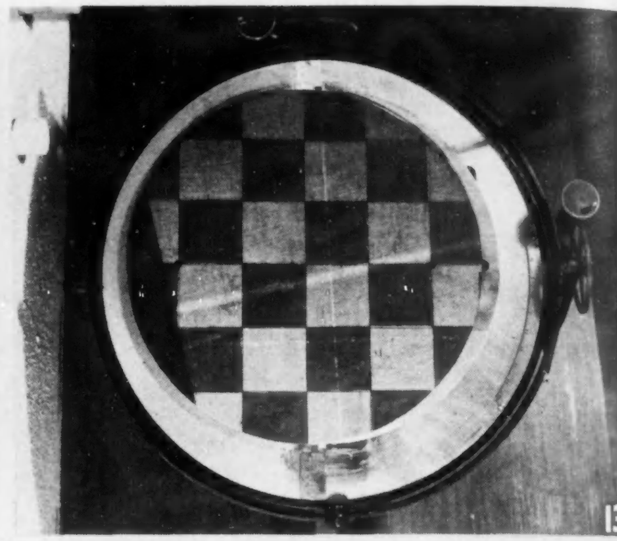
The car is set up with indicator dials measuring the steering angle of each wheel, rear axle steering and steering wheel angle mounted on the front of the car in the camera field; these dials are operated by Autosyn motors as in the skid-pad test. Fig. 10 shows the car set up for the test. Stops are put on the steering wheel, one to the right and one to the left of the straight-ahead position; they are adjusted by trial so that the driver can enter the checkerboard squarely, jerk the steering wheel against the right stop as quickly as possible, back against the left stop as quickly as possible, and come out square with the checkers but in the adjacent traffic lane. Stops are used so that the test can be repeated.

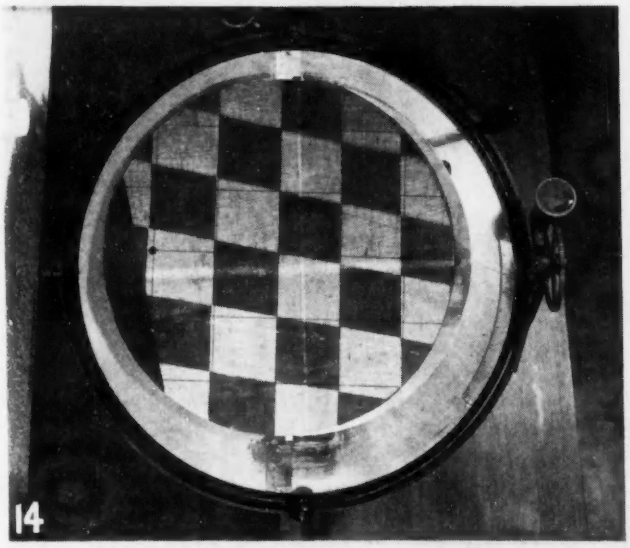
The test is run in this manner with the camera operating from before the turn is started until the car is straightened up. The camera, therefore, makes a simultaneous record

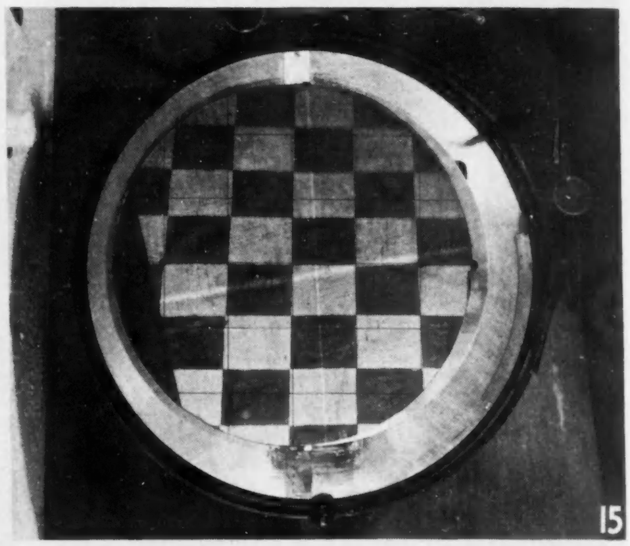


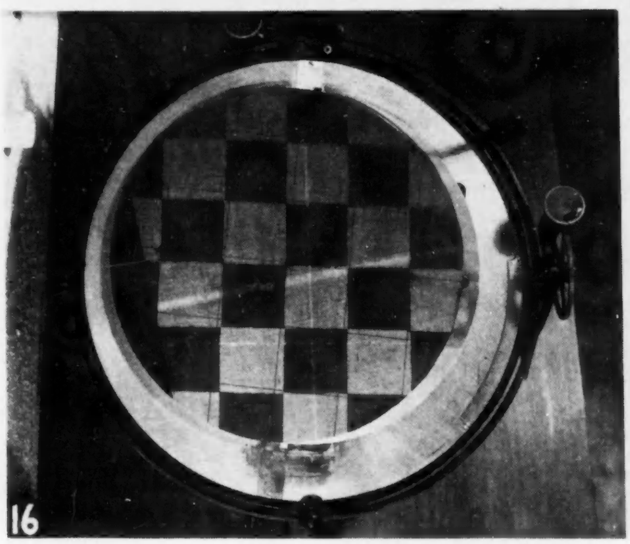
■ Fig. 11 – Apparatus used in reading film – Checkerboard test

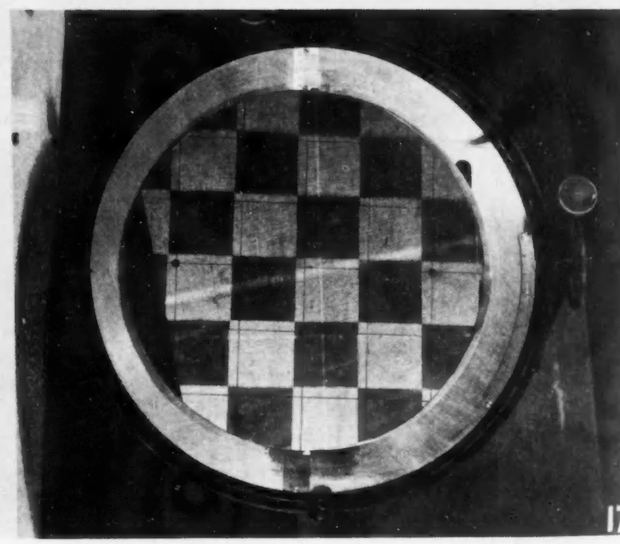












- Fig. 12 – Top of table used as screen – Checkerboard test
- Fig. 13 – Projected squares properly adjusted on screen – Checkerboard test
- Fig. 14 – Roll angle alone out of adjustment – Checkerboard test
- Fig. 15 – Pitch angle out of adjustment – Checkerboard test
- Fig. 16 – Course angle out of adjustment – Checkerboard test
- Fig. 17 – Both linear positions out of adjustment – Checkerboard test

of the indicator readings and the aspect of the checkerboard at the rate of 24 frames per sec.

The key to this test method lies in the realization that the pictures of a checkerboard pavement can be projected on a checkerboard screen so that the squares coincide only when the angle between the optical axis of the camera lens and the pavement is the same as the angle between the optical axis of the projector lens and the screen and when the distances along the optical axis between the camera and pavement and between the projector and screen are in the ratio of the sizes of the two sets of squares. In other words, the aspect of the camera lens to the pavement and the projector lens to the screen must be the same. To eliminate possible projection errors, the same lens is used in the camera and projector.

Fig. 11 shows the apparatus used in reading the film. At the left is a still projector for 35-mm film, mounted on a bar so that it can be moved back and forth. The object at the right is essentially a screen ruled in 2-in. squares and mounted in gimbals so that it can be rotated simultaneously about the three mutually perpendicular axes AA', BB', and CC', and with adjusting screws and circular scales so that it can be moved easily and its angular position determined. Means are also provided for allowing and measuring linear motion of the face along the BB' and CC' axes.

The line DD' is the optical axis of the lens and, in the zero position, the cross-hair image will fall on the screen at O. The linear movements are then adjusted so that the projected squares somewhat coincide with the rulings on the screen. The projector may have to be moved somewhat to get the size of the squares right; since the ratio of sizes of screen rulings and pavement squares is 1/12, the projector is 1/12 as far from the screen as the camera was from the point of intersection of the optical axis and pavement. The screen is then adjusted about its axes so that the squares coincide with the rulings, and the various angles are read. The car is started parallel to the direction of the squares, so that deviations measured about the AA' axis are the instantaneous angles between the car axis and the original course; this quantity is called course angle. Roll angle is measured about the CC' axis and declination plus pitch angle is measured about the BB' axis. The declination is found statically from the camera height and pitch by subtraction.

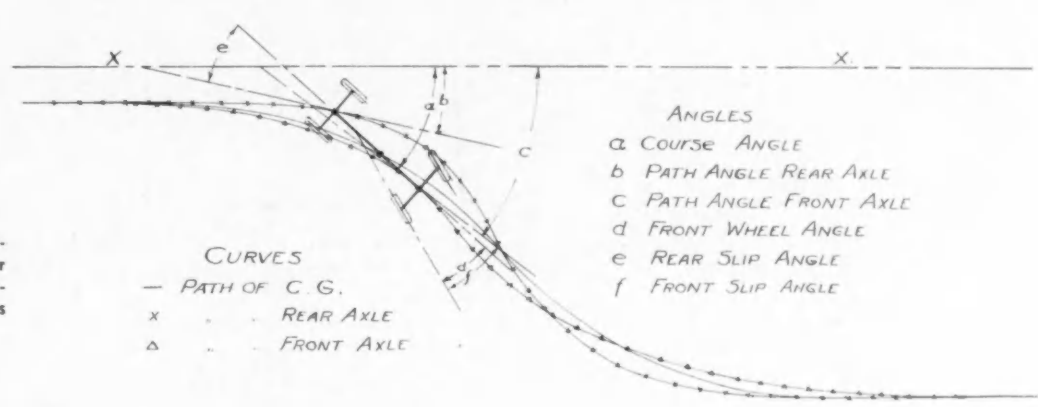
Fig. 12 shows the top of the table used as the screen. Fig. 13 shows the squares properly adjusted. Fig. 14 shows roll angle alone out of adjustment; note that the projection of the squares is oblique. Fig. 15 shows pitch angle out of adjustment; note that the projections are rectangular but of different sizes in the upper and lower parts of the picture. Fig. 16 shows course angle out of adjustment; the projections are square and parallel but not parallel with the ruling. Fig. 17 shows both linear positions out of adjustment; the squares do not coincide with the rulings.

It is evident that the aspect of the car with respect to the plane of the pavement can be completely determined with good accuracy by this method, and that the distance along the optical axis between the camera and a point on the pavement can also be found. Since the indicators of steering angle, steering wheel angle, and rear axle steering can also be photographed in each frame, it is evident that both the car position with respect to the pavement and the relative motion between any parts of the car can be simultaneously recorded at arbitrary increments of time.

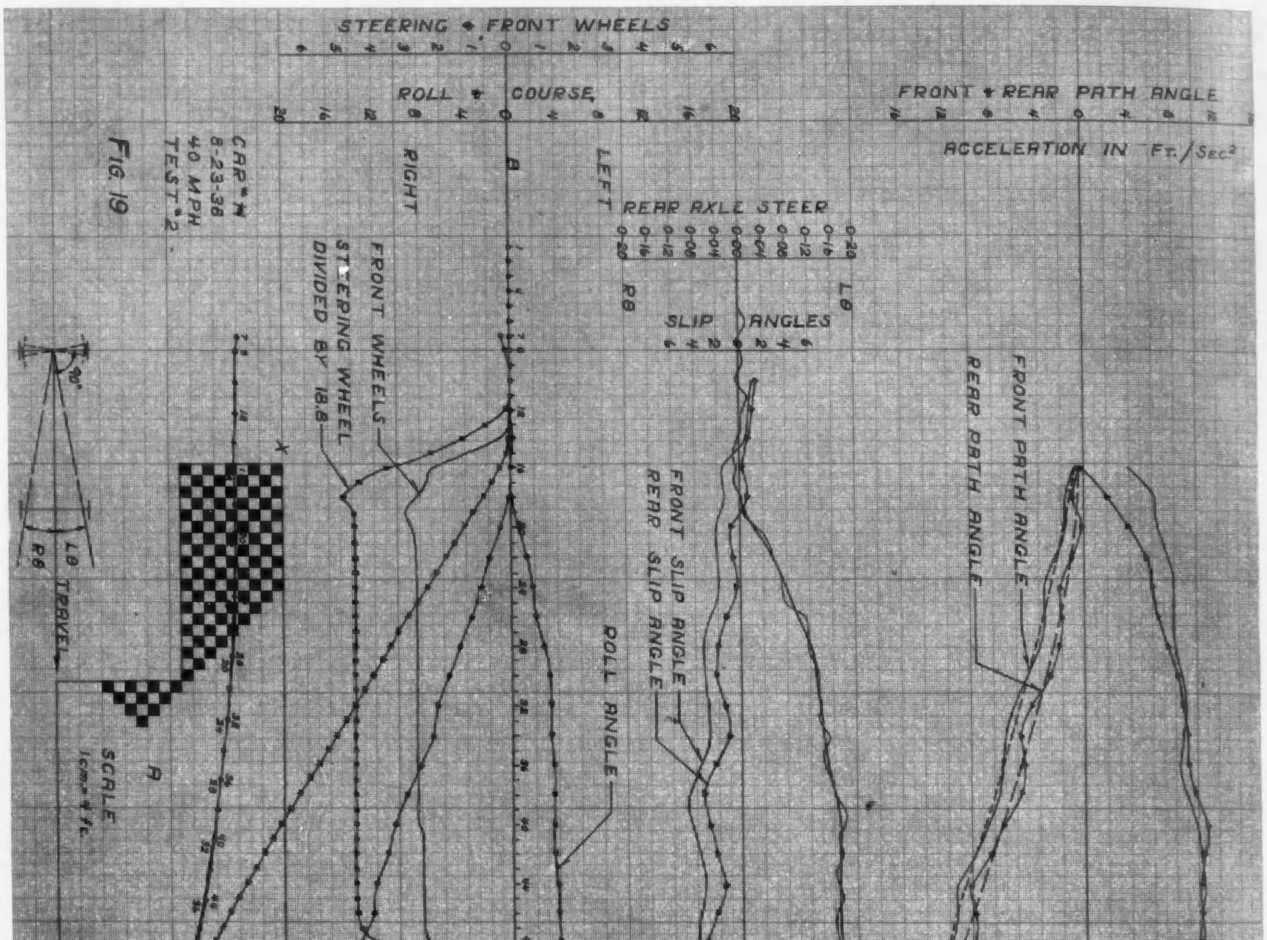
Since we can find the place where the optical axis intersects the pavement and can measure its length and the camera height and have the course angle, we can determine the position of the vertical line through the camera lens and, consequently, can find the position of the front and rear wheels at every frame. For practical purposes it may be assumed that the path of the front and rear wheels is found continuously. If we know the path, we can find the instantaneous direction of travel, and the angle between the direction of travel and the car axis at the rear wheels is by definition the attitude. We have measured the steering angle and can, therefore, calculate front slip angles. Furthermore, since we know the distance between consecutive points and the tangents at each point, we can, under the assumption of uniform curvature between alternate points, calculate the radius of curvature and the radial acceleration.

Fig. 18 is an exaggerated diagram of a car on the test course showing the angular relations. For simplicity we ordinarily consider the average of the front wheel angles; for special purposes we sometimes consider the individual wheels.

Fig. 19 shows the form in which the results are presented. The lower part of the sheet shows a scale drawing of the test course with the path of the optical axis and the point below the camera lens indicated. The scale above this is in terms of the frames on the film, one point for



■ Fig. 18 – Exaggerated diagram of car on test course showing angular relations



■ Fig. 19 – Form in which results of checkerboard test are presented

each frame, with the values plotted beginning with the first frame in which the optical axis intersects on the checkerboard. This is also a time scale and, if the speed is constant, a distance scale. All the values except path are plotted on it.

The first curves to leave the base line are the steering-wheel and front-wheel angles. The steering wheel angle divided by the average steering angle should give the front-wheel angle; the difference is due to steering linkage flexibility.

The next curve is the theoretical course angle. This is a hypothetical course which would be taken by a car with no slip angle as shown by the front wheel angle, and the comparison of its gradient with that of the actual course angle is a measure of understeer. This curve is found by accumulating the front-wheel angle. The actual course angle is measured about the axis AA' in Fig. 11. Understeer can also be measured by comparison of front and rear slip.

The roll-angle curve has an appreciable lag over the front-wheel angle and front acceleration and may lag behind the rear acceleration.

The front and rear slip angles are the difference between the path angles and the course angle; the path angles show considerable irregularity, which may be due in a large part to the probable errors in measurement. In order to compute lateral accelerations, a smooth curve is drawn through the path angle points. This probably gives a good approximation of the average lateral acceleration but leaves the peak values undetermined.

Either the skid-pad or checkerboard tests can, of course, be extended to a much larger scale to care for higher speeds or longer or different types of transition with no fundamental change in the procedure. The principal errors in both tests as conducted are due to the human factor; in the skid-path test the driver must follow the circular course very accurately for the basis of the calculation of side force, and in the checkerboard test the driver must perform the transition in the same manner to obtain comparable results on a series of cars. The accuracy would be improved considerably by increasing the scale so that the path deviations would be of little relative importance. Still greater accuracy would be attained by substituting a mechanical system of steering the car and eliminating the personal factor entirely.

These tests, it should be pointed out again, are tests of car control and not of car handling. The ultimate objectives are to determine quantitatively the attributes of the most precise control and by correlating these with the opinion of a large number of people to determine what characteristics give optimum handling.

Advantages of Express Highways to Trucks and Buses

THE advantages of a modern express highway in relation to bus and truck operation may not be too well understood. These highways will, in general, be constructed to higher design standards than is customary on ordinary highways. Outstanding features include the separation of highway and railway grades, thus eliminating all stops not required for fuel, two full lanes of pavement in each direction separated by a medial safety zone, easy grades, sweeping curves, and adequate shoulders. On a highway of this type where stops will be unnecessary except for fuel, it will be possible to obtain a high average speed in comparison with the top speed. The reduced time in covering a given distance between two points can be utilized by many types of truck operations to hold the good will of the shipper and thus retain present business, but more important it will result in the trucker or bus operator obtaining *new* business in greater volume.

The grade separation feature will permit operators to schedule trips on an economical basis suited to their particular businesses and the characteristics of the motive power being operated and to be certain that the schedules will be met during reasonable weather conditions. This will assure the shipper of "on-time" deliveries, which will be a powerful factor in retaining present and obtaining new business. The uncertainties in traffic, congestion and delays on the present highways makes the guarantee of "on-time" deliveries uncertain, and the trucker usually has to make a liberal allowance when working out schedules. This increases assured delivery time to the worst average driving time in order to "sell" the customer on the dependability of shipping by truck. Therefore, the consistency in operating conditions, freedom from stops, and traffic congestion on a modern express highway will be of inestimable benefit to the trucker and bus operator in obtaining and holding business.

An interesting comparison may be made between the Pennsylvania Turnpike and a closely paralleling main State route where, between common points, there are 939 road and street intersections, 25 stop lights and 11 railway grade crossings, while on the Turnpike there is none.

The provision of two full lanes in each direction will prove of great benefit in permitting continuous operation at a selected optimum speed inasmuch as it will be unnecessary to wait in line for a favorable opportunity to pass slower moving vehicles and momentum on upgrades will not be lost. This, together with grade separation, will enable close delivery schedules to be worked out.

Express highways undoubtedly will be constructed with easier grades than is customary on ordinary highways and this will effect operating savings in fuel and oil. Lower-powered trucks may be utilized for the same payload or increased payloads may be handled with present motive power as compared with existing highways. In territories where the grade differentials are large, additional savings will be available in the item of wear and tear, such as lessened tire and brake wear, reduction of strain on engine and transmission.

The provision of easy curves, divided roadways, grade separation and moderate grades will permit faster time schedules and quicker deliveries with less accident hazard than is experienced on present-day highways. This will increase dependability of service as well as lower insurance rates.

Excerpts from the paper: "Toll Roads and Truck and Bus Transportation," by Charles M. Noble, Pennsylvania Turnpike Commission, presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 7, 1941.

ECONOMICS of Substituting

WHETHER considered from cost or chemistry, the rubber industry from the standpoint of its basic material, natural rubber, has shown the greatest instability. All of the early developments on rubber substitutes were brought about by high prices of natural rubber or extreme shortages. The last twenty years have seen the price of rubber vary from 75c per lb to 2½c per lb, largely due to the varying effects of the English and Dutch restrictions. The high of 1910 was $3.06 per lb. It has been estimated that crude plantation rubber can be produced for a profit at 6 to 10c per lb. Thus it can be seen that industry has presented the chemist with the interesting problem of producing material of a comparatively stable price from year to year.

The earliest substitute in this country for natural rubber that achieved any importance was what is known as Guayule rubber obtained from the Guayule bush in Mexico and Southwestern United States. Considerable amounts of this rubber in the past appeared in the rubber factories but never in large enough proportions to become of commercial significance.

Chemists noticed early that rubber had the same empirical formula as isoprene and some even broke down the rubber into isoprene and repolymerized it into a rubbery-like mass. They, however, were unable to find commercial sources of isoprene, and this fact is still true today. Turpentine and other natural-occurring materials somewhat similar to isoprene also were tried as a basis for making synthetic rubber, but with no success.

GREATLY improved resistance to oils, greases, and solvents is the characteristic of synthetic rubbers of greatest interest to the automotive engineer, Mr. McCortney points out. The number of new synthetics, he reveals, has grown so rapidly that "we now have reached the 57 varieties." In some cases, he states, the synthetics can be used interchangeably – in other cases only one synthetic will do the job. "Thiokol," "DuPrene," "Neoprene," "Lignin-Neoprene," "Koroseal," "Buna," "Perbunan," "Chemigum," "Ameripol," "Butyl," and "Vistanex," are among the trade names of the synthetic rubbers discussed and compared in this paper.

Although capacities for the production of the various synthetic rubbers are being doubled and quadrupled as fast as humanly possible, Mr. McCortney draws attention to the fact that we are now facing an acute shortage of these materials. This shortage, he explains, is largely due to defense requirements, with the aviation industry being by far the greatest user. The Chrysler Corp. alone, he announces, can easily double its use of synthetic rubber without using it in a single place that would involve a cost penalty.

Early Synthetic Rubbers

A similar material, dimethyl butadiene, was used as a basis for synthetic rubber before the World War and, even then, complete tires of synthetic were produced. Another similar compound, butadiene, was used first during this period for polymerizing into a rubber-like material. During the World War Germany was, of course, shut off from all supplies of natural rubber and desperately tried to use methyl rubber to make up for her national deficiencies. The soft rubber was anything but satisfactory and very susceptible to extremes of temperature. By the end of the War the Germans were making synthetic methyl rubber at the rate of 150 tons a month, producing 2500 tons total.

It has now been over ten years since the first American synthetics appeared on the market. These were Thiokol and DuPrene, the latter re-christened Neoprene. The first Thiokol, known as *A*, was a reaction and polymerization product of ethylene dichloride and sodium tetrasulfide, both of which were and still are quite readily available in large quantities. Next came Thiokol *D*, of lower sulfur content and a different hydrocarbon. Type *F*, the latest development, has some of the qualities of both *A* and *D*.

DuPrene, now Neoprene, is based on research work of Father Nieuwland of Notre Dame. This synthetic truly can be said to be derived from coal, limestone, and sea-water, all of which are available in unlimited quantities. From coke and lime, calcium carbide is formed, which in contact with water gives acetylene. Acetylene is reacted to form mono-vinyl-acetylene, and the addition of hydrogen chloride gives chloroprene, which is in turn polymerized to form Neoprene.

The present Neoprenes now available are *E, G, GN*, and *I*. They differ both in means of preparation and composition in that some are co-polymers of chloroprene with other agents.

The modern German synthetics are improvements of the original German butadiene rubbers. Butadiene is a gas that can be made in Germany starting with coal and limestone the same as chloroprene but without the addition of hydrogen chloride. In this country it is being produced by the cracking of petroleum. The butadiene rubbers that have received principal attention in the last few years have been Buna *S*, a co-polymer of butadiene and the readily available styrene. This "rubber" is used in Germany purely as a substitute for natural rubber. Buna *N*, or Perbunan and Perbunan "Extra" are those of particular interest in this country, due to special properties which shall be taken up later. These are co-polymers of butadiene and acrylonitrile, the latter another material which falls in the class of ready availability. It is interesting to note that all of

[This paper was presented at a meeting of the Detroit Section of the Society, Detroit, Mich., Nov. 11, 1940.]

Synthetic Rubber in Automobiles

by W. J. McCORTNEY
Chrysler Corp.

the synthetics mentioned so far have their prototypes in Japan and Soviet Russia. In these countries they are made under different names and proudly proclaimed as their individual contributions to the field of synthetic rubbers. It is the opinion of most chemists that they deserve little or no credit for the basic development.

Vistanex is a polymer of isobutene and has little importance in the rubber industry except as a compounding material for other rubbers.

Koroseal is a plasticized vinyl chloride polymer that has been used in the automotive industry somewhat but has given way to the other synthetics for the more important uses. Koroseal has, however, found many other uses outside of the automotive industry where high temperatures are not encountered.

■ Many Recent Developments

The last year has seen the development of a bumper crop of American synthetics. Goodyear has its "Chemigum," Goodrich has its "Ameripol"[1] – both butadiene co-polymers; Standard Oil, its Butyl rubber, and also American-made Perbunan. Much publicity has been given synthetic rubber as a substitute for natural rubber in the case of disturbances in the Far East. At the present time the average cost of synthetics is around 60c per lb. It can be seen readily that the substitution of such an expensive material would be very much of an upset if used in tires for the automotive industry. It is true that everything possible should be done to develop synthetic rubber for a necessary war-time substitution in this country, but at the present time its use as a favored material in its own right is paramount. (Personally, I do not think the time has yet come when we can consider synthetic rubber for tires.)

At the present time, peculiar as it may seem, we are facing an acute shortage of synthetic rubber. Right here it might be appropriate to say that the Chrysler Corp. alone uses over 100,000 lb of synthetic rubber per month. The present production of synthetic rubber is probably around 800,000 lb per month which is extremely small in comparison with the total rubber used per month in this country. In fact, such a monthly production would not run one of the larger rubber companies for two days. The Chrysler Corp. easily can double its use of synthetic rubber without using it in a single place that would involve a cost penalty over material now being used. It is such developments that must come first to build up this country's capacities for synthetic rubber so that the price can be brought down to a point which will more nearly approach natural rubber. This will, in turn, prevent any violent upsets through substitutions for natural rubber. The present shortage of synthetic rubber is largely due to defense requirements, with the aviation industry being by far the greatest user. Capacities for the various synthetic rubbers are being doubled and quadrupled as fast as humanly possible. While this shortage should be relieved within the next few months, defense requirements may be increased still further. It is to be hoped, however, that increased production of the raw synthetic can keep pace with all of this. Such a temporary shortage should in no way be a deterrent for the development of new uses of synthetics. It has been pointed out before that such developments are necessary to lower costs and bring about freedom from dependency on overseas sources of such a critical defense material as natural rubber.

To make a quick survey of the immediate possibilities for increasing amounts of synthetic rubber each one will now be taken up briefly. Thiokol is the least critical of any of these materials in that it can be made by what chemists commonly term "washtub" reactions; that is, reactions in which no extreme pressures or temperatures are necessary. The production of organic chlorides and sodium polysulfides is simple and may be stepped up in short notice. The production of butadiene is dependent on the cracking of petroleum, and the production and purification of butadiene is the present bottle-neck in the production of Chemigum, Ameripol, and Perbunan. Production of butadiene rubbers and Neoprene also involves heavier and more complicated equipment than does the manufacture of Thiokol. However, Neoprene is now being produced in far the greatest volume, and it is understood that its production is being doubled beginning this month.

Butyl rubber, the youngest of the synthetics, is only now getting ready to make the jump from pilot plant to large-scale production. Advanced information states that butyl rubber can be made from mixed petroleum gases that are much simpler to obtain than butadiene.

■ Interest to Automotive Engineer

The big interest in synthetic rubbers to the automotive engineer is a greatly improved resistance to oils, greases, and solvents. A few years ago the problem of synthetics was quite simple in that the rubber technician had only to decide whether to use Thiokol or Neoprene. Now, with the new synthetics available and improved varieties of the old, we have reached the 57 varieties. In some cases the synthetics can be used interchangeably – in other cases only one specific synthetic will do the job. For the application of synthetics we will take up the uses on Chrysler Corp. vehicles since the author is, of course, much more familiar with these uses, and it is believed that more different synthetic rubber parts are used here than on any other manufactured product.

On our cars, Thiokol finds by far its greatest usage as a coating for gasket materials. All varieties of paper for gaskets are coated with a thin layer of Thiokol that is resistant to oil and will not break down under high tem-

[1] Hycar – Hydrocarbon Chemical & Rubber Co.

peratures. Under moderately high temperatures and pressures, however, the Thiokol becomes quite thermoplastic and flows into all variations of metal caused by stamping or by tool marks, forming a perfect seal between the paper and metal and also sealing the paper so that it acts as a solid stable material instead of a wick. Thiokol, due to its greater plasticity than the other synthetics and tendency towards high permanent set, must be used with fabric, metal, or some sort of reinforcement. Thus we use Thiokol in hoses where the fabrication prevents set, and the Thiokol imparts almost perfect resistance to solvents and oil swell and also to deterioration. Recent developments show that Thiokol may possibly be produced without this tendency towards extreme flow.

Neoprene, due to its longer manufacture and ready availability in the past, has long been the general-duty synthetic. The earlier Neoprenes such as *E,* while inferior to most of the other synthetics in swelling resistance, are simple to process and have long had general acceptance in the automotive industry. Seals that come in contact with oil and greases at not too high temperatures such as seals for suspension parts, engine seals, gearshift seals, axle seals, and such parts that do not meet extreme service, are commonly made from Neoprene *E.*

Recently Neoprene *E* has become particularly interesting to us in the manufacture of Ligno-Neoprene gaskets. Ligno-Neoprene, a development in our laboratories, is made from approximately 80 parts of lignin and 20 of Neoprene *E.* The lignins used are extremely cheap and available in almost any quantities as by-products from paper industry. For one part of rubber to take up four parts of filler is quite remarkable to any rubber chemist, particularly when the rubber retains much of its flexibility. These Lignin-Neoprene gaskets show little or no deterioration under heat up to 650 F, and in hot engine oils. Such gaskets are used for valve-spring cover gaskets, axle pinion carrier gaskets, remote-control gearshift housing gaskets, and other such uses. Neoprene *E* is also used in parts where deterioration by oil would be harmful, but swelling is desirable such as in the crankshaft rear-bearing cap gasket. The newer Neoprenes have become progressively better in oil resistance until the newest, Neoprene *I,* is on preliminary examination equivalent to butadiene polymers for oil resistance. Neoprene *GN* has found a field because of its good oil resistance and ability to be formulated into compounds that have good resistance to cold. One such compound is that used on the propeller shaft boots that replace leather. These boots have at least five times greater life than the old leather boots when used under the most severe conditions. You will note that for many of the seal applications, Neoprene also is used as a substitute for leather, a material that is variable in nature. Leather is also unsatisfactory where temperatures reach more than 220 F and where water and moisture are present. Neoprene *I* due to its extreme newness has not been used to date but should fulfill many important functions on the automobile in the future. Neoprene latex can be handled similarly to the latex of natural rubber and is now being used to dip cork gaskets to form more perfect sealing mediums.

Chemigum has only recently been announced but we have been using it on our automobiles for over a year. Its first usage was brought about through the failure of natural rubber and the then-available Neoprenes to withstand the fancy anti-freeze solutions being sold throughout the country. These solutions contained up to 10% mineral oil and, when diluted with water, attacked all rubber parts, particularly those near the upper surfaces of the radiator coolant. Such conditions caused Neoprene and rubber to swell at a remarkable rate. Without the Chemigum compounds the pressure-cap gaskets failed to operate, the thermostats remained open, and heaters refused to function. In the case of the bypass hose on some of our truck models, Neoprene and Chemigum had to be substituted for the rubber. With the thermostat closed, a layer of oil formed in the hose causing complete failures sometimes within ten days when natural rubber was used. Thus synthetic rubber prevented the necessity of a tear-up and complete changes in design. It also would be desirable to use synthetic rubber for all radiator hoses but as yet cost penalties have been too much and none of the automotive manufacturers have been able to provide anything but specially compounded natural rubber. Chemigum also may be used interchangeably with Neoprene on many of our synthetic-rubber parts. It has been possible with Chemigum to obtain diaphragm materials that were equivalent to the best natural rubber in cold resistance and completely unaffected by gasoline or gasoline vapors. Such a diaphragm is now being used in the operating unit of Chrysler Corp. automatic transmissions.

Chemigum, Perbunan, and Ameripol are considerably harder to process than natural rubbers and the more complicated shapes are commercially impossible to make.

The B. F. Goodrich Co.'s recently announced "Ameripol" is quite new to the automotive industry. It should find ready acceptance since we have already found it possible to substitute Ameripol in some cases for Neoprene when threatened with a shortage of the latter material. Ameripol, in common with the other butadiene co-polymers, has very excellent resistance to oil and solvent materials.

Perbunan is very similar in its behavior to the other butadiene rubbers and was finding extensive usage before the present war shut off the supply from Germany. Interest in Perbunan no doubt will be revived as soon as Standard Oil has available American-made Perbunan.

As has been mentioned before, Standard Oil's Butyl rubber is the baby of the synthetic-rubber family. Little of this rubber has been available for even laboratory work but advanced information has been particularly interesting. While it lacks the higher resistance to oils and gasoline, its greater aging resistance, with the advanced information that this rubber remains rubbery at extremes of temperatures where natural vulcanized rubber would be brittle and useless, will undoubtedly offer many interesting possibilities to the automotive engineer.

EXCERPTS FROM DISCUSSION

Properties of Neoprene KN

— E. R. Bridgewater

Head, Rubber Chemicals Division, E. I. du Pont de Nemours & Co., Inc.

NEOPRENE *KN* is a Neoprene which, as polymerized originally, is quite tough and quite hard. It has the unusual properties of being plasticizible by the addition of very small amounts, two portions or so, of certain chemicals, particularly organic basis. This plastic material, incidentally, is a compound containing about 45% of Neoprene and the balance being fillers. This exceedingly highly plasticized synthetic rubber is a product we have been having a lot

of fun with. What its commercial application may be I do not know, but we have been able to do a lot of tricks in the laboratory that have been very interesting to us. Let us take, for example, half of a golf ball or rather a ball of Neoprene vulcanized in a golf-ball mold. I should like to tell you how it was made because it illustrates quite amply some of the unusual properties of this material. We roll the material into a rough ball and put it in the golf ball mold, put the cover on the mold, do not place any bricks on it or put it in a hydraulic press, and put it in a warm place near a radiator over night. Or, if you choose to add a little accelerator before you put it in the mold, you can dispense with the radiator. The next morning you open the mold and have a perfectly molded little ball.

Now due to the extreme plasticity, this type of Neoprene can be made into cements of very unusual properties, cements which are almost as non-viscous as water. In this form it has proved to be a very interesting calking compound, a material for laying up acid-proof brick, glass bricks, and so on. It is adaptable in general to molding rubber. That is, you can push this material which you understand contains no volatile solvent, into any kind of a crack or crevice, allow it to air-vulcanize, and the result is a fully vulcanized piece of Neoprene such as the golf ball just described which has, in general, all of the properties of ordinary Neoprene.

More Data on Thiokol

—J. W. Crosby

Manager, Technical Service, Thiokol Corp.

WE have always been able to keep ahead of requirements with Thiokol and, in these days of national defense and shortages of material, we are still able to keep our customers going and think we shall be able to take care of any defense requirements as well.

In regard to these reactions I might point out that, while the rubber industry only uses several types of Thiokol of the rubber-like variety, owing to the possibility of varying neopolysulfide and hydrocarbon and a mixture of hydrocarbons, you can get a variety of materials. Some of them are rubber-like and some are not. Some have found a place to a small extent in the plastic industry and some are still in the experimental stage.

In reference to the properties of the material, the distinct advantages are probably excellent solvent resistance, good aging properties, and extremely high permeability. On the other hand, it has disadvantages. Public enemy No. 1 is cold flow as Mr. McCortney mentioned and also it has a rather strong odor, resembling sweet violets to some people. For that reason, for the cold flow particularly, its application is limited in the automotive industry to the parts that Mr. McCortney mentioned.

Now our objects of research are very simple: 1. To eliminate the cold flow and; 2. to overcome the odor. We have made some progress on the odor and we think that very shortly we will have some improvement for our long-suffering friends in the rubber industry. Now the other material, as far as cold flow is concerned, is purely experimental at this stage, but we hope before long that we will have materials which will have more general application than do the materials at the present time.

Synthetic Rubber for Tires

—W. L. Semon

Research Director, Hydrocarbon Chemical & Rubber Co.

SYNTHETIC rubber must be considered under two heads. First there is the problem of getting a replacement for natural rubber. Approximately 75% of the rubber used in this country is used in tires. Therefore, we are faced with the supply of rubber suitable for the manufacture of tires and it must be realized that this rubber must compete with natural rubber which is maintained on an artificially high price due to restrictions on the plantations and by the producers. In the first place, it can be seen that synthetic rubber, to compete in this field, is facing rather a severe competition.

Now careful estimates have been made as to the cost of building plants for making this type of synthetic rubber and estimates have been made of how much this rubber would cost if produced on various scales. It has been estimated that it would be uneconomical to build and operate a plant for the production of this type of rubber if it were to be made on a scale smaller than 100 tons per day. On this basis the material can be produced for approximately 25c per lb. At the present time natural rubber is selling for around 20c per pound. Now this would seem to be a substantial premium over the price of natural rubber. However, if we calculate the increase in cost which the use of such rubber would make in the price of a tire, we reach the rather astonishing figure that it would increase the cost of materials in the tire, less than 4% of the list price of a first line tire. Variations greater than this would occur due to fluctuations in the price of rubber and of cotton and the public has come to accept these variations with very little thought.

I want to submit to you, therefore, the importance of developing our own source of synthetic rubber for use in tires, for it will set a ceiling which will get a rubber value beyond which the price of natural rubber will not rise. You are all acquainted with the introduction of the Ameripol tire, which was put on the market some months ago. There is one rather interesting property of this material and that is its high resistance to tear. Another specific advantage of tires of this type is the fact that they have quite high resistance to skid – considerably higher than tires made of natural rubber in the same mold design.

Qualities of Butyl

—I. E. Lightbown

Commercial Rubber Department, Standard Oil Co. of N. J.

BUTYL rubber was invented and developed entirely in the United States by the Standard Oil Development Co. It is entirely different from other known synthetic rubbers. It is 100% hydrocarbon synthetic polymer. This means, of course, that, as in other synthetic rubbers, the resins, proteins, and water-soluble constituents found in natural rubber are completely absent from Butyl rubber. It is not oil-resistant, being like rubber in this respect, although it has strange quirks of character – more resistance to such simple aromatic solvents as benzene.

From a theoretical standpoint, as well as from a practical one, it has great interest. I say from a theoretical standpoint because Butyl rubber before vulcanization has an unsaturation lower than would be expected from a material which can be vulcanized by ordinary methods.

Specifically, Butyl rubber has only 1 to 2% of the chemical unsaturation of natural rubber. I need not dwell on the fact that theories of vulcanization and of elasticity may have to be modified greatly to allow for an inclusion of the fact that Butyl rubber vulcanizes and is elastic. The older theories which would have forces exerted by residual double bonds in natural rubber as being responsible for reversible elasticity must be discarded. The assumption that the rubber molecule had to be duplicated fairly closely to cause vulcanization must also be discarded.

After vulcanization, Butyl rubber has become essentially a chemically saturated hydrocarbon. One of the major failings of natural rubber is that only a small percentage of the available unsaturation is satisfied during vulcanization. Recent studies concerning this phenomenon have shown that, when more than about 2% of the available double bonds are linked with sulfur, the natural rubber begins to over-cure. This leaves a great many double bonds in vulcanized soft rubber still active and ready to combine with oxygen, ozone, acids, and so forth.

If you keep this difference clearly in mind, you can understand some of the properties of Butyl rubber. You can understand, for instance, why it is extremely resistant to aging without our resorting to artificial methods, such as the addition of resisters. You can understand why even strong sulfuric or nitric acids do not affect it. You can understand why ozone, which cracks rubber within a few seconds or minutes, has almost no effect on Butyl rubber. It has a specific gravity of 0.91, about as low as, or slightly lower than, natural rubber, somewhat lower than Perbunan which is 0.96, and considerably lower than some of the other synthetic rubbers which are well above 1.0.

Butyl rubber can be handled in regular factory equipment without major modifications. It molds particularly well, following even the most intricate mold designs and without laminating. I think it can be safely said that it is better than rubber in this respect.

Specification for Chemigum

—R. P. Dinsmore

Technical Director, Goodyear Tire & Rubber Co.

IF you ask what Chemigum is like, I might give you a simple specification like this: It has an amber color; its texture is somewhat of the milled smoked sheet; its odor is faintly aromatic; it has an extrusion plasticity of 200 lb and 94 C. of 0.1 to 0.3 cc/min. It has

(Concluded on page 106)

LUBRICATING OILS for

PURPOSE of this paper is to discuss a number of the factors that should be considered in the selection and use of lubricating oils with particular emphasis on the need for engine or service tests, as the usual specifications are only identifying tests indicating general characteristics.

The authors show that some of the performance characteristics of lubricating oils, such as limiting cranking temperatures, oil mileages, and gear-shifting temperatures, can be predicted from viscosity determinations, but the other items of interest to the users, such as stability, ring-sticking, gumming, and wear, cannot be predicted from inspections. Data are presented to show how compounded oils meet the severe present-day conditions of increased mechanical and thermal loads imposed on the lubricating oil especially in high-speed diesels. Compounded oils, they point out, cannot be judged by simple laboratory tests, and should therefore be selected on a basis of their performance in full-scale engines.

Such physical tests as gravity, color, flash point, fire point, viscosity, pour test, Conradson carbon, and Neutralization Number are discussed in the light of their present usefulness. The significance of viscosity and viscosity index is explained fully.

THE problem of evaluating and selecting internal-combustion engine oils to assure satisfactory performance and minimize failures is a complicated one, and involves consideration of physical inspections, chemical analyses, specifications, SAE numbers, advertising claims, and service experience. These data often appear to be contradictory and, to the practical man who is charged with the responsibility of maintaining operating costs at a minimum, the selection of the best lubricant for a given service is not always simple. It is the purpose of this paper to discuss a number of the factors that should be considered in the selection and use of lubricating oils and, more particularly, to point out the need for engine or service tests, as the usual specifications are only identifying tests indicating general characteristics.

■ Inspections

In the manufacture of lubricating oils for internal-combustion engines, a number of special physical tests are used which are often referred to as inspections. These tests are valuable to the manufacturer for maintaining uniformity of appearance and quality, but their relation to the service value of the lubricant is less direct. The various tests and their service significance may be described as follows:

Gravity – The API gravity, given in degrees, is a means of expressing the density of an oil. The relationship between specific gravity and API gravity is:

$$\text{Specific Gravity} = \frac{141.5}{131.5 + \text{API Gravity}}$$

The API gravity may be used as a rough measure of the degree of paraffinicity of an oil but, in this day of solvent refining and synthetic products, it is unreliable for showing the source or type of the oil. Its greatest value, therefore, lies in its use as a refinery control test.

Color – The color of an oil has no special significance, although most refiners control color carefully to avoid questions with respect to variation in quality.

WHEN t (SAYBOLT VISCOSITY) IS 100 OR LESS

$$\text{KINEMATIC VISCOSITY (CENTISTOKES)} = 0.226\,t - \frac{195}{t}$$

WHEN t (SAYBOLT VISCOSITY) IS OVER 100

$$\text{KINEMATIC VISCOSITY (CENTISTOKES)} = 0.220\,t - \frac{135}{t}$$

$$\text{SPECIFIC GRAVITY} = \frac{141.5}{131.5 + \text{A.P.I GRAVITY}}$$

$$\text{ABSOLUTE VISCOSITY (CENTIPOISES)} = \text{SP. GR.} \times \text{CENTISTOKES}$$

■ Fig. 1 – Determination of absolute viscosity from Saybolt viscosity and API gravity

Flash Point – The flash point of an oil is the temperature at which it gives off sufficient vapor to form, momentarily, a combustible mixture with the air above the oil. In the ASTM Flash Point Test a flame is used to determine whether or not the oil is giving off a combustible mixture, and the flash point is the temperature at which the first "flash" is noted. This test should not be confused with the spontaneous ignition temperature, which is the temperature at which the oil will burn without ignition from an outside source.

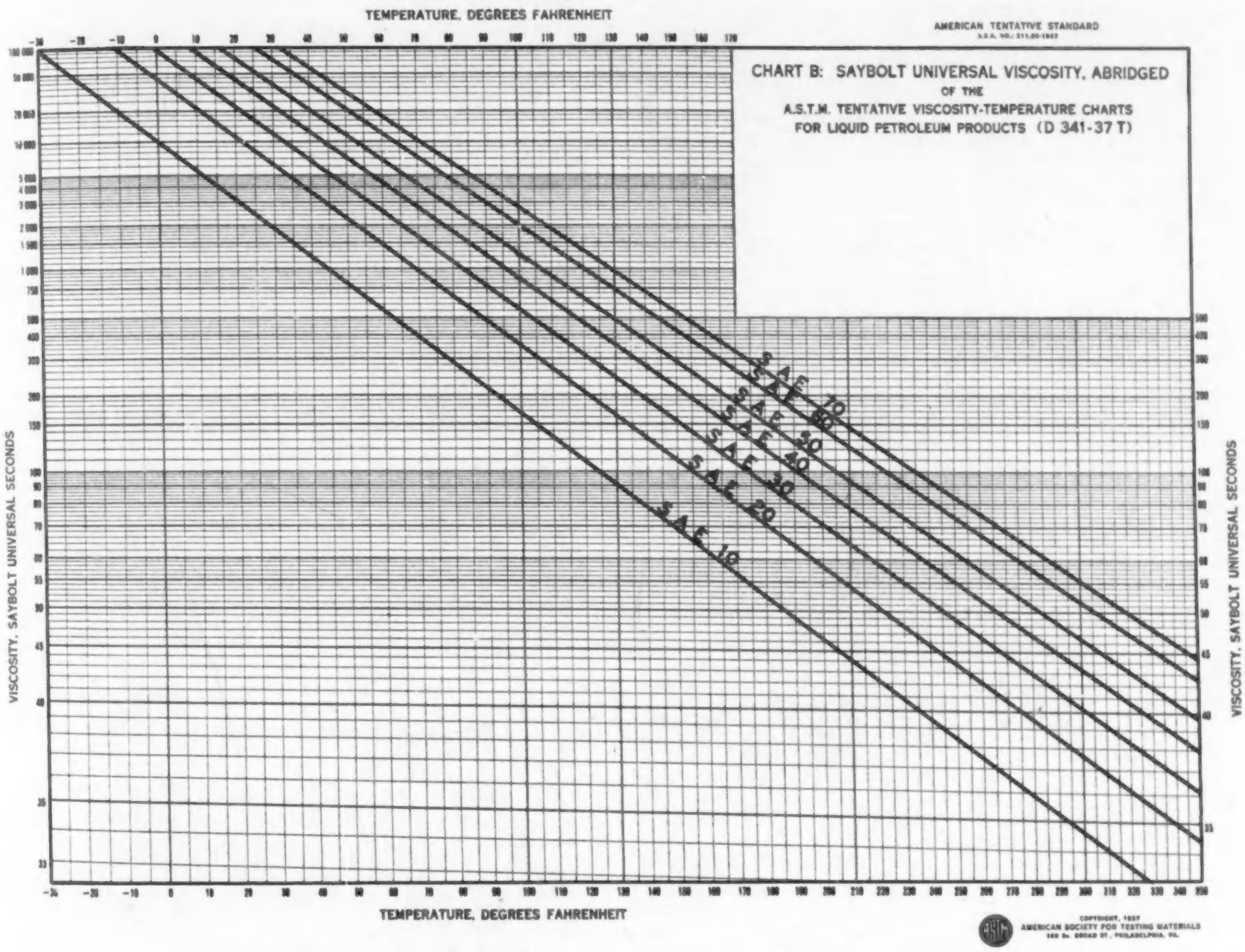
Fire Point – The fire point is the temperature at which an oil gives off vapors at a sufficient rate to support continuous combustion. The fire point of most oils is higher than the flash point by approximately 60 F.

The flash and fire points are indices of the volatility characteristics of the lightest fractions of the lubricating

[This paper was presented at the Pacific Coast Regional Meeting of the Society, Seattle, Wash., Aug. 16 and 17, 1940.]

Internal-Combustion Engines

by LLOYD H. MULIT and FRANK W. KAVANAGH
Standard Oil Co. of Calif.



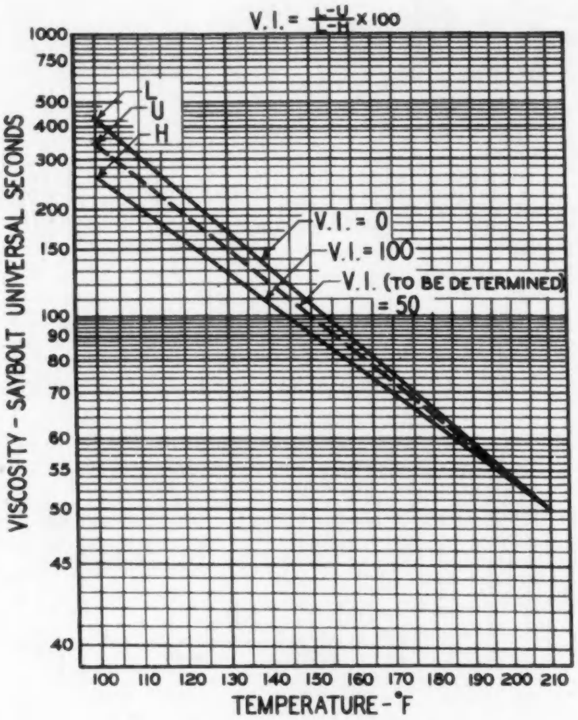
■ Fig. 2 – Viscosity-temperature characteristics of typical motor oils of various SAE grades

oil. These inspections are only indirectly related to the oil-consumption characteristics of the oil as will be explained more fully in a later section. At the same time, small amounts of dilution will cause large reductions in flash and fire points without changing any essential characteristics of the oil.

Viscosity – The service characteristics of lubricating oils are closely related to viscosity characteristics, and the influence of viscosity on performance will be discussed in detail in a later section. The viscosity used in specifying oils is usually given in terms of seconds Saybolt Universal, and is essentially the time required for 60 cc of the oil to flow through a standardized capillary tube at a given temperature. The SAE classifications are given in terms of Saybolt viscosities and, although numerous other viscosity systems have been devised, it is believed that the Saybolt Universal system is eminently satisfactory for purposes of determining and specifying viscosities of oils to be used in different services.

Pour Test – The temperature at which an oil ceases to flow when chilled in a 4-oz bottle is its pour point. This test correlates reasonably well with the ability of the oil to flow to the suction side of an automotive oil pump. An oil should not be used at a temperature below the pour point.

Conradson Carbon – The Conradson carbon, sometimes called carbon residue, is the percentage of carbonaceous material remaining after an oil has been evaporated rapidly



■ Fig. 3 – Illustration of viscosity index

in the absence of air. Lubricating oils are composed of approximately 85% carbon and 15% hydrogen, and it is obvious that the Conradson carbon is not a measure of the actual carbon content, but is an indication of the residue that will remain after destructive distillation of the oil. The Conradson carbon test may be used to predict the carbon-forming tendencies of an oil in service in comparison with oils of similar oxidation stability and volatility characteristics, but does not by itself form an absolute measure of carbon formation. It is also an excellent test for used oils to determine the amount of change that has taken place in service.

Neutralization Number – The Neutralization Number of an oil is the number of milligrams of normal potassium hydroxide required to neutralize 1 g of the oil. Relatively high values may be obtained for any acidic materials, some of which are very harmful to engine operation while others have no deleterious influence. Certain compounding materials, such as oil-soluble soaps frequently used in diesel lubricants, may be destroyed by the chemical action of the potassium hydroxide, and oils containing these materials show fictitiously high neutralization numbers. The neutralization number of a used uncompounded lubricating oil is a good criterion of the amount of oxidation that has taken place, but it does not form a reliable measure of the corrosiveness of an oil because the acids formed by the oxidation of oils vary, some combining readily with bearing materials and others being relatively inactive.

■ Significance of Viscosity

Viscosity is that property of a fluid by virtue of which it resists shearing force. It is expressed in fundamental units as poises or, in the case of kinematic viscosity, as stokes but, for practical use, the arbitrary Saybolt scale is very satisfactory. The units in this system are seconds, and viscosities expressed are seconds Saybolt Universal, abbreviated S.S.U.

Saybolt viscosity is a convenient and usable system because it is determined in an extremely simple form of testing apparatus. Its utility has been extended by the ASTM temperature-viscosity chart D341-37T on which the viscosity-temperature relationship is a straight line. This chart is based on the empirical formula, log (centistokes $+ 0.8) = A \log T + B$, where T is in deg absolute (deg F $+ 460$) and A and B are constants depending upon the viscosity properties of the oil. In actual practice, however, the performance of an oil is dependent on its absolute viscosity. The relationship between the Saybolt viscosity and absolute viscosity may be determined by the formulas shown on Fig. 1. Due to the very complex nature of viscosity and viscometry, the formulas included on this chart are empirical, but are eminently satisfactory for engineering calculations. The formulas shown were tentatively adopted by the ASTM but were later discarded in favor of a table which now appears in the ASTM Standards. They are, however, within 1% of absolute values. To obtain the viscosity of a lubricant in a service unit to an accuracy of 1%, the temperature of the unit must be known to the nearest 0.2 F, which is very difficult to determine in service.

S.A.E. VISCOSITY NUMBER	VISCOSITY RANGE - SAYBOLT UNIVERSAL, SEC			
	AT 130° F.		AT 210° F.	
	MIN.	MAX.	MIN.	MAX.
10	90	LESS THAN 120	---	---
20	120	" " 185	---	---
30	185	" " 255	---	---
40	255	---	---	LESS THAN 80
50	---	---	80	" " 105
60	---	---	105	" " 125
70	---	---	125	" " 150

W GRADES - NOT S.A.E. NUMBERS

GRADE	VISCOSITY AT 0° F. MIN.	VISCOSITY AT 0° F. MAX.
10 W	5,000	10,000
20 W	10,000	40,000

■ Fig. 4 – Crankcase oils – SAE viscosity numbers

Hydrocarbon lubricating oils are characterized by very rapid change in viscosity with change in temperature. To illustrate this point, oils of various SAE grades have been plotted on Fig. 2. This plot shows that, with the SAE 10 grade oil, decreasing the temperature from 210 F to 0 F changes the viscosity from 44 to 10,000 S.S.U. In a similar manner the SAE 70 grade oil is increased in viscosity from 145 to 100,000 sec Saybolt as the temperature is decreased from 210 F to 32 F.

Close examination of the chart will show that, in the higher ranges of temperature, approximately 70 F is required to double the viscosity while, at lower temperatures, the viscosity doubles for each 10 F.

While all hydrocarbon lubricating oils show a rapid change in viscosity with temperature, this rate of change varies with different types of oils, and a system known as viscosity index (V.I.) has been developed for expressing this relationship. This system was developed by Messrs.

Davis, Lapeyrouse, and Dean and is explained in detail in the *Oil and Gas Journal* of March 31, 1932[1]. The viscosity-index system is illustrated in Fig. 3. To determine the viscosity index of an oil having known Saybolt viscosities at 100 and 210 F, a set of tables is used (as included in the magazine article[1]) to obtain the viscosity at 100 F of a 0 V.I. oil (L) and the viscosity at 100 F of a 100 V.I. oil (H), each having the same viscosity at 210 F as the oil for which the viscosity index is desired. The viscosity index is then determined from the formula:

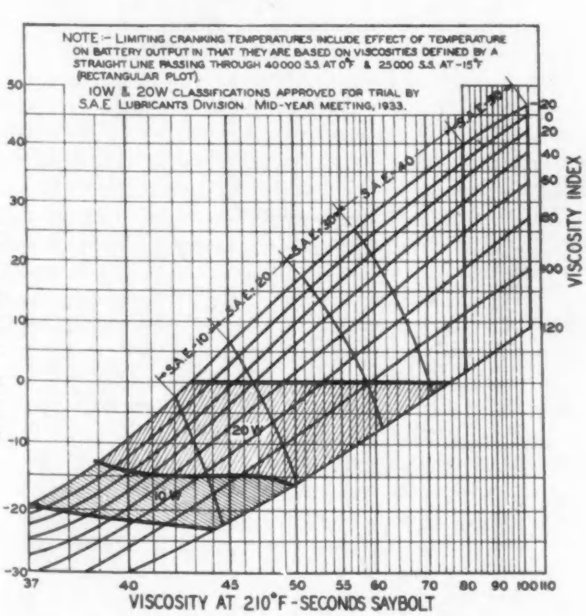
$$\text{Viscosity Index} = \frac{L - U}{L - H} \times 100$$

where U = Saybolt viscosity at 100 F of the test oil.

It will be noted from Fig. 3 that the viscosity index is, in its simplest terms, the percentage by which an oil deviates from the 0 V.I. toward the 100 V.I. oil. An oil having a steeper viscosity-temperature slope than the 0 V.I. oil would have a negative viscosity index, whereas an oil having a lesser variation in viscosity with temperature than the 100 V.I. oil would have a viscosity index above 100.

In the Davis, Lapeyrouse, and Dean viscosity-index system just described, the H or 100 V.I. oils were representative of Pennsylvania oils, and L or 0 V.I. oils were similar to the acid-treated Texas or California oils produced at that time. Thus, it was possible when the system was first devised to identify the source of oils with reasonable accuracy by means of their viscosity indices. However, modern treating methods have enabled the refiner to improve viscosity indices markedly, and oils from 30 to 50 V.I. are now prepared from crude that would produce only 0 to 10 V.I. oils by the older methods. Similarly, Mid-Continent oils have been increased from 75 to 100 V.I., and solvent-treated Pennsylvania oils of 110 V.I. are not uncommon. Besides solvent treating, another means of improving viscosity index has been developed, which consists of adding small amounts of extremely viscous polymers to SAE 10 and 20 grade oils. These polymers have relatively greater effects on the high-temperature viscosities than on those at lower temperature, and it is possible by their use to increase greatly the viscosity index of any type of lubricating oil.

[1] See the *Oil and Gas Journal*, Vol. 30, No. 46, March 31, 1932: "Applying Viscosity Index to Solution of Lubricating Problems," by G. H. B. Davis, M. Lapeyrouse, and E. W. Dean.



■ Fig. 5 – Motor lubricating oils – Limiting cranking temperatures

Considering these newer types of oils and the great number of combinations and blends that are possible, it is obvious that viscosity index can no longer be used to identify the source of an oil. It is still valuable, however, for expressing the relative change in viscosity with temperature.

S.A.E. VISCOSITY NUMBER	VISCOSITY RANGE SAYBOLT UNIVERSAL	CONSISTENCY. MUST NOT CHANNEL IN SERVICE AT °F
80	100,000 SEC AT 0°F - MAX.	-20
90	800 TO 1500 SEC AT 100°F.	0
140	120 TO 200 SEC. AT 210°F	+35
250	200 SEC AT 210°F MIN.	--

■ Fig. 6 – Transmission and rear-axle lubricants – SAE viscosity numbers

The SAE viscosity-number system for crankcase oils is shown in Fig. 4. These viscosity numbers express ranges of viscosities only and are not related in any manner to the purity or service value of the lubricants.

In order to show the service significance of the viscosity characteristics of oils of different SAE grades, they have been plotted in Fig. 5 against the temperature at which the average car can be started with these oils in the crankcase, provided, of course, that the pour point of the oil is lower than the crankcase temperature. This so-called "limiting cranking temperature" was determined from the known factors; namely, that a viscosity of 40,000 S.S.U. is a practical maximum at 0 F, whereas a viscosity of less than 25,000 S.S.U. is required at a temperature of −15 F. This lower allowable viscosity at −15 F is the result of lower battery output and poorer gasoline volatilization. The 10-W and 20-W classifications were never formally adopted by the SAE, but are of definite interest because many lubricants are specified according to these limits. It should be emphasized that the chart represents conditions where both the engine and the battery have reached an equilibrium temperature with the atmosphere, and that the values shown on the chart may deviate considerably from any individual automobile.

Fig. 5 illustrates a convenient method of determining the starting characteristics of an oil, but tends to overemphasize the importance of viscosity index. For example, 10 points in V.I., or the difference between 90 and a 100 V.I. oil, amounts to only 3 F or, expressed differently, about 1 to 2 sec in viscosity at 210 F.

Although not strictly within the subject of this paper, it should be mentioned that gear oils are also designated by means of an SAE Viscosity Number System. The limits of the Gear Oil Numbers taken from the SAE Handbook are shown in Fig. 6. It will be noted that the crankcase oils have numbers of 10 to 70, and the gear oil grades are numbered 80, 90, 140 and 250. It might be assumed from this sequence that all gear oils are more viscous than all motor oils, but such is not the case, and there is a wide

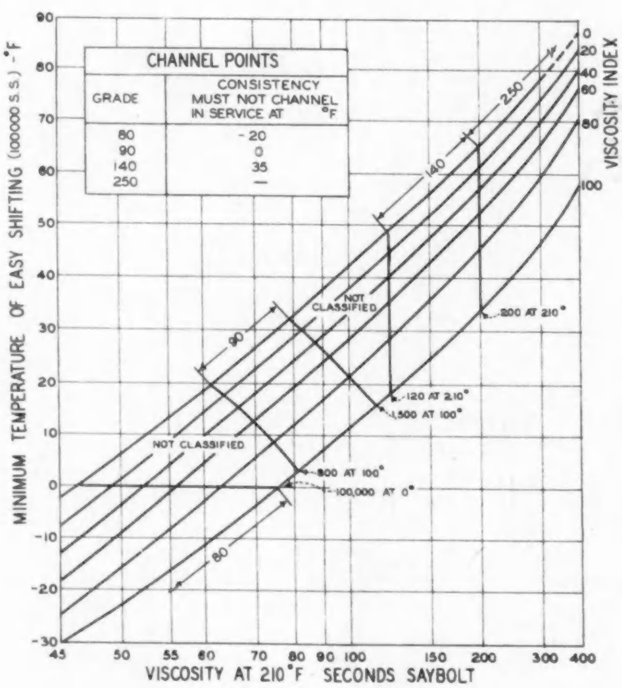
overlapping of the two systems. For conventional mineral oils, the overlapping may be explained as follows:

Gear-Oil Viscosity No.	Crankcase-Oil Viscosity No. of Same Oil
80	30
90	40 and 50
140	70
250	Heavier than any motor oil classification.

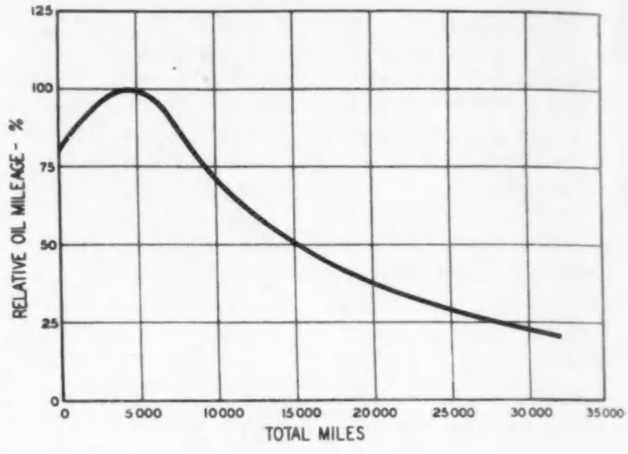
The various grades of gear oils have been plotted on Fig. 7 against a service criterion, the lowest temperature at which easy shifting can be accomplished. This chart illustrates the three types of service for which the different grades of gear oil are designed. The SAE 80 grade is primarily a low-temperature lubricant; consequently, with this oil, the most important viscosities are the ones at low temperatures so the viscosity limit is specified at 0 F. The SAE 140 and 250 grades are designed for heavy-duty and low-speed service conditions, and their limits are based on viscosities at 210 F, with no regard for their low-temperature viscosity characteristics. The SAE 90 grade is an intermediate lubricant requiring some characteristics of each of the foregoing two types, and is specified on viscosities obtained at the intermediate temperature of 100 F. As in the case of the cranking chart, the limiting shifting temperature criterion on Fig. 7 was based on average conditions.

■ Oil Consumption

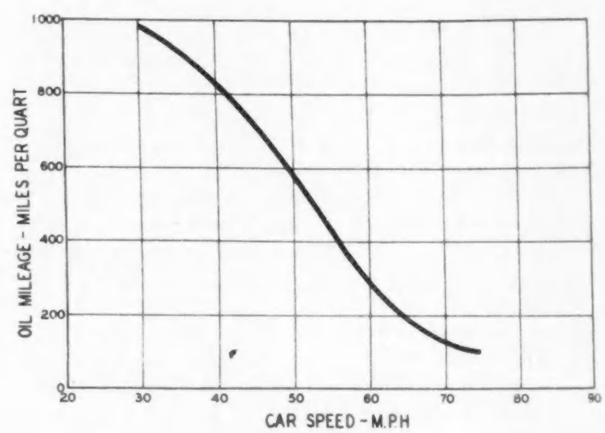
While the viscosity characteristics of lubricating oils are measured easily and can be expressed accurately by formulas, oil consumption is a more complex problem because of variations in engines and operating conditions. Fig. 8



■ Fig. 7 – SAE viscosity numbers for gear oils – Limiting shifting temperatures



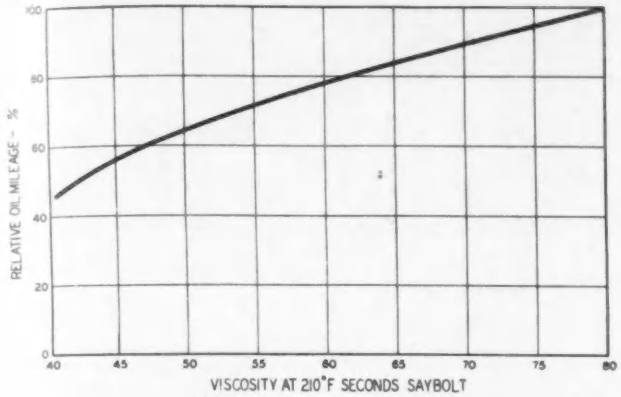
■ Fig. 8 – Relationship between engine age and oil mileage



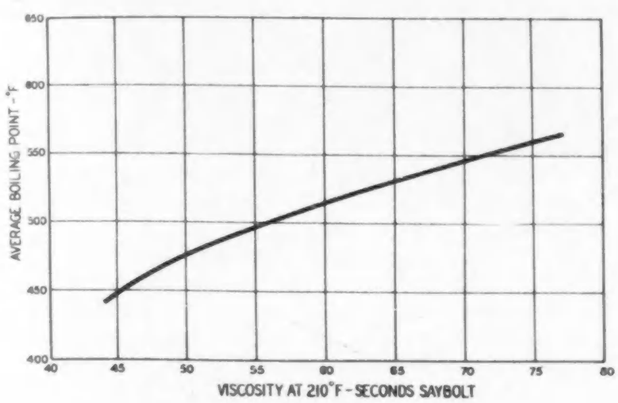
■ Fig. 9 – Relationship between car speed and oil mileage

shows the relationship between engine age and oil mileage. It will be noted that with a new car, oil consumption tends to improve during the first few thousand miles of use and, after reaching a maximum, there is a gradual decline. At 30,000 miles of use about five times as much make-up oil is necessary as is required when the car is at its best oil mileage. This chart represents an average condition, and since, as will be shown later, engine wear may vary over wide limits, the rate at which oil consumption increases will vary over similarly wide limits. The trend shown is purely relative and does not indicate that all cars become "oil hogs" in a period of 30,000 miles. Thus, for example, one car may consume less than 1 pt of oil between drains when new, and 1 to 2 qt per 1000 miles when two to three years old; a second car may consume 1 qt per 1000 miles when new, and 5 to 6 qt when two or three years old.

Another variable having a very important influence on oil consumption is car speed. A typical plot of the influence of speed on oil consumption is shown in Fig. 9. In this case almost four times as much oil was consumed at 60 mph as at 30 mph. The rate at which oil is consumed is influenced also by the viscosity grade. The influence of oil grade or oil viscosity is shown on Fig. 10, where it will be noted that oil mileage becomes greater as the viscosity is increased. The reason for the decrease in consumption with increase in viscosity is based on two oil properties – viscosity and volatility – which are related as shown by



■ Fig. 10 – Relationship between oil mileage and viscosity



■ Fig. 11 – Relationship between viscosity and volatility for a group of 90 V.I. oils

Fig. 11. The boiling points shown on this chart were obtained at a high vacuum and should be considered on a relative basis only.

The more viscous oils flow more slowly to the top of the cylinders where they are consumed, and the less volatile oils are consumed more slowly because of their greater resistance to vaporization. In different engines the relative effects of volatility and viscosity vary but, in all cases, improving either of these properties improves oil mileage.

In spite of the apparent advantage in oil consumption for the heavier grades, it should be emphasized that some consumption is necessary if the top rings are to be lubricated properly. In addition, the lighter oils, because of better circulation and lower absolute viscosities, tend to establish early lubrication and prevent excessive wear, and also decrease engine temperatures and increase fuel mileages. The selection of the viscosity grade to provide optimum overall performance, where starting does not require even a lighter grade, should therefore be the lightest grade that will give a reasonably good oil mileage; for example, 500 miles per qt.

■ Engine Wear

It is well known that gasoline and diesel engines wear out, but just how they wear has been the subject of considerable study and research. These investigations have brought out the fact that engine wear is always maximum at the top of the top-piston-ring travel. This may be due to several factors, including:

1. Ring flutter.
2. The breaking of fluid films as the piston goes through zero velocity.
3. The characteristic of wearing surfaces on which some evidence has been obtained that the wear rate is maximum where rubbing speeds are lowest.

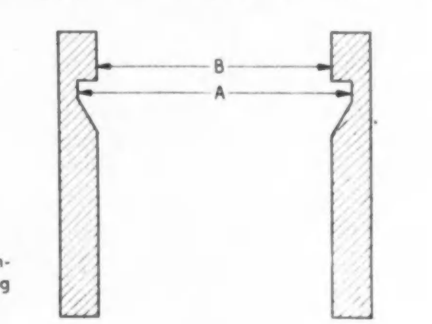
In order to investigate engine wear effectively, a standardized system of obtaining reliable wear measurements is necessary. A method found very useful for this purpose is illustrated in Fig. 12, the wear at any time being the difference between *A* and *B*, both of which are obtained with micrometers. If *A* and *B* are not exactly the same at the beginning of a test, it is, of course, necessary to correct the final value in order to obtain the wear rate during the test period. This method of establishing a base line eliminates such important sources of error as:

1. Cylinder warpage.
2. Micrometer adjustments.
3. Micrometer or cylinder temperature.
4. The "feel" of the man making the measurements.

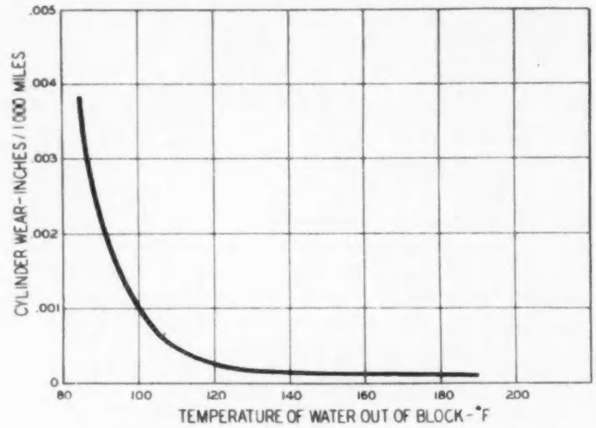
In our laboratories, we use micrometers equipped with verniers which read to the nearest 0.0001 in. Wear measurements obtained by this means with different micrometers never vary by more than 0.0002 in.

In cylinder wear the most important engine variable is the cylinder-wall temperature. This temperature is not easily controlled, but is related to the temperature of the water from the block and, by varying this water temperature, extremely large variations in wear rates are obtained. A plot showing the effect of block temperature is included as Fig. 13. These results show that to obtain long engine life the water thermostats should always be set to maintain the block temperature above 140 F. Each time an engine is started there is a short interval during which the engine block is being raised from ambient to equilibrium temperature and, during this time, the destructive effects of low-temperature operation exist. For this reason, engines operated intermittently almost invariably show a very high wear rate for a given mileage.

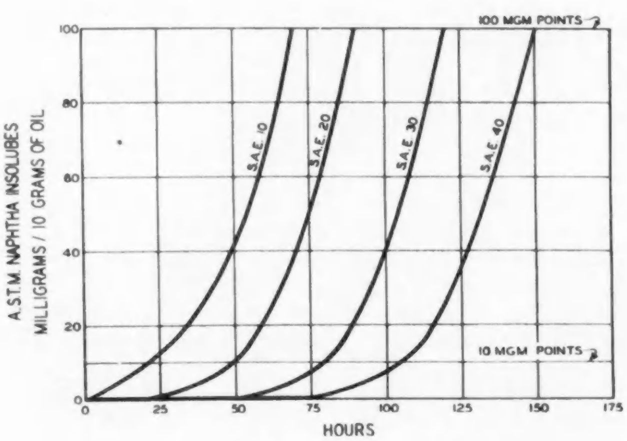
The cause of the extremely high wear rate at low operating temperatures is corrosion of the cylinder walls and piston rings by acidic combustion products. For each pound of fuel burned, there is formed by combustion over one pound of water and about three pounds of carbon dioxide. When the temperature of the cylinder wall is



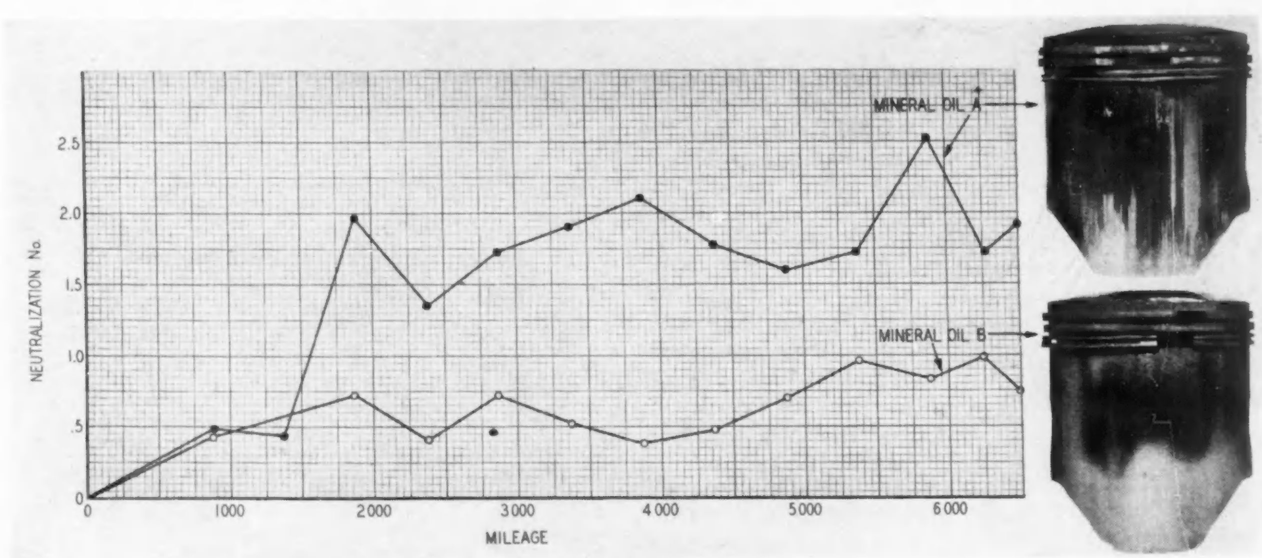
■ Fig. 12 – Method of measuring engine wear



■ Fig. 13 – Engine wear – Effect of temperature – Laboratory tests in 1938 passenger-car engine



■ Fig. 14 – Indiana oxidation test results – Effect of grade



■ Fig. 15 – Stability to oxidation – Development of acidity and varnish – 1938 passenger-car engine

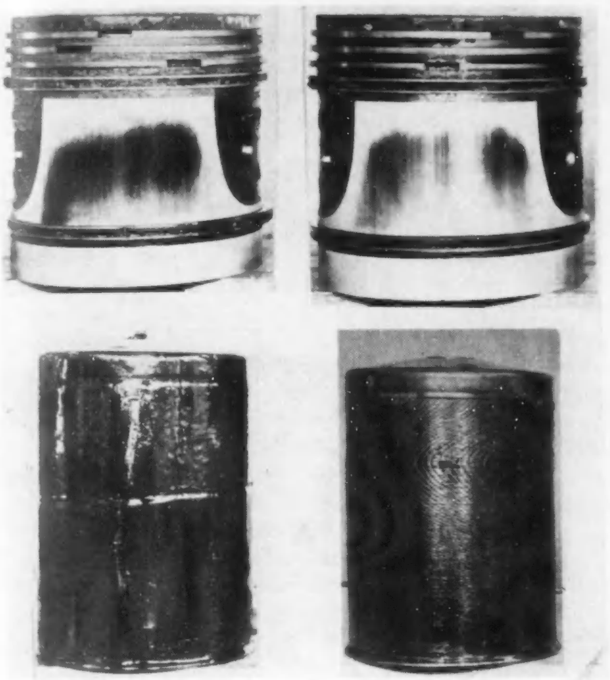
below the dew point of the exhaust gas (about 140 F) the water vapor condenses and absorbs the carbon dioxide, forming an acidic solution which readily attacks the iron cylinder wall. The corrosion product is chiefly iron oxide or rust which, under the name of rouge, is a common grinding compound. Low-temperature operation, therefore, accelerates wear in two ways: first, by causing corrosion of the cylinder and piston rings, and secondly, by the corrosion product causing abrasive wear. A very complete investigation of engine wear was recently summarized in a book by C. G. Williams.[2]

■ Oil Stability

All oils used in internal-combustion engines are exposed to high temperatures and intimate contact with oxygen. Under these conditions, the oil oxidizes or combines chemically with oxygen which, if carried far enough, results in the formation of acidic and resinous materials. The acidic products may be corrosive to some engine parts, while the gummy products are the cause of ring-sticking, piston varnish, and sludge formation. Stability toward oxidation is therefore a very important property of a lubricating oil. With respect to operating temperatures, the best compromise to avoid both excessive oxidation and the high wear encountered at low jacket temperature, is in the range between 160 to 180 F, which may be maintained by means of thermostats or radiator shutters.

One of the most popular laboratory oxidation tests is the Indiana Oxidation Test. In this test, the oils are agitated with a metered amount of air in glass tubes at a temperature of 340 F, and the amount of sludge or naphtha-insoluble material is determined at frequent intervals. Results from Indiana Oxidation Tests with a series of oils are shown in Fig. 14. It will be noted that the SAE 10 grade was the poorest oil tested and that, as the SAE number was increased, better results were obtained. This was not due entirely to greater inherent stability of the SAE 40 oil since the oxidation products are more soluble

[2] See "Collected Researches on Cylinder Wear," by C. G. Williams of the Institution of Automobile Engineers.



MINERAL OIL C MINERAL OIL D

■ Fig. 16 – Effect of type of oil on engine deposits – Pistons and filters after 65,000 miles of operation

in the heavier oils than in the lighter oils. Also, this apparent advantage is offset in engine tests, because of higher operating temperatures and lower make-up rates with the heavier oils. Furthermore, the Indiana Oxidation Test is made in glassware, and results with equivalent grades are often obtained that do not correlate with service. This inconsistency is particularly noticeable with certain highly treated oils and may be attributed to the fact that no metallic catalysts are present in the Indiana apparatus.

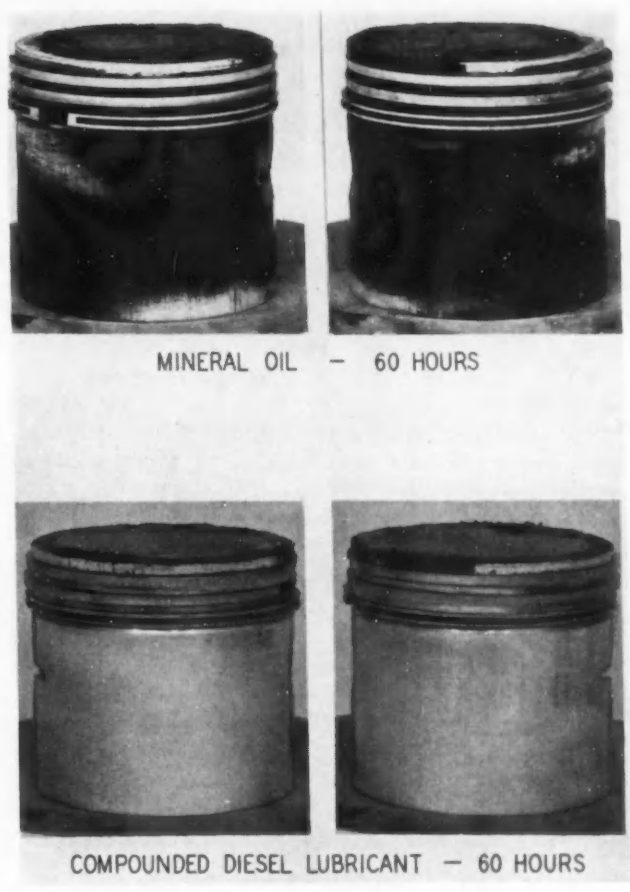
Marked differences in the oxidation stability of lubricants exist in engine service, as are illustrated on Figs. 15 and 16. Fig. 15 shows the development of acidity in two mineral oils tested in a 1938 passenger-car engine. The acidity in each case should be considered only as an index of the amount of oxygen absorbed, the true criterion of the oil performance being the condition of the pistons at the end of the test. The oil that absorbed a relatively large amount of oxygen caused a heavy layer of gum to be formed on the piston skirts, and it may be noted from the vertical scuffing marks that engine failure was imminent. The other piston had a much lighter deposit on the skirt, and obviously could have been used for a much longer period of time with no difficulty.

The tests illustrated in Fig. 16 were made under relatively mild operating conditions, and less difference between the two oils was noted. Mineral Oil *C* formed a greater amount of sludge and carbonaceous material than Mineral Oil *D*, however, and the differences were most apparent in the amount of clogging of the upper oil ring and the sludge deposits on the oil filters. The purpose of Figs. 15 and 16 is to illustrate the fact that differences exist in the service characteristics of lubricating oils and that the ordinary inspections do not predict these differences. In the case of oils *A* and *C*, the viscosity indices were 10 to 15 points higher than those of oils *B* and *D;* yet the latter oils gave better engine results.

■ Compounded Oils

Lubricating oils prepared from the most suitable crudes by the most advanced refining methods are not always adequate for such special services as use in high-speed diesel and aircraft engines, and the need for additives to obtain further improvements in this field is recognized by the major engine and petroleum manufacturers. Compounded lubricants are not new, but have been used for many years – for example, steam cylinder oils, extreme-pressure lubricants, and all types of greases. The addition of chemical compounds to lubricating oils may be considered analogous to the manufacture of alloy steels with the compounding material like the nickel, chromium, or molybdenum, adding special properties not possessed by the original material.

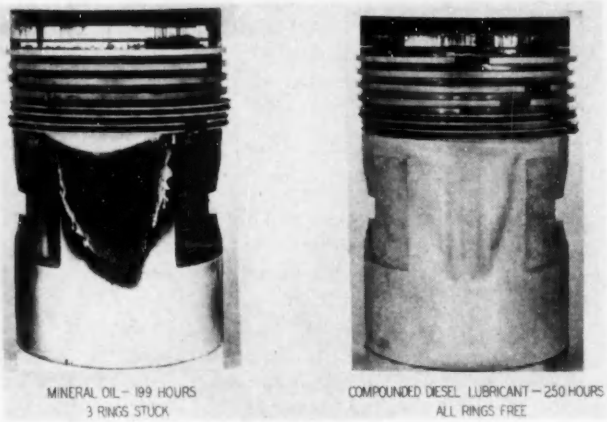
The engine having the greatest need for compounded lubricating oils is the high-speed diesel engine, as few of these engines can be operated satisfactorily with uncompounded mineral oils. The compounds used to provide satisfactory diesel-engine lubricants have the ability to prevent ring-sticking, gum deposits, sludge deposits, and excessive engine wear. In a similar manner, the use of compounding materials causes a marked improvement in breaking-in oils. The use of appropriate chemicals for this purpose reduces the amount of slow-speed, light-load oper-



■ Fig. 17 – Effect of compounding material on piston deposits – Single-cylinder laboratory engine

ation necessary with uncompounded mineral oils, and promotes a better surface condition which improves subsequent engine operation.

Examples of the beneficial effects of compounding materials in lubricating oils are shown in Figs. 17 to 19 inclusive. Fig. 17 shows the piston from a one-cylinder laboratory engine. The piston used with the straight mineral oil was heavily coated with a dark brown film of gummy material, and the oil ring slot was almost completely clogged with carbonaceous material. The use of the compounded oil eliminated almost all of the deposits on the piston skirt and in the oil ring groove. Attention is also called to the marked difference of the ring belts of the two pistons.



■ **Fig. 18 – Effect of compounding material on ring-sticking and piston deposits – Tractor diesel engine**

Fig. 18 shows diesel-engine pistons that were used under moderate-load conditions. In these tests the engine was operated at its rated load for continuous service, and it may be noted that the uncompounded oil formed some piston deposits which, progressively, would have resulted in unsatisfactory operation, while the piston used with the compounded oil was in excellent condition.

The compounded oil used in this test has been used for several thousand hours in many similar engines with pistons almost as clean as the one used for the illustration.

Fig. 19 illustrates the effect of compounding material in obtaining improved operation in a truck diesel engine. These tests were somewhat accelerated, but represent almost exactly the conditions found after about 40,000 to 50,000 miles of operation in normal trucking service.

■ **Fig. 19 – Effect of compounding material on ring-sticking and piston deposits – Truck diesel engine**

All of the oils, both straight and compounded, shown on Figs. 17 to 19, inclusive, are commercial lubricants and are available to the users. These charts illustrate the beneficial effects of compounding, but there are several peculiarities of compounded oils that should be brought to the attention of the user.

First, compounded oils cannot be reclaimed successfully. This is due to the fact that, in reducing gummy deposits, the compounding materials are consumed and, since the reclaiming operation cannot supply additional compounding, the reclaimed oils are always inferior to the fresh oils. Secondly, compounded oils tend to form a more stable foam than ordinary mineral oils, and the presence of the foam is more noticeable because of its persistence, although the actual amount of foam may be only slightly greater than with an uncompounded oil.

Compounded oils can be evaluated satisfactorily in laboratory engines or in service tests, but their performance cannot be predicted from their inspections. In this regard, it should be emphasized that the addition of small percentages of compounding material to an oil has a large influence on its service performance; yet it causes almost no change in the normal inspections, and these values are therefore meaningless with respect to the performance characteristics of the lubricant.

In summary, some of the performance characteristics of lubricating oils, such as limiting cranking temperatures, oil mileages, and gear-shifting temperatures, can be predicted from viscosity determinations, but the other items of interest to the users, such as stability, ring-sticking, gumming and wear, cannot be predicted from inspections. The trend of engine design, in both the gasoline and diesel fields, has resulted in an increase in the mechanical and thermal loads imposed on the lubricating oil and, in some services, such as diesels and aviation engines, satisfactory operation can be obtained only by the use of compounded oils. Such oils cannot be judged by simple laboratory tests and should therefore be selected on a basis of their performance in full-scale engines.

Synthetic Rubber in Automobiles

(*Discussion concluded from page 97*)

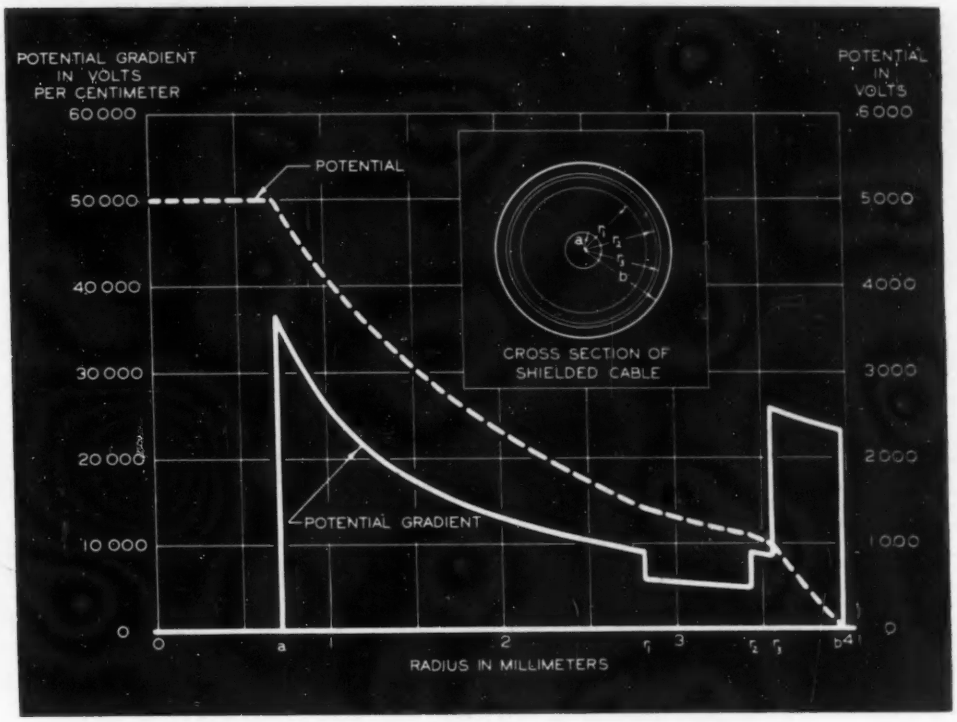
a specific gravity of 1.06 and, when it is stored in the dark, it does not undergo any change but, in the light, it has a slight discoloration.

Now as to its manipulation. It works best on a cold mill and preferably one that is opened to a maximum of 3/16 in. The best plasticizer is pine-tar oil which is compatible in very high proportions. Some form of carbon black is always desirable and may follow the plasticizer on the mill. Stearic acid is difficult to mill into Chemigum. The best way is to put it in the natural rubber plasticizer. If a hot mill is used for breakdown and mixing, the rubber takes on the appearance of scorched stock and becomes rough and grainy but it smooths out again immediately if put on a cool mill. The relative mill capacity for a 40% black master batch is 65% that of natural rubber. However, it can be mixed very rapidly in a Banbury mixer although, for best results, it is desirable afterwards to sheet it out through a tight mill.

Natural rubber can be blended with Chemigum in almost any proportions, but this is done most readily by combining the black master batches. A 50-50 mixture with natural rubber handles in most respect like a 100% natural rubber stock. Chemigum tends to be soft and plastic when warm. In fact it resembles chewing gum more than crude rubber in that state, but when it is cold it has a tendency to be dead and rather non-adhesive. It does not exhibit the adhesive strength and "tooth" of natural crude rubber. Fifteen per cent of crude rubber can be used to produce a very good surface tackiness and has no serious effect on such properties as oil resistance. The various tack-producers commonly used for natural rubber can be used with some beneficial effect with this material.

SUPERCHARGED Aircraft Ignition Harnesses

by CARL E. SWANSON
Electrical Engineer, Northwest Airlines, Inc.



■ Fig. 1 – Plots of potentials and potential gradients existing at points within 5/16-in. I.D. metallic shielding concentrically positioned around conventional 7-mm rubber - neoprene - lacquer covered copper conductor ignition cable. For this plot the potential difference between the conductor and the shield was chosen arbitrarily as 5000 v

IT is a fact well known to operators of commercial and military planes that the problem of establishing and maintaining proper ignition performance under all conditions of weather and service constitutes one of the most difficult and costly problems associated with airplane operation. It has been estimated in the past that at least one-half of the flight interruptions chargeable to airplane powerplants can be traced either directly or indirectly to improper ignition performance. One large airline has reported that the cost of its ignition maintenance is approximately equal to the cost of its oil.

It is the opinion of most operators that of the three main elements of the ignition system, namely, the magneto, the harness, and the spark plugs, the harness has been by all

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Nov. 1, 1940.]

SUPERCHARGED aircraft ignition harnesses have recently been introduced into commercial aviation as a means of preventing the entrance, formation, and accumulation of impurities within the shielded distribution systems universally employed on airplanes using high-sensitivity radio receivers.

This paper presents a discussion and description of the operation and performance of the supercharged harness.

Introductory to this material, and in the interest of a better understanding thereof, there is presented a general review of the fundamental character of the troubles experienced with ignition distribution. Included in this is a description of the chemical nature of the problem and a mathematical analysis of certain electrical stresses within the system which are related to the formation of corona.

A summary is made of various other designs of harnesses which have been set forth in an effort to eliminate the ignition distribution problem and it is pointed out wherein each has its weaknesses.

In support of the contention that the supercharged harness makes possible complete relief from the troubles associated with harness contamination a perfect service record of 5,000,000 engine miles is cited.

odds the chief offender. Failure of the harness to provide proper ignition distribution has necessitated not only frequent and costly service of this unit but also has necessitated frequent and costly spark-plug removals. These considerations in combination with many others, not the least of which is the fact that an improperly functioning ignition system is always a potential threat to safety, have established this problem well forward in rank of importance.

The problem of securing satisfactory transmission of electrical energy from the magneto, or other source of electrical impulses, to the spark plugs is much more difficult on aircraft engines than it is on other types of internal-combustion engines. Two of the most important considerations which serve to make the problem more difficult are: first, that aircraft ignition systems must be completely shielded electrically by metallic conduit if high-sensitivity, high-frequency radio receivers are used; and second, that aircraft ignition systems must operate throughout a wide range of atmospheric pressures. Most of the ignition distribution troubles peculiar to aircraft installations can be traced to either one, or the other, or a combination of both, of these points of difference.

As a consequence of the fact that shielded distribution must be employed, there are formed dead air spaces within the system. These dead air spaces become contaminated with undesirable and injurious gases, acids, fluids, and oils. These impurities originate from three sources and can best be studied by considering them separately.

First, there are those undesirable substances which enter the system through leaks or openings in the shielding. Rain, sleet, anti-icer fluid, and oils entering through these leaks or openings are a constant source of trouble, especially when present within the spark-plug shield chambers. These fluids, when present within the ignition harness manifold, are not particularly harmful but, when present within the shield chambers, cause electrical spark-over therein with attendant failure of the plug to operate.

Second, there are those impurities which originate from electrical and chemical phenomena which take place within the shielding. Reference is here made to conditions and effects associated with the presence of electrical corona within the system. When air is present in regions subject to electrical potential gradients in excess of certain critical values, which in turn are dependent on air density, a phenomenon known as "corona" occurs. If the potential of an electrical conductor be increased sufficiently a faint glow will appear at the surface of the conductor and a hissing sound will be heard. The phenomenon, of which the glow and sound are indications, is called corona.

J. J. Thomson and R. Threlfall[1] clearly showed that ozone formation in a silent discharge tube was associated with the production of the luminous glow. Ozone (O_3) attacks rubber very rapidly and is injurious to most of the common metallic surfaces, particularly when moisture is present.[2]

Also associated with the presence of corona is the formation of nitric oxide (NO). For some time a subject of controversy this fact was finally established by Berthelot[3] and by Warburg and Leithäuser[4] who showed definitely that NO was produced by the action of the silent discharge on air.

In the presence of excess atmospheric oxygen the NO is further oxidized to form nitric dioxide (NO_2).

$$2NO + O_2 \rightleftarrows 2NO_2 \qquad (1)$$

Richardson[5] showed that this action is reversible, the NO_2 beginning to dissociate into nitric oxide and oxygen above 140 C. At ordinary pressures this dissociation is not complete until 620 C, being only 13% complete at 279 C.

The action of NO_2 on water was investigated thoroughly by Foerster and Koch[6]. The NO_2 is absorbed by the water and a mixture of nitric acid (HNO_3) and nitrous acid (HNO_2) is formed.

$$2NO_2 + H_2O = HNO_3 + HNO_2 \qquad (2)$$

Under ordinary conditions free nitrous acid is unstable and decomposes as follows:

$$3HNO_2 = HNO_3 + H_2O + 2NO \qquad (3)$$

The liberated NO combines again with atmospheric oxygen as described in (1) and the action goes on until practically all of the NO_2 goes into nitric acid. This acid has a deteriorating effect on the electrical insulation and the metal shielding of the ignition system, but a more injurious effect for consideration is the fact that these acids make any films of moisture that might exist in the system very good conductors. In particular, if such acid films exist within and on the parts within the spark-plug shield chamber, electrical spark-over and so-called carbonization of the spark-over path is almost certain to result. When this happens, the common practice of running the engine at high speed to heat and thereby to dry out the spark-plug shield chamber will not restore proper operation to the system, the "carbonized" path remaining a more or less permanent short circuit.

Third, and lastly, there are those impurities which enter the ignition system from the combustion chambers through spark-plug core leakage. That such leakage is appreciable in quantity is evidenced by the fact that certain spark-plug manufacturers specify that leakages as high as 0.5 cc per min per plug at 150 lb per sq in. pressure shall not be cause for rejection. Water vapor is of course present in large quantities in the exhaust gases. There are also present appreciable quantities of hydrobromic acid and sulfuric acid in the condensate of exhaust gases. These substances, being injected directly into the spark-plug shield chambers, obviously jeopardize the functioning of the ignition system by causing deterioration and lowering of the electrical insulation therein.

As a consequence of the fact that aircraft ignition distribution systems are subject to wide variations of atmospheric pressures, there are still other considerations and phenomena which add to the troubles just described.

As before mentioned, airplane engines and their associated ignition systems are more or less constantly covered with films of various kinds of fluids and oils. Some of these are sprayed on the engines during periods of cleaning and inspection. Others accumulate on the engines during periods of flight. In modern airplanes an anti-icer fluid consisting of a mixture of alcohol and glycerine is applied through nozzles or jets to the blades of the propellers while the airplane is flying in weather conducive to the formation of ice. These fluids, together with water encountered when flying through rain and sleet, spray over the entire engine

[1] See Proceedings of the Royal Society, Vol. 40, 1886, pp. 340-342: "Some Experiments on the Production of Ozone," by J. J. Thomson and R. Threlfall.

[2] See p. 12: "Ozone – Its Manufacture, Properties, and Uses," by A. Vosmaer, D. Van Nostrand Co., Inc., 1916.

[3] See *Comptes Rendus*, Vol. 142, 1906, p. 1367; see also *Annales de Chimie et de Physique*, 1906; both by M. Berthelot.

[4] See *Annalen der Physik*, Vol. 20, 1906, pp. 743-750, also Vol. 23, 1907, pp. 209-225: "Über die Oxydation des Stickstoffs bei der Wirkung der Stillen Entladung auf die Atmosphärische Luft," by E. Warburg and G. Leithäuser.

[5] See Transactions of the Chemical Society, Vol. 51, 1887, p. 397: "Action of Heat on Nitrogen Peroxide," by A. Richardson.

[6] See *Zeitschrift fur Angewandte Chemie*, Vol. 21, 1908, p. 2161 and 2209, by Foerster and Koch.

and ignition harness with great force from the propeller backwash. Rapid descent after a period of high-altitude flying supplies an additional force which drives these fluids into the ignition system. On descending to lower altitudes there is a time interval during which the pressure outside the harness is greater than the pressure within. The result is that these fluids are actually forced through any leaks that might exist anywhere in the shielding.

Then too, as before stated, the advent and extent of the corona within the system is dependent on the density of the air within. As the density of the air is diminished, the minimum potential gradient required to produce corona is diminished. Air density diminishes with an increase in altitude, therefore, the voltage required to produce corona also diminishes with an increase in altitude. Ignition systems are therefore subject to a condition of intensified corona activity at high altitude.

In view of the fact that the presence and the extent of the corona within the shielding are related definitely to the existence and the intensity of the harmful chemical actions therein, it is in order to devote some attention to the technical aspects of the corona problem. In doing so, emphasis will be placed on the study of conditions existing within the spark-plug shield chambers and within the shielding conduit covering the ignition cables from the plugs to the harness manifold. It is within the shield chambers and these closely adjacent regions that the presence of ozone and acids is most likely to cause failure of insulation or spark-over, and it is true, in general, that these regions are subject to some of the highest electrical stresses found in the distribution system. Then too, since there is no definite arrangement of the cables within the manifold and the large conduits connecting to the magnetos, it is quite impossible to construct any analytical representations of conditions therein.

Fortunately, it is a simple problem in electrostatics to compute the potentials and the potential gradients existing at all points within a cylindrical metallic shield surrounding an insulated conductor if it is assumed that the conductor is positioned concentrically within the shield. By solving this problem it is possible to predict approximately the voltages at which corona will make its appearance and to determine the effect which the various circuit dimensions and constants have upon the presence and extent of this phenomenon.

Let:

a = radius of the conductor (assumed to be circular in cross-section).

r_1 = radius of the outer surface of the first insulation layer.

r_2 = radius of the outer surface of the second insulation layer, etc., etc.

r_{n-1} = radius of the outer surface of the last insulation layer.

b = radius of the inner surface of the grounded shield.

x_i = distance from the center of the conductor to an interior point of the i th insulation layer.

$$r_{i-1} < x_i < r_i \qquad i = 1, 2, 3, \ldots\ldots\ldots\ldots n$$

E_i = electric intensity at a point distant x_i from the center of the conductor.

V_i = potential at a point distant x_i from the center of the conductor.

k_i = specific inductive capacity of the material comprising the i th insulation layer.

λ = e.s.u. charge per unit length of the conductor. (Chosen positive)

The electric intensity E at all points of a cylindrical surface concentric with the conductor may be computed by the use of Gauss' Law[7]

$$\int_S E \cos\alpha \, dS = \frac{4\pi}{k} \Sigma q \tag{4}$$

where S is a surface enclosing the free charges Σq, and α is the angle between the lines of force and the outward drawn normal to the surface.

In this particular case the lines of force are everywhere radial so for a unit length equation (4) yields

$$2\pi x_i E_i = \frac{4\pi\lambda}{k_i}$$
$$E_i = \frac{2\lambda}{k_i x_i} \tag{5}$$

The potential at any point in the system is obtained by computing the work required to move a unit positive charge from the inner surface of the shield to that point. It is thus determined that V_0, the potential of the conductor, may be expressed as

$$V_o = \lim_{\epsilon \to o} \left[\int_{a+\epsilon}^{r_1-\epsilon} E_1 dx_1 + \int_{r_1+\epsilon}^{r_2-\epsilon} E_2 dx_2 + \cdots \int_{r_{n-1}+\epsilon}^{b-\epsilon} E_n dx_n \right] \tag{6}$$

where $0 < \epsilon$.

The potential gradients $\frac{dV_i}{dx_i}$ are given by the relationship

$$\frac{dV_i}{dx_i} = - E_i \tag{7}$$

Substituting Equation (5) in (6) and (7) and eliminating λ between the resulting equations it is determined that

$$\left|\frac{dV_i}{dx_i}\right| = \frac{1}{k_i x_i} \frac{V_o}{\left[\frac{1}{k_1}\log_e\left(\frac{r_1}{a}\right) + \frac{1}{k_2}\log_e\left(\frac{r_2}{r_1}\right) + \ldots \frac{1}{k_n}\log_e\left(\frac{b}{r_{n-1}}\right)\right]} \tag{8}$$

A graph of this equation applied to a typical section of shielded aircraft cable appears in Fig. 1.

It has been established experimentally by Peek[8] that in the case of concentric cylindrical conductors the critical potential gradient at which visual corona appears at the surface of the inner cylinder is given by the formula

$$g_v = \left|\frac{dV}{dx}\right| = 31{,}000\,\delta\left(1 + \frac{0.308}{\sqrt{\delta r}}\right) \text{ v per cm} \tag{9}$$

where r is the radius of the inner cylinder and δ is the ratio of the mass of a unit volume of air under the circumstances in question to the mass of a unit volume of air at 25 C and 76 cm pressure. The dependency of δ upon temperature and pressure is in turn expressed by the relationship

$$\delta = \frac{3.92b}{273_{\cdot} + t} \tag{10}$$

where

b = barometric pressure in cm

t = temperature in C

With Equations (8) and (9) it is possible to compute the minimum potential of the conductor required to pro-

[7] See p. 45: "Principles of Electricity," by Page and Adams, Fourth Printing, D. Van Nostrand Co., Inc.

[8] See pp. 61-64: "Dielectric Phenomena in High-Voltage Engineering," by F. W. Peek, Third Edition, McGraw-Hill Book Co.

duce corona in the air space between the outer surface of the last insulation layer and the inner surface of the shield.

It is a matter of interest to make this computation using the dimensions of the familiar aircraft harness employing 7-mm low-resistance cable. To do this it shall be assumed that the conductor is circular in cross-section and has a diameter of 0.056 in. which is the diameter over the stranding of the conventional cable consisting of 19 strands of No. 29 B. & S. gage wire. For purposes of computing the dielectric stress in air not immediately adjacent to the conductor itself, no correction of diameter will be made in connection with the aforementioned substitution of a single circular conductor for the multi-strand cable. It also shall be assumed that the first insulation layer is of rubber 0.083 in. thick having $k_1 = 3.5$, that the second insulation layer is of neoprene 0.024 in. thick having $k_2 = 5.6$, and that the last insulation layer is of lacquer 0.005 in. thick having $k_3 = 3.0$. These dimensions approximate very closely those employed by several manufacturers and add up to give an outside diameter for the cable of 0.280 in. Lastly, it shall be assumed that the commonly used 5/16-in. inside diameter metallic hose shall constitute the shielding. Expressed in centimeters the dimensions to be substituted in Equation (8) are: $a = 0.071$, $r_1 = 0.282$, $r_2 = 0.343$, $r_3 = 0.356$ and $b = 0.397$.

In Equation (10) a choice of $b = 76$ and $t = 15$, to correspond with standard air at sea level, makes $\delta = 1.035$. If then the assumption is made that the gradient required for the appearance of corona at the surface of the lacquer is the same as that required for the appearance of corona at the surface of a smooth conductor having the same radius, Equation (9) yields

$$g_v\ (15\ \text{C},\ 0\ \text{ft}) = 48{,}300\ \text{v per cm} \tag{11}$$

as the required stress. To produce this stress, the potential of the conductor, as given by Equation (8), is

$$V_o\ (15\ \text{C},\ 0\ \text{ft}) = 9{,}450\ \text{v} \tag{12}$$

The writer has performed numerous laboratory tests on standard aircraft cable positioned within tubular metallic shields, bell-mouthed at the ends to prevent high concentration of stress in those regions, and has found very excellent correlation between observed and computed values of potential required for the advent of corona in the air space between the insulation and the shield.

It is of interest to note that, in the foregoing computation a variation of as much as 50% in the magnitude of the air-gap clearance produces less than a 10% variation in the potential required for corona.

The possible departure from fact embodied in the hypothesis that the conductor be considered concentric with the shield will not vitiate the general conclusions drawn from the foregoing analysis. With the maximum eccentricities possible in conventional harness designs the electric lines still remain essentially radial and the electric intensities are practically the same at all points in the air space between the insulation and the shield because all points therein are practically equidistant from the conductor. As a consequence the difference of potential across the air gap is very nearly proportional to the gap width. Thus the gradient varies very little throughout the air space. In any event, any corrections made for the errors introduced by this assumption would not be large and would have the effect of lowering the minimum potentials required for the production of corona.

For standard air at 20,000 ft, $b = 35$ and $t = -25$, making $\delta = 0.554$. The corresponding critical gradient and potential are

$$g_v\ (-25\ \text{C},\ 20{,}000\ \text{ft}) = 29{,}100\ \text{v per cm} \tag{13}$$

$$V_o\ (-25\ \text{C},\ 20{,}000\ \text{ft}) = 5{,}600\ \text{v} \tag{14}$$

It must not be forgotten that Equations (11), (12), (13), and (14) apply only to those regions within the individual shields which are at or near the temperature of outside air. Within the spark-plug shield chambers and within the portions of the harness immediately adjacent to these chambers, the temperatures are very much higher. In these high-temperature regions the air densities are relatively low and consequently the critical corona gradients are less. Assuming that the temperature within the shielding adjacent to the plugs is approximately 100 C higher than the temperature of outside air, the following significant values are obtained:

$$g_v\ (115\ \text{C},\ 0\ \text{ft}) = 37{,}800\ \text{v per cm} \tag{15}$$

$$V_o\ (115\ \text{C},\ 0\ \text{ft}) = 7{,}380\ \text{v} \tag{16}$$

and

$$g_v\ (75\ \text{C},\ 20{,}000\ \text{ft}) = 22{,}500\ \text{v per cm} \tag{17}$$

$$V_o\ (75\ \text{C},\ 20{,}000\ \text{ft}) = 4{,}350\ \text{v} \tag{18}$$

Neglecting effects attributable to reflections from the unfavorably terminated ends of the distribution lines, it may be said that the peak line voltages present in the distribution system are determined by the breakdown voltages of the spark plugs. These sparking voltages are, in turn, dependent on a large number of variables, including the spacing and temperature of the spark-plug electrodes, and the pressure, temperature, and composition of the fuel gases in the combustion chambers at the time of sparking. While it is obviously quite impossible to fix any definite bounds to the range of sparking voltages existing in general service, it seems quite reasonable to presume that the maintenance of high manifold pressures at high altitudes by means of fuel superchargers has the effect of holding the sparking voltages more nearly constant than otherwise would be the case. If one were to select an approximate range of values for the peak sparking voltages in modern high-output aircraft engines, taking into account maximum gap settings, that selection would probably fall in the neighborhood of 5000 to 6000 v.

Under the foregoing circumstances it is to be concluded that, while it is unlikely that there will be any extensive corona activity between the cable and the shielding at low altitudes, corona will probably make its appearance in the high-temperature regions within the elbows adjacent the plugs during operation at high altitudes. As before mentioned, the effects of eccentricity between the cable and the shielding will support this tendency by lowering the voltage required.

For points at or near the surface of the conductor itself the electric stresses are very high. Substitution in Equation (8) of $V_0 = 5000$, $k_i = 3.5$, and $x_i = 0.071$ together with the other constants of the illustrative example, gives for the value of the stress in the rubber adjacent the conductor

$$\left|\frac{dV_1}{dx_1}\right|_{\substack{x_1 = 0.071 \\ k_1 = 3.5}} = 36{,}600\ \text{v per cm} \tag{19}$$

Had air been present in the space occupied by the rubber, the result would have been

$$\left|\frac{dV_1}{dx_1}\right|_{\substack{x_1 = 0.071 \\ k_1 = 1}} = 105{,}700\ \text{v per cm} \tag{20}$$

To compute from Equation (9) the "apparent strength" of air adjacent the conductor, correction must be made for

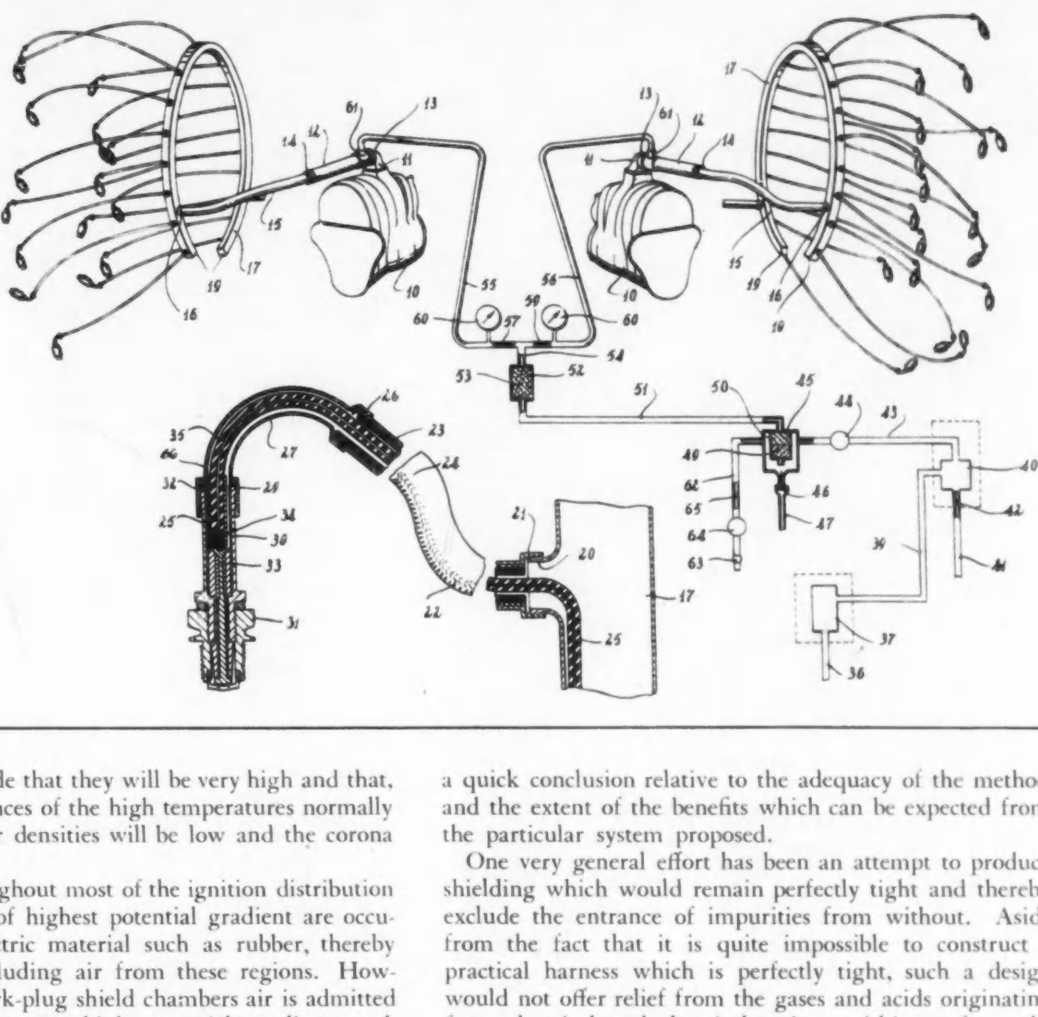
the effects of stranding[9]. When this is done, r becomes 0.061 and the gradient determined is

$$g'_r \text{ (15 C, 0 ft)} = 71{,}000 \text{ v per cm} \qquad (21)$$

From Equations (20) and (21) it follows that, even at sea level and with cold air, there would be corona present.

Now, while these circumstances and conclusions do not apply directly to conditions within the shield chamber, they do serve to show that, if air be present adjacent the conductor or to small metallic parts attached thereto, there will be extremely high stresses there. The geometry of the construction within the shield chambers is such as to eliminate any possibility of obtaining a simple solution for the stresses therein; however, the aforegoing considerations the very heart of the problem. In addition to the theoretical support heretofore presented, flying experience confirms this statement, for it is within these regions that by far the greatest number of failures occur. Materials within these regions actually disintegrate from the actions of the gases, acids, and vapors therein, and it is not uncommon to have the previously mentioned springs in these regions rust through and break.

There shall now be presented a brief description and discussion of various methods and schemes which have been employed in an effort to eliminate the troubles associated with ignition distribution. Reference to the theory and discussion heretofore set forth will be found to give



■ Fig. 2 – Circuit arrangement used in a supercharged ignition system

enable one to conclude that they will be very high and that, under the circumstances of the high temperatures normally present there, the air densities will be low and the corona particularly intense.

Fortunately, throughout most of the ignition distribution system, the regions of highest potential gradient are occupied by some dielectric material such as rubber, thereby quite effectively excluding air from these regions. However, within the spark-plug shield chambers air is admitted to regions subject to very high potential gradients and, consequently, the aforementioned electrical and chemical actions which form harmful gases and acids are present there. Small springs providing suitable resilient connections within these shield chambers are generally employed, and these springs are constructed of materials having very small radii of curvature. Air adjacent such surfaces is subject to very high potential gradients.

All the foregoing discussion is designed to acquaint the reader with the fundamental character of the ignition distribution problem and to emphasize the very important fact that the interior of the spark-plug shield chamber is a quick conclusion relative to the adequacy of the method and the extent of the benefits which can be expected from the particular system proposed.

One very general effort has been an attempt to produce shielding which would remain perfectly tight and thereby exclude the entrance of impurities from without. Aside from the fact that it is quite impossible to construct a practical harness which is perfectly tight, such a design would not offer relief from the gases and acids originating from electrical and chemical actions within or from the gases, vapors, and fumes entering through spark-plug core leakage. It is to be concluded that this method of attack reflects a most elementary and incomplete understanding of the problem.

Two other rather well-known efforts to solve the problem may be classified under one heading as they both employ the same theoretical principle. Reference is here made to designs which employ wire constructed with metallic shielding woven tightly over the dielectric surrounding the conductor, and to designs which employ the conventional loosely fitting shielding but have the space therein filled with some plastic material or with some sort of paste or compound. These two designs have in common the prin-

[9] See p. 81: "Dielectric Phenomena in High-Voltage Engineering," by F. W. Peek, Third Edition, McGraw-Hill Book Co.

ciple of eliminating or minimizing the amount of dead air space within the shielding, and consequently have some merit. Accumulation of water and fluids in the region between the wire insulation and the shielding is thus held to a minimum and, by excluding most of the air from these same spaces, there is greatly diminished the possibility of any large amount of corona activity therein. Unfortunately, however, neither of these two designs provides for the elimination of the troubles existing in the spark-plug shield chambers which are, as before pointed out, the most important regions to consider. The two methods have the common fault of failing to attack the heart of the problem. In fact, in both of these types of harnesses, the spark-plug shield chambers are shut off and isolated from all other regions. The result is that the impurities coming from the corona action within the shield chamber, and from the gases backing up through the spark-plug core, are trapped in the shield chamber, thus producing a very high concentration of contamination therein. Efforts to exclude air from the shield chambers by packing them with grease or compound have proved very unsatisfactory. The substance packed within the chamber tends to become charged with the condensate of exhaust gases entering through core leakage and, under the influence of heat, the grease or compound may harden and bake onto the inner surface of the mica cigarette in such a manner as to make it quite impossible to clean the interior of the plug at overhaul periods without spoiling it. In addition to all this, the "filled harness" has certain other undesirable features relating to the maintenance, shipping, and service of these units which are too well known to warrant discussion here.

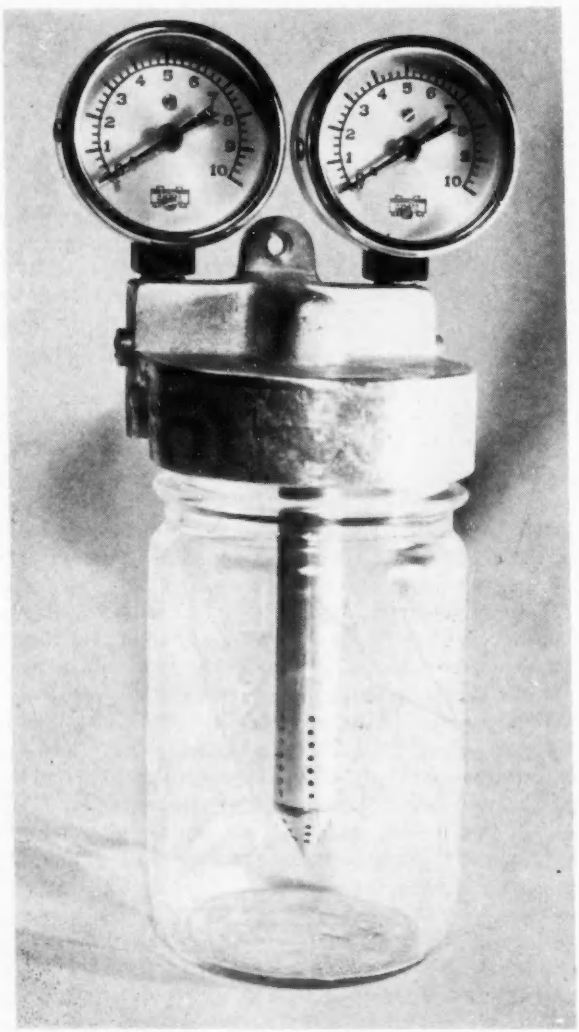
A third design of ignition harness is the type which employs tightly fitting grommets within the spark-plug elbow fittings and elsewhere within the system in an effort to prevent the conduction of water and other fluids into the shield chambers through the harness. Even though the method be successful in this respect, it is almost certain to be unsatisfactory for the reason that it too isolates the shield chambers, causing a very high concentration of contamination therein. In other words, this system has most of the faults characteristic of the "filled harness" and the type of harness employing the tightly woven shielding. In addition to these faults, it has an additional undesirable feature. The tightly fitting grommets pinch the wire causing a weakening of the insulation at these points. Experience has confirmed the contention that electrical puncture of the dielectric is most likely to occur at these points of stricture.

It is thus evident that all the systems just described have some inherent serious faults or inadequacies of some sort. None of them has features which meet all the fundamental requirements of a successful ignition distribution system.

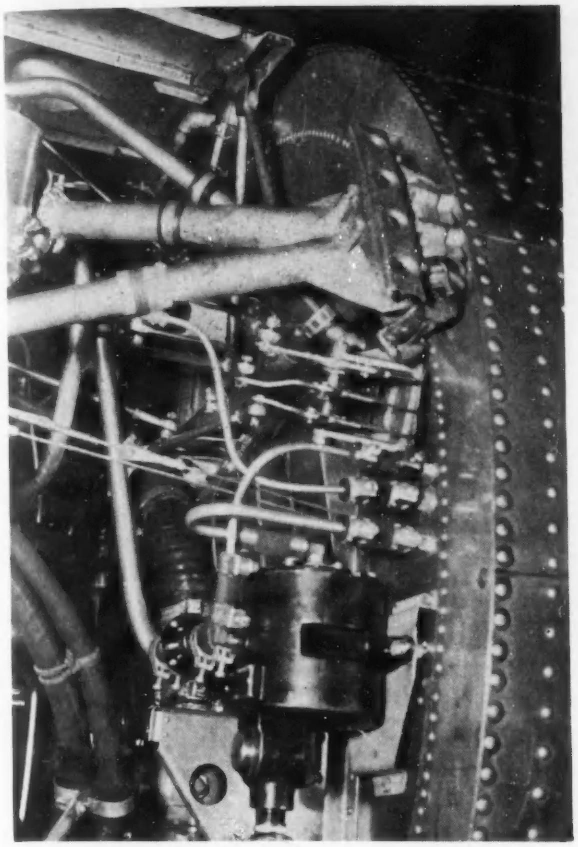
The supercharged aircraft ignition distribution system is a system which supplies air free from harmful ingredients at a pressure maintained in considerable excess of that of the surrounding atmosphere to the space within the spark-plug shield chambers, and is a system which provides ways and means by which the contaminated air may be removed from these regions. It is a feature of the system to utilize for the conduction of this air the type of ignition harness which already constitutes a conduit system to the spark plugs because, through such a type of harness, clean air can be transmitted to and into all the spark-plug shield chambers and adjacent spaces without adding tubing or additional conduit of any sort. The very important consideration of providing means for the escape of contaminated air from within is accomplished by utilizing for this purpose the inherent leaks of the harness or, if these prove insufficient, to provide such leaks purposely by drilling holes or otherwise making openings in the shielding.

A diagram illustrating the circuit arrangement used in the supercharged ignition system is shown in Fig. 2[10]. Part No. 37 is an engine-driven oil-lubricated pump which supplies air through oil separators 40 and 45, through a dehydrating unit, 52, into the shielded distribution systems at the fittings 61. Constrictions or small orifices, 57 and 59, are built into the lines, 55 and 56, to limit the air escape in case of excessive leakage in either harness, and to enable the gages, 60, to function as flowmeters as well as pressure indicators. The gages, the orifices, and the container for the drying material are built into a single unit illustrated in Fig. 3. The oil separator, 45, embodies an air cleaner consisting of a container filled with cotton, 50, and copper wool, 49, and the unit is provided with a relief valve, 46, which effects a release of excess air and operates to maintain the pressure in the system at a value

[10] United States Patent No. 2,213,478.



■ Fig. 3 – Container for drying agent threaded into an aluminum head fitted with pressure gages and incorporating constricting orifices to prevent excessive flow rates – The pointed and perforated tube is for the conduction of air into the desiccant



■ Fig. 4 – View of inboard side of right engine nacelle showing oil separator unit, incorporating a pressure relief valve, mounted to the firewall

in excess of that of the surrounding atmosphere. Fig. 4 shows the oil separator, 45, mounted on the inboard side of the right engine firewall.

If a source of compressed air free of oil vapor is available, the circuit may be simplified considerably by the omission of the oil separators. In such cases only the relief valve and the dehydrating unit are required. Should it be desired to supercharge the system from a high-pressure air hose while the plane is on the ground, a connector is provided in the right wheel well for this purpose. This connector has built into its structure a constriction, 65, and a spring-actuated ball-type check valve, 64. The constriction limits the flow of air to a rate somewhat greater than that required by the harnesses and the excess air passes out through the relief valve, 46, which, in turn, operates to maintain the system pressure at approximately the same value that exists during flight. The air-hose connector assembly is shown in Fig. 5.

A careful study of the features and operation of this system, together with reference to the previously explained fundamental nature of the ignition distribution problem, will disclose the fact that this supercharged, air-conditioned system provides the necessary relief from all the various forms of contamination encountered in service and thus insures reliable and trouble-free performance of the distribution network.

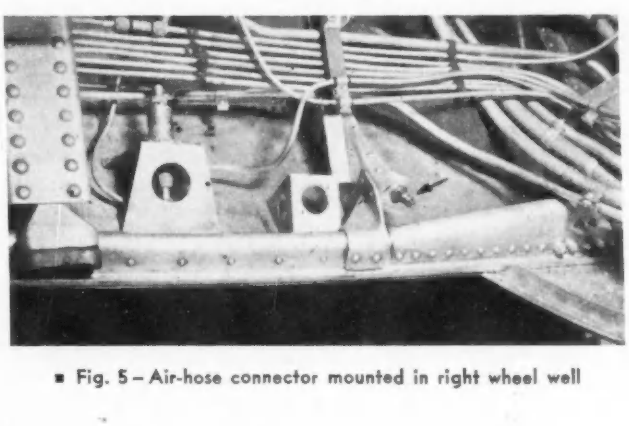
The presence of air within the harness at a pressure in adequate excess of that of the surrounding atmosphere will prevent the entrance through leaks of water, anti-icer fluid, and oils into the harness during all flight conditions including periods of rapid descent. The further feature of providing means for supercharging the harness from a hangar air-hose line or from a small electric-driven pump, while the airplane is on the ground with the motors stopped, will enable one to keep the system clear during long layovers in conditions of extremely high humidity or heavy rainfall.

By maintaining the air pressure within the shielding in considerable excess of the air pressure without, that is, by supercharging the system, the density of the air within is increased thereby suppressing the corona and the chemical actions associated therewith. Should there still be some nitric oxide or nitric dioxide formed within the shielding, it will be forced out through holes or openings provided for this purpose or forced out through the inherent leaks in the harness. Furthermore, by keeping the interior of the harness dry, as this method does, there will exist no moisture therein to combine with the nitric dioxide to form nitrous and nitric acids.

The gases and fumes which enter the harness from the combustion chambers through spark-plug core leaks are likewise removed. It is to be noted that it is not absolutely essential that holes or leaks be present in the immediate vicinity of the spark plug. The vapor pressures of all gases are very high at the high temperatures prevailing near the plugs, thus a powerful influence exists to cause the gases and vapors to be diffused back into the harness and there be forced out through the leaks or openings, wherever they might be. It is a fact, however, that leakages very frequently do appear at the connections of the harness elbows to the plugs.

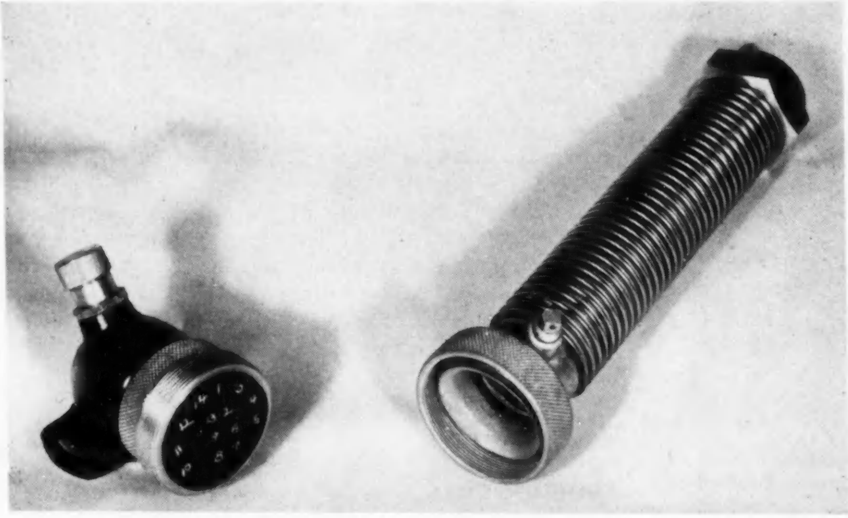
The fact that the insulation strength of air decreases rapidly with the decrease in air density which attends an increase in altitude, is a matter of great importance in modern aviation. To compensate for the loss of insulation solely by means of increases in air-gap clearances involves a large increase in the physical dimensions of the system and a considerable increase of weight and cost. Now a supercharge pressure of 5 lb per sq. in. over outside atmospheric pressure will maintain approximately sea-level pressures within the spark-plug shield chambers and all other spaces connected thereto while flying at an altitude of about 11,000 ft. Higher supercharge pressures will, of course, give even greater insulation gains. As the airplane increases its altitude, this type of air insulation failure makes its appearance in the spark-plug shield chambers. It is also true that this type of failure frequently appears within magnetos and high-tension distributors.

The feature of using the ignition harness conduit system



■ Fig. 5 – Air-hose connector mounted in right wheel well

for the distribution of the supercharged air simultaneously will make the magneto and the high-tension distributor recipient of all the benefits received by the spark-plug shield chambers because these units normally have their metallic casings or radio shielding directly attached to the metallic conduit or shielding of the ignition harness. Construction of the metallic casings or radio shielding of the magneto and the high-tension distributor must, of course, be such as to prevent excessive air leakage therethrough, or else these units will have to be shut off from the ignition harness by means of suitable air seals. Typical construction for such air seals is illustrated in Fig. 6. Obviously, if these units are shut off from the ignition harness, they will not receive the aforementioned benefits of the supercharge. Since the magneto and high-tension distributor are subject to failure attendant upon loss of air insulation and the presence of gases, acids, and moisture, especially under conditions of exceedingly high-altitude operation, construction of these units suitable for supercharging and attachment to the ignition harness is not only desirable but is essentially provident.



■ Fig. 6 – Magneto air-seal construction – Such seals are not required if magnetos are designed for supercharging. The section at the right shows air-intake fitting closed temporarily with pipe plug

The air which is injected into the ignition harness and transmitted therethrough to the spark-plug shield chambers, and to the magnetos and distributors if these be a part of the system, should be, as before stated, free from harmful ingredients. Best results can be obtained by providing for this purpose air which is not only clean, but air which has very low moisture content. For this reason it has been found desirable to draw this air from a region where the temperature is low and from a region which is protected from rain, sleet, snow, and ice, and to pass this air through some chemical drying compound which will remove essentially all of the water vapor content. By providing extremely dry air for injection into the ignition system, the possibility of having condensation of water vapor take place within the system is eliminated, and the ability of the air to absorb and carry out any moisture that might already exist within will be increased greatly. It is not to be inferred from these comments that it is always absolutely essential that the input air be subjected to a drying process. An air intake located in a region of low temperature is preferable to one located in a region of high temperature for the reason that the air from the low temperature region will, in general, have the least absolute humidity.

Activated alumina is one of several drying agents considered suitable for drying the input air. Activated alumina has the property of adsorbing moisture at substantially 100% efficiency until it has taken up between 12 and 14% of its dry weight. Beyond this point it continues to adsorb, but at decreasing efficiencies, until saturated, at which time it contains 20 to 25% adsorbed water. This material is commercially available in a color indicator grade and is indigo in color when fully reactivated. During the adsorption of moisture the material changes progressively through pink to white, and therefore provides an indication of its degree of saturation when used in a transparent container. Reactivation is accomplished by heating the desiccant within the temperature range of 350 to 600 F. Flow rates of approximately 10 to 20 cu ft per hr per lb of activated alumina are suitable for complete drying of air. Higher flow rates can be employed when complete drying is not required. Silica gel is another desiccant having similar properties and possessing the merit of adsorbing more moisture per pound of material when saturated.

In concluding this summary of the features of the supercharged system, attention is directed to the fact that the continuous flow of air through the ignition harness to the spark-plug shield chambers has a desirable cooling effect on the wire contained in these shield chambers and on the wire contained in the spaces adjacent thereto. The insulation on the wire in these regions frequently is found burned and charred due to the high temperatures existing within and near the spark plugs. This is especially true when engines are operating with high power output at high altitudes.

The supercharged ignition system has been subjected to severe laboratory and flight tests. In the laboratory, the entire ignition manifold together with all the individual conduits to the spark plugs, and the spark plugs themselves, which were in turn fitted to carbon dioxide "bombs," were submerged completely in water for 25 consecutive hr. At the end of this period the system was still functioning perfectly. A rigorous examination was then made of the interior of the system to determine whether or not any moisture was present therein. The examination disclosed

that the entire system was perfectly dry. Some water was then purposely admitted within the system and the submersion test was resumed. After a short period of time the interior of the system was re-examined and again found to be perfectly dry, thus proving that the method functioned to remove moisture from within. By special permission of the Civil Aeronautics Authority, the system was then installed on one engine of a transport airplane regularly operating from Chicago, Ill. to Seattle, Wash. After and throughout more than 500 hr of such operation under all conditions of flight and weather, the interior of the ignition distribution system and the spark-plug shield chambers remained entirely free from all moisture, fluids, oils, and other impurities. Many of the flights made during this period were made under conditions of heavy rainfall, or under conditions when much anti-icer fluid was used. During the first 250 hr of the aforementioned flight test the ignition harness on the other engine of the same airplane admitted sufficient moisture and anti-icer fluid to necessitate frequent draining of the manifold and drying of several of the leads within the shield chambers. Both ignition harnesses had like construction but, during the first 250 hr of the flight test, only the one ignition system was recipient of the benefits of the supercharge.

At the present time (Sept. 6, 1940) the entire Northwest Airlines Douglas DC-3 fleet is equipped with supercharged ignition distribution systems on both engines of each airplane. To date the accumulated record of flight service on these new harnesses exceeds 5,000,000 engine miles. At no time during this period has there been a single case of trouble attributable to harness contamination and there is not a single instance of any moisture or any other fluid being found within one of these systems.

The insulators which surround the ignition wire within the spark-plug shield chambers have been found to remain dry and clean throughout an entire engine service period of 600 hr, and complete harnesses have been reinstalled on newly overhauled engines without any replacement of insulators.

There has also been a very noticeable decrease in spark-plug trouble. In fact, the "off schedule" spark-plug removals have been practically eliminated.

A container holding approximately ¾ lb of activated alumina has been found to suffice for periods ranging from more than three weeks of operation in cold weather to approximately one week of operation in summer weather under conditions of high temperature and much rainfall. These figures apply to a two-engine airplane having 28 spark plugs per engine.

Radio reception also has been improved very greatly by the addition of the supercharged harnesses. Formerly pilots would have their communications disturbed by noises associated with spark-over and leakages in the spark-plug shield chambers, especially under conditions of rain. By eliminating this trouble the new system has improved greatly the reliability of communication.

DISCUSSION

P&W Integral Ignition System

– E. K. Von Mertens

Pratt & Whitney Aircraft Division, United Aircraft Corp.

It has been my good fortune to have had the opportunity to follow Mr. Swanson's work on supercharged ignition harnesses from the very beginning. I believe I have the right to express to him the appreciation of Pratt & Whitney Aircraft for keeping us well informed of the progress of his work from the very start.

My reason for giving this discussion is to clarify the position of the engine manufacturers. The truly great success of the supercharged ignition harness at Northwest Airlines seems to call for an explanation of the engine manufacturer as to why he does not jump at the opportunity to install this type of harness on new engines; or, are there reasons and objections against this type of ignition system?

I will discuss four main reasons why we have not accepted the supercharged harness for our engines, but first let me explain why these reasons are not a criticism of the work of Mr. Swanson's – they are a matter of viewpoint. Mr. Swanson's task was to improve the performance of a piece of equipment for his airline. He was permitted to make only minor changes; he had to use approved and standard parts and he was naturally limited by the safety regulations of the CAA, his airline, and the engine manufacturer. In contrast, we the engine manufacturer, had no such limitation, but our goal also is a different one. We not only have to consider operation of the equipment as a commercial carrier but also for many other conditions. We also have to consider to keep the ignition equipment in step with engine and installation developments, and we have to try to improve the entire ignition system so that it will operate without any attention between engine overhauls.

This difference in viewpoint leads to the first reason why we have not accepted the Swanson harness at this time. To solve our requirements we have developed a much improved ignition system for our latest engines. This system has been undergoing constant severe tests for the last year in our laboratories as well as in flight.

The second and most important reason against the use of pressure harness is its increase in weight and added complication to the engine. As Mr. Swanson has stated clearly in his paper, the most important benefit of his system is derived in the spark-plug shield. We know that this point could actually be called the bottleneck of engine operation, and we all know that something drastic should be done to improve the performance of the spark-plug barrel, cable sleeve, and spark-plug elbow. However, we do *not* agree that the solution should be found by adding a pump, a desiccator, many feet of pipe with valves, and controlling instruments. In other words, we believe that the effort necessary to use the pressure system is too cumbersome to solve a rather simple problem.

Our third objection is derived from the mechanical influence upon an ignition harness. The loose cables within the harness are constantly subjected to vibration and chafing action. We all have seen the horrible examples of chafed cables and we all know that one single sharp corner within the harness will almost guarantee a failure after a certain length of time. I know that the supercharged harness will retard failures of partly chafed cables by preventing moisture forming in the harness, but it has no provision made to prevent the chafing itself.

The cable manufacturers for years have complained that the mechanical strength required in standard cables just to increase the resistance against chafing and to permit the mechanic to pull the wire in place without stripping or breaking the cable, has handicapped them greatly in improving the electrical qualities. I will later show how this can be accomplished simply and effectively.

Our last reason for not adopting the supercharged system is the fact that we have, in our latest-type engines, to consider temperatures which are far above present standards. Materials used in present installations would not be sufficient to stand this rise in temperatures and breakdown of ignition harnesses through excessive heat would likely be most frequent. This overheating might be prevented by supercharging the harness during flight in the air, but the dangerous overheating occurs immediately after shutting down of the engine on the ground.

I would like to add to my criticisms a constructive thought and give a short outline of the new ignition system as used for the latest engines of Pratt & Whitney Aircraft. We believe the most important change was made when we decided to treat the new ignition system as one unit instead of three separate units namely magneto, harness, and spark plug. This decision leads to a design used on our latest-

type engine which has the magneto and harness as integral parts of each other. In fact, the distributor of the magneto is now a part of the harness. This was done to enable us to enclose all cables over their full length with a dense, high-dielectric compound. The harness is made as a casting of high strength. The next step was a natural one. Since the compound was preventing the movement of the cables and since the wire could be laid into the harness and need not be drawn through, it was possible to disregard all strength requirements of the cables. We selected a wire which would rather stand the heat requirement to be expected on future engines.

To make the integral design possible, we moved the magnetos from the rear section to the front of the engine and gained three new advantages. The magneto itself was removed out of a high-temperature zone onto the cool nose section of the engine; we shortened the cable length from magneto to spark plug by more than one-half, reducing the capacitance accordingly; and also we were able to keep the weight of the ignition system, in spite of the filling of the harness, the same as on present engines.

The leads from the manifold to spark plugs are handled in a similar way. We are using the same high-temperature cable as in the manifold which is cast into a conventional lead and elbow with high-dielectric compound. Now all is well except for the zone of the spark-plug bushing and the spark plug well which, as we know, is the region where the supercharging is doing its best job.

Let us for a moment go back to Mr. Swanson's paper. As he mentions, there are three major points to be considered: First, entering of undesirable substances from the outside; second, gases leaking through the spark plug; and last, the chemical action which accompanies corona discharge. We have added one more point to this, namely, the influence of heat. Our goal was to design a new spark-plug cable connection which would stand up under these conditions. We have solved this through design of a ceramic cable connector which is suspended rigidly within the spark-plug barrel through a joint similar to a high-pressure pipe connection to prevent any undesirable substances to reach the cable end.

To eliminate the influence of spark-plug leakage, we have made the spark-plug connector gas-tight so that no gases can get at the cable end. The filling compound of the spark-plug lead joins with the connector and prevents any flash-over possibility from the cable end to the ground. This sealing of the cable end has eliminated most of the failures derived from plug leakage and corona influences. The only flash-over path from the connector spring to the top of the spark plug has never been very serious, and it has been possible to increase the length of this path considerably without making a change on the spark plug. Finally, a ceramic material was selected to permit the use of higher temperatures without danger of carbonizing the connector. To improve the condition in the spark plug barrel further, we have made a spark-plug design which, in case of leakage, will not permit the gases to get into the spark-plug well, but will blow them harmlessly to the outside.

I hope that this discussion has helped to clarify our situation. We think that the supercharged ignition manifold has a definite function at a definite place but, as an engine manufacturer, we are forced to solve the ignition problems from every angle for all conditions.

U.S. Armored Force – Development and Employment

IN the press there has been considerable confusion as to the difference between the motorized unit and the mechanized or armored unit – and there is a vast difference. The motorized unit is one which transports a part or maybe all of its supplies, its weapons, and personnel to the battlefield in trucks. There the unit dismounts and goes into action, and most of its vehicles are sent to the rear. In very few instances are they employed as weapons on the battlefield. On the other hand, an armored unit transports all of its personnel in armed and armored vehicles, most of which, such as tanks, are employed as weapons. This type of unit can fight mounted, dismounted, at a halt or in motion – usually by a combination of all these methods.

Role of Armored Force and Fire Power

The Armored Force is particularly designed to execute all the mobile missions of the ground army – rapid strategical moves, deep envelopments, pursuit, exploitation of the break-through, and so on. It works intimately and constantly with observation and combat aviation. The ratio of machine-gun fire power in armored units to that of other arms is very great. For example, the Infantry streamlined division of eight or nine thousand men carries and operates about 250 0.30-caliber machine guns; the armored division of 11,000 men carries 4000 0.30-caliber machine guns – about fourteen times as many as the Infantry Division!

■ Limitation of Armored Forces

I do not mean by the foregoing that the Armored Division is more useful and more powerful in all instances than the Infantry Division, because it is not. The Armored Division is sensitive to terrain, cannot work in mountainous country and marshes, and is slowed down in terrain cut up by numerous stream lines. It is a powerful striking force but is weak in man-power for holding missions. It must be employed skillfully and forcefully at the appropriate time and place and be followed up promptly by other arms in order to attain the greatest success in extensive operations.

■ Tactical Employment

In discussing the tactical employment of an up-to-date armored unit, I wish to point out particularly the difference between present methods and World War methods. Today we are using tanks that can travel 50 mph and in the World War they had slow, cumbersome vehicles of a speed of 2 to 3 mph. In the World War we employed these armored vehicles piece-meal in small units and directed them at the most difficult defenses which were holding up our Infantry. This procedure is now entirely obsolete. For example, we now use the reconnaissance and infantry elements, and frequently supporting troops, to seek out the enemy line, to find a soft or weak spot in his dispositions. On this soft spot we concentrate support fire – machine gun, artillery, bombardment aviation and, when this spot is further softened up, we then drive through with masses of tanks, not at limited objectives but deep into rear installations and nerve centers to paralyze the very heart of enemy resistance. The tanks are then followed up by all available troops to give the knockout blow to an already staggering enemy. We may liken the armored unit to a spearhead directed at a soft and vital spot in the enemy's armor. When thrown forward into this vital spot, the sustained power behind the spearhead is the might of normal troops of all arms.

■ Other Vehicles in the Armored Force

An Armored Force does not consist entirely of tanks. There are tank battalions, of course, but in the armored division there are many other types of vehicles included in its make-up. The division consists of elements of all arms and services, mounted in armored vehicles peculiar to their missions.

Supply of the Armored Force

Upon its organization on July 10, 1940, the Armored Force consisted of 7411 officers and men and about 1800 vehicles of all types. Since that time it has expanded about 350%. The total strength of the Armored Force as now projected by the War Department will be about 84,000 officers and enlisted men and 20,000 vehicles, or an expansion of approximately 1200%.

From the foregoing, the magnitude of the task of equipping and supplying this force can be appreciated. We must immediately freeze the design of equipment and the factories must secure the necessary jigs, dies, and fixtures and go into mass production, which they are doing exceedingly well for scout cars, light tanks, and half tracks. Three large manufacturing concerns are now being tooled up to produce our medium tank, and early this spring we expect this tank to begin to flow in large quantities. Our truck transportation presents no serious problem as industry is capable of producing large quantities on short notice.

School

An Armored Force is the most complex and highly integrated of the organizations found in a modern army. Its demands for trained technicians are heavy and cover almost the entire field of occupational specialists. We are utilizing some civilian schools and facilities to assist us in this training. We have also established an Armored Force School with a capacity of 6000 students, and there we train the specialists required to operate, maintain, and repair the instruments and vehicles with which this force is equipped. Enlisted personnel are trained to be skilled workmen in a particular field, such as motor mechanics or radio operators. They are therefore given repeated practice in the operations they are to perform. Officers are trained as inspectors and supervisors of these operations and, while they are required to perform in detail such operations, emphasis is placed in their instruction on the theory of the operation and the technique of inspection and check of the operations performed by the mechanic.

Replacement Center

Each arm is required to establish a Replacement Center which receives untrained selectees and trains them to take their place as members of the more or less complicated fighting teams of their branch of the service. The normal Replacement Center trains men for one branch of the service and the training is more or less similar for all trainees. The Armored Force Replacement Center must train men for all the arms and services which compose the Armored Force. It too, then, becomes a highly technical component of this Force.

The Armored Division

At the start I would like to state that the Armored Division differs in two important respects from the Infantry and Cavalry Divisions: first as to mission, and secondly in the fact that the major elements of the Armored Division are all dissimilar. For example, in the Triangular Division of the Infantry there are three similar combat teams each of a regiment of infantry, battalion of artillery, engineers, and so on. In the Armored Division we have our major groups or echelons of different sizes, with different equipment and armament to perform different tasks.

Our Observation Aviation Squadron conducts general observation missions to our front and flank; these are the eyes of the commander pushed out to a distance of about 150 miles to gain early information of enemy movement. There are 13 planes in this squadron. They conduct general observation, liaison, and spotting and registration for the artillery for 24 hours a day. This means working in relays with care taken to have a reserve for critical periods.

The Reconnaissance Battalion – This unit constitutes the ground eyes of the commander, and helps give him, along with that from the air, the general picture ahead. It operates several hours in advance of the bulk of the Division, and checks terrain in question, brushes aside small groups, or passes around them, operates on the enemy's front and flanks to determine his movements and positions in order to get timely information back to the Division Commander. It consists of two Reconnaissance Companies in scout cars, one company of light tanks, and a rifle company in carriers. Radio, plane, and messenger keep it in touch with the Division.

The Armored Brigade constitutes the striking force of the whole division. It contains a total of 274 tanks in its two light regiments, 110 medium tanks, in its medium regiment, 24 howitzers in its artillery regiment, and innumerable machine guns, anti-tank guns and 81-mm mortars. It contains an engineer battalion for the purpose of repairing or constructing bridges, removing obstacles and, in general, keeping the brigade rolling. Since it occupies 41 miles of road space, it usually marches in multiple columns. The entire Division is built around the brigade. Everything else is intent on maneuvering the brigade into position where it can strike the decisive blow.

The infantry regiment differs from the customary one in that it moves in armored half-track carriers and has a large number of anti-tank weapons. Supported by an artillery howitzer battalion, it supports the brigade by seizing ground impassable for tanks, holding ground which the tanks have captured, and protecting the brigade when the latter must refuel, reorganize, or rest.

Within the service echelon, the Ordnance Company supplies the skilled mechanics and the spare parts required to maintain the fighting vehicles and weapons in operation. The Quartermaster Battalion not only supplies gasoline, oil, and rations but also maintains the wheeled vehicles of the division. The Medical Battalion collects and evacuates wounded personnel from all the units of the division.

Summarizing, I would like to call attention to three items:

1. The various type units making up an Armored Division: Reconnaissance – Air and Ground.

The two main combat elements – The Brigade with its mobile, hard-hitting tanks supported by artillery, and

The Infantry – Artillery element for attack, defense, delay and security.

2. This Division with its strength of 11,000 men, 19,000 weapons, and 2,500 vehicles – most of them armored – constitutes the fastest, hardest-hitting ground combat unit in the U. S. Army today.

3. This organization admits of much elasticity and enables a commander so to group his elements as to best meet a particular situation.

Excerpts from the paper: "The Development and Employment of the Armored Force," by Major-Gen. Charles L. Scott, Acting Chief of the Armored Force, presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 8, 1941.

Problems Relating to

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RESULTANT performance of the aircraft velocity-type supercharger is based upon inherent design form, together with added factors, including rotative speed, altitude, inlet-air condition, and outlet-air condition imposed upon it. Unfortunately, the supercharger and its controls frequently have been designed with concepts of utmost simplicity, manufacturing ease, or convenience of inheriting earlier designs as primary governing factors. If, instead, the performance problem had been studied carefully in all cases and designs had been formulated without permitting serious compromises to exist, we should still find that we had not been inconvenienced greatly in properly accomplishing desired ends in this relatively small mechanism. The most critical part of the airplane, considering today's performance problems, would be in a more suitably advanced stage.

Fundamental Effects

Let us briefly consider some fundamental effects of a customary gear-driven centrifugal supercharger in an aircraft engine.

The mass flow of supercharged air reaches a maximum during take-off. Under such circumstances in an engine designed for altitude operation the throttle is perhaps not opened fully; or, if it is, the engine speed, and therefore the impeller speed, is limited by action of a predetermined minimum propeller pitch. The supercharger designed for altitude is inherently capable of producing a far greater manifold boost than is safely permitted for take-off use. As the aircraft is put into a climb with the throttle partially closed, at low altitude the supercharger is dissipating much power to no useful purpose. The throttle is then gradually opened if a constant power is to be maintained as the critical power altitude is approached. At the latter altitude the air volumetric flow through the supercharger reaches a maximum, and the total pressure rise through the supercharger is approximately the manifold boost pressure referred to ambient atmospheric pressure. Only at this time is the supercharger used efficiently, and it may be that compromises in the supercharger design, due to low-altitude considerations such as prevention of induction air heating or surging tendencies, may have caused the design efficiency at the critical power altitude to be less than it otherwise would be.

Accordingly, speed-change methods of driving the supercharger have been devised and put into use to some extent. In the future the previously discussed elementary method of driving the supercharger unaided by special controls may be expected to disappear from aircraft, even in those intended to operate at comparatively low altitudes.

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Nov. 1, 1940.]

To state the problem further, it must be mentioned that the supercharger power consumption is a large factor in the power output of the engine. Whenever the engine air throttle is not fully opened, such as during low-power cruising at high altitudes with a high blower speed ratio, there is an unnecessary waste of engine power involved in

COMPARISONS are drawn, in this paper, between engine supercharger and cabin supercharger flow-control problems. Some new methods of obtaining efficient flow control are discussed. Interdependent factors existing between flow control and impeller speed control must be recognized.

It is pointed out that design features in the supercharger should be correlated closely with the type of control applied.

Effects arising from the connection of superchargers to receivers of large volume are presented. Necessity for regulation of flow, pressure, and rate of pressure change in pressure cabins requires the solution of numerous new problems. The advantage of simultaneous design of the super-

pumping air. The fuel consumption is thus raised and the wasted power is converted into heat in the induction air. All other things being equal, the heating of the air further raises the fuel consumption by requiring a richer fuel mixture to avoid detonation. The question of the attainable delivery head in a supercharger is not the only factor to be considered. The deciding factor may be the temperature rise instead, since an engine operates without knock, using a given fuel, only up to a certain induction air temperature.

Let us consider the corollary applied to the cabin supercharger, understanding that these superchargers have been driven at fixed speed ratio with respect to the engine mainly for mechanical simplicity. The efficiency aspects are even less favorable then compared to the engine, but the losses are confined to a power consumption of far less magnitude. Therefore, the subject drive arrangement is more readily excusable than it should be in the case of the engine.

Unlike the engine supercharger, the cabin supercharger can have a condition of maximum impeller speed simultaneously with its constant-flow control exerting maximum

the CONTROL of FLOW in SUPERCHARGERS

effort to keep the flow from increasing above the desired value. This tends to heat the air considerably. It is therefore necessary during warm weather to prevent excess heat from being introduced into the cabin at low altitude by closing off the supply of supercharged air to the cabin. At this time the cabin receives ventilation from a rammed auxiliary air system. An appropriate amount of air is simultaneously spilled from the supercharger by a flow-responsive or thermally-responsive surge relief valve. An alternative method is to declutch the supercharger from the engine.

charger and controls, and the desirability of integral supercharger and control units, are stressed. It has been necessary to overcome many mechanical problems in order to produce units of a type suited for pressure cabin operation. Typical control constructions are described.

Aircraft manufacturing concerns are now extensively involved in contributing to the supercharger art. Trends in engine superchargers and in engine supercharger controls will be stimulated by developments in the field of cabin supercharging. Recognition of the paramount importance of engine superchargers in high-altitude flight and the prospects for considerable performance gains encourage the exercise of an aggressive design attitude in the production of new features for superchargers.

At intermediate altitudes, when supercharging must be applied to the cabin, a suitable method of dissipating excess heat such as an intercooler or, preferably, in the case of a luxury air liner, a true air-conditioning system operating on a refrigeration cycle may be provided.

Both engine and cabin superchargers are being built with multispeed gears and clutches, which justifiable mechanical complication greatly improves the flexibility of the superchargers. However, even with a two-speed gear, the performance of the engine can be wasteful unless flight operating schedules are fitted to the known characteristics of the supercharger, but this is difficult due to weather conditions.

From the standpoint of the cabin supercharger complication is involved if the supercharger is constructed as a single unit with an accessory gear box, for the minimum impeller speed may be set by requirements of the other accessories unless there is a special change speed gearing division between the gear box and supercharger.

Hydraulic drives of the positive-displacement type with infinitely variable speed controls can be provided for both engine and cabin superchargers. In the case of the engine some complication and added weight is involved, but there is a fruitful field for development progress. Considering cabin superchargers, the added weight is less justified purely from fuel-conservation standpoints but, if the general arrangement of the airplane advocates the placement of cabin superchargers far from the engines, the hydraulic drive falls into a position of considerable desirability.

Fluid couplings used either as torque converters or in conjunction with gearsets may offer some advantage from weight standpoints compared with the positive-displacement drives. By construction, proximity of the power-generating member to the power-receiving member is required for high efficiency. However, far more attention may be directed to this type of drive in the future.

It might be mentioned that mineral oil is the present preferred liquid for this type of transmission due to its lubricating value. Heavier liquids, such as "Aroclor," having high specific gravity, low viscosity, and low vapor pressure may be more desirable from purely power-transmission standpoints. Mercury would, of course, be the most desirable transmission fluid but, among other reasons, due to bad effects on couplings and cooler, it has not come into use. Oil turbines operating off the aircraft hydraulic system are alluring from control, simplicity, and reliability standpoints, but to date it is not possible to report that their efficiency is comparable to that of positive-displacement oil drives.

■ Electric Drives

Electric drives provide variable speeds for superchargers with reasonable efficiency by the use of special field controls or selective series-paralleling arrangements. There is no particular incentive to use an electric supercharger drive for engines because of bulkiness, weight, and the fact that drive efficiency is not as high as desired. However, in the case of cabin superchargers of small size, this type of drive provides flexibility and sufficiently good efficiency. It is particularly suited to "pack" control units, incorporating the control and supercharger assembly as an easily removable unit in the aircraft cabin.

Engine exhaust turbine driven superchargers constitute a comprehensive subject which hardly can be dealt with in this paper, except to mention that advantages of speed variability and exhaust energy regained at high altitudes are partially reduced by some practical disadvantages, such as the resulting massive and complicated installation and

greater difficulty of applying jet exhausts. Advances in the direction of greater compactness of the installation are anticipated. Kinetic energy waste at the bypass gate can be minimized by directing discharge counter to the direction of flight. While the exhaust turbine supercharger is highly suitable for use in conjunction with engines in some aircraft, its prospects for cabin supercharging are less encouraging, partly due to the too great dependence upon the power conditions in the engine.

Mechanical-driven two-stage superchargers with selective staging and with intercooling are more efficient than two-speed single-stage superchargers. Intercooling between stages, as is also customary with turbo-superchargers, improves efficiency of compression and is suited to cabin superchargers designed to operate at altitudes over 25,000 or 30,000 ft.

In the same general category with the exhaust turbine supercharger is the supercharger boost employing secondary working fluid generated by waste heat from the engine. This arrangement is recommendable from control and installation standpoints and may offer the maximum powerplant efficiency. It permits the use of exhaust outlet jets, but constitutes a difficult design problem in order to capitalize on its inherent advantages. Similar to the exhaust turbine supercharger, it recovers engine waste heat but is less dependent on short-period exhaust stack conditions and possesses potential short-period overload capacity due to heat storage.

Separate gasoline engines driving superchargers at variable speed for engine or cabin supercharging reduce reliability, increase service cost, and are excessively heavy.

■ Air Controls Applied

Special air controls to obtain effects of variable-speed control can be applied to superchargers. Kinetic energy ordinarily wasted in the performance of direct air throttling is converted into useful energy. The suction air is caused to react propulsively against the supercharger impeller system. This offers a flexible, simple, compact and inexpensive control for engine and cabin superchargers. Its exact efficiency related to variable-speed drive controls is being investigated at present.

Referring now principally to the subject of applied control of engine or cabin superchargers operating at a given speed, it must be emphasized that the speed or flow control and the design features of the supercharger should be closely correlated. Thus, from practical standpoints, it is desirable that superchargers and controls be designed simultaneously.

Pressure-side throttling is not ideal for centrifugal superchargers with radial impellers as regards effect on the engine output, because the engine operating point travels into the supercharger characteristic field, the travel being farther the greater the critical altitude and supercharger pressure. Nor is it desirable to use pure pressure-side throttling in cabin superchargers. Pressure-side throttling involves the presence of more dense air in the supercharger itself tending to produce a greater pressure rise and temperature rise when frequently not wanted. It should be mentioned, however, from the standpoint of academic interest, that one may introduce an intercooler between a cabin supercharger and pressure-side "throttle" performing adiabatic expansion by doing useful work with the air, to introduce cold air into the cabin.

If, in a given installation, the pressure-side throttle is to be used in order to provide an integral control unit for pressure cabins, particular service advantages of the composite unit being duly recognized, then performance advantage results from the installation of a complementary suction side valve. The latter valve is closed as the duct pressure exceeds the maximum differential cabin pressure at altitude by a predetermined small value, the force to close the suction side valve being obtained from the pressure side of the supercharger. The suction side valve acts not only as a flow-control throttle and surge damper, but also as a supplementary emergency cabin pressure differential control and as a means of reducing the air forces which the main flow valve would otherwise have to control. It furthermore tends to reduce noises in the main control unit in the central portion of the aircraft. As in the case of control by a pressure-side throttle alone, there is still sufficient heat generated in the air from the supercharger to jacket and heat the cabin outflow valve for preventing ice formation. By test the power consumption and consequently the temperature rise in the blower during take-off condition have been reduced by combined pressure-side and suction-side throttling. Pressure rise in the blower during take-off was reduced to about two-thirds of the value with pressure-side throttling alone.

■ Utilized to Heat Cabin

Cabin superchargers can be overloaded intentionally or can be made to work against a manually closed discharge valve to put additional heat into the cabin during cold weather in the event of failure of a heating system.

In one of the pressure-cabin control systems developed by the Lockheed Aircraft Corp., the flow into the cabin is maintained constant in the supercharger itself by special flow and power-economizing controls.

Within the supercharger itself, a reciprocable spring-loaded piston is displaced in accordance with the total differential pressure existing in a venturi plus the air pressure drop in a steam-heating radiator core attached to the supercharger pressure side. The piston cooperates through a follow-up air motor control for the suction-side butterfly valve. One side of the motor piston is exposed to supercharger discharge pressure subject to the follow-up control and the opposite side to supercharger inlet pressure to effect an actuating force.

The described inflow system alone would maintain a substantially constant dynamic flow through the supercharger regardless of impeller speed or altitude by suction-side throttling. However, since engine speed varies considerably and the supercharger must operate throughout a very great altitude range, a cooperative control for power conservation when the supply capabilities of the supercharger tend to exceed the cabin requirements greatly is employed as an adjunct to the flow control. A nozzle ring for admitting air into the impeller near the periphery thereof is connected to the suction-side duct via a backflow check valve. The nozzles in the ring face in the direction of motion of the impeller. The size of the nozzles is determined by normal cabin inflow requirements while the engine is operating at climbing speed at low altitude with the suction-side control valve being completely closed.

During periods when the supercharger performance requirements are severe, the suction-side throttle is opened automatically to a considerable extent. The air pressure in the impeller channels opposite the nozzle ring is then great enough to force air backward through the nozzle

ing, but this is opposed by the check valve; in other words, the supercharger operates conventionally, but with a relatively small amount of suction-side throttling being employed. However, when there is an excessive flow tendency at lower altitudes, the inflow motor will nearly or completely close the suction-side valve. During the latter condition, the pressure of the air in the impeller channels opposite the nozzle ring falls below inlet duct pressure because of the action of the supercharger diffuser beyond. The air is drawn through the nozzle ring, being adiabatically expanded and accelerated so that it enters the impeller at a speed near or even slightly exceeding that of the impeller at the point of admission. The result is that the air temperature and pressure rise through the supercharger is considerably reduced. In addition, the nozzle ring comprises a supplementary air inlet which bypasses a large portion of the impeller channels, forming, as it were, a selective two-stage blower from a single impeller and diffuser.

■ Minimizing Surge

Superchargers with purely radial impellers and vane diffusers tend to surge when the intake quantity goes below that corresponding to maximum delivery head. This disturbance can be held partially in check if the throttle member is mounted very close to the impeller inlet. Thus, if the induction flow is reduced below the design value, tending to produce instability in the diffuser, the nozzle ring may assist greatly in prevention of pulsation, due to air inertia and damping effects in the nozzles.

The flow control of the described type always maintains a high operating efficiency and yet permits a fixed drive speed relationship to exist between the supercharger and engine. If a gear box is incorporated with the supercharger for driving other accessories, the latter units are then provided with sufficient operating speed at low altitudes without necessity to employ a gear separation scheme.

Deleterious inertia effects which might arise from a driven assembly of this size are avoided by employing an overrunning clutch to prevent engine backfire or reverse torque from acting upon the supercharger drive. In addition, the use of the overrunning clutch limits the magnitude of the torque as the drive passes through any resonant condition when the engine is coming up to normal cruising speed. It also greatly reduces the drive forces during very rough engine operation, if such should exist.

The described control provides constant flow to the cabin. Were it not employed, the supercharger would tend to discharge cyclically first excessive, then insufficient quantity of air into the large elastic reservoir represented by the cabin. At very high altitudes small changes in engine speed or in altitude of the aircraft might even result in occasional backward flow in the supercharger, were no check valve placed in the supercharger discharge duct.

In conjunction with this cabin inflow control scheme, a variety of pressure events in the cabin, suited to operating conditions, are attained by regulating an outflow orifice from the cabin to the atmosphere. The cabin outflow valve is positioned by remote pneumatic control which has the advantage of bringing both the cabin absolute pressure sensitive mechanism and the cabin pressure rate sensitive mechanism to the flight compartment, where these controls can be readily adjusted manually. Very delicate mechanisms, of which the latter can be called characteristic, are best located on a shock-mounted instrument panel where they are not required to withstand severe vibration and where they can be removed readily for inspection. In addition, these mechanisms are combined with a supercharger load adjustment valve in a compact unit having symmetry of appearance enabling the operator to ascertain at a glance the exact adjustment of the pressure cabin control adjustments.

The outflow valve is actuated by the air-pressure differential between the cabin and the suction side of the supercharger. In order that suction properly may position the outflow valve, sensitive pneumatic controls are interposed in the vacuum line. The first control, for insuring stability, consists of a spring-loaded follow-up piston located directly above the outflow valve air motor piston. Control vacuum is admitted to the upper side of the follow-up piston. In order to raise the outflow valve, the vacuum first must raise the follow-up piston against the spring force. This upward motion causes the stem of the follow-up piston to separate from the top of the air-motor piston. Inasmuch as there is a bore in the stem of the follow-up piston communicating with the upper side thereof, the upward motion permits the vacuum to extend to the upper side of the motor piston as well, thereby raising the motor piston. However, the poppet and the air motor piston cannot overtravel because this would close the gap between the stem and the air motor piston.

Assuming now that the valve is to be lowered, the vacuum imposed upon the upper side of the follow-up piston is decreased slightly to allow the spring to lower it. Then the supporting vacuum above the air motor piston is destroyed by the stem closing its end gap. The cabin outflow air force on the poppet plus the direct spring force tend to lower the air motor piston. However, the poppet and air motor piston can travel downward only as far as the decrease in imposed vacuum permits the follow-up piston spring to expand.

Accordingly a follow-up control is produced so that changes in outflow air force acting upon the poppet cannot affect its position.

It is obvious that the movement of the outflow valve from uppermost to lowermost position can be produced by a relatively small change in vacuum regardless of cabin pressure. Vacuum control range is determined by the spring rate and the magnitude of the working vacuum within the range is determined by the spring force.

■ Limiting Pressure Differential

Cabin pressure controls must be made subservient to a differential pressure limit control established by the strength of the fuselage. Therefore, in the subject controls the limit differential is permitted to act directly upon the outflow valve follow-up piston by means of a pneumatic tube extending from the upper side of the follow-up piston past an adjustable pressure-reducing valve to the atmosphere. When the limit cabin pressure is reached, the other controls are overcome and the outflow valve remains open far enough to prevent further rise in cabin pressure relative to outside pressure.

The double use of the follow-up piston for both cabin differential pressure limitation and for intermediate cabin control pressure as well, yields a smooth and more continuous control. Heretofore, a mechanical pickup system had been used in order to cause the differential pressure limit to come into effect, and this latter method, while

generally satisfactory, cannot have so perfect a continuity of action due to mechanical friction.

The suction at the inlet of the supercharger between the butterfly valve and the impeller, which is used indirectly to position the cabin outflow valve, varies considerably due to changes in engine speed and altitude. Accordingly, an accurate reducing valve is included in the control so that the cabin pressure sensitive elements are provided with a generally constant working vacuum compared to cabin pressure. Accordingly, change of engine speed or of altitude of the aircraft does not require the sensitive elements to seek a new position of equilibrium.

■ Cabin Altitude Control

The cabin absolute pressure or "altitude" control employs an altimeter bellows exposed to cabin pressure to position mechanically a balanced beam valve. The vacuum extending from the reducing valve to the cabin outflow valve is required to pass through this beam valve. Due to the follow-up in the cabin outflow valve unit the remote control afforded is stable. Previously it had not been possible to control the outflow valve by a remotely placed cabin altitude sensitive device because the outflow valve could overrun and would hunt very badly, especially as a result of interconnecting line capacity. Cabin "altitude" adjustment may be provided within any desired range, sea-level to 12,000 ft, for example. Once the adjustment knob is placed at a given value of either barometric pressure or pressure altitude indicated on its dial, the cabin pressure commencing at the altitude indicated will not change perceptibly during further increase of altitude until the cabin differential pressure limit is reached.

The cabin change of pressure rate or "vertical speed" control includes a beam valve which operates an air relay in series with the "altitude" control beam valve to increase or decrease the vacuum imposed upon the outflow valve. The "vertical speed" beam valve is positioned by a change of pressure rate sensitive element of construction basically similar to that of a vertical speed indicator, but more rugged.

There are also other forms of pressure rate responsive devices which are suitable for pressure cabin controls, such as an altimeter bellows restrained in speed of motion by means of an oil dashpot, and compensated for changes in temperature as well as air density in order to sense apparent vertical speed in the cabin.

A third control knob on the adjustment panel actuates a compound valve to stop the flow through the pressure altitude control, or bypass vacuum around the pressure altitude control, as the case may be, thereby providing a direct emergency or ground check manual control for the outflow valves.

There are various ways in which the pressure cabin control adjustments can be set to effect different automatic operations during flight. The simplest mode of operation consists in having the cabin "vertical speed" control set to "infinity." Then the "altitude" control will govern. Assuming that the pressure altitude control is set to 8000 ft, the cabin pressure will at all times be equal to the outside atmospheric value between sea level and 8000 ft. Upon the attainment of 8000 ft, as the aircraft continues to ascend, the cabin pressure will no longer follow the atmospheric pressure but will remain constant until an altitude of 20,000 ft, taken for example, is reached. If there is further ascent beyond 20,000 ft, the cabin pressure will always remain at the limit differential pressure with respect to outside atmospheric pressure.

Let us now assume that the cabin rate of pressure change control is set to a value of 200 fpm, instead of to the value "infinity," and the previous flight repeated at an actual aircraft speed of ascent greater than 200 fpm. Then the "vertical speed" control will cause pressure in the cabin with respect to the atmosphere to become positive immediately during ascent. Accordingly, when the aircraft reaches 8000-ft altitude, there will be a considerable positive pressure in the cabin but, upon further increase of altitude, between 8000 and 20,000 ft, the cabin pressure will no longer change because the "altitude" control will then take over. During ascent beyond 20,000 ft, control will be afforded by the differential pressure limit control for the protection of the cabin structure. During descent, from any altitude greater than 20,000 ft down to 20,000 ft, the cabin pressure may fall below the limit differential if necessary, in order that the 200 fpm rate of pressure change may not be exceeded. If the aircraft is piloted down to 15,000 ft, for example, and leveled off, the pressure will stabilize out at the rate of 200 fpm to the setting of 8000 ft altitude. If the aircraft is caused subsequently to descend further at any rate of speed, the cabin pressure will not change until 8000-ft elevation has been reached. Below 8000 ft the cabin pressure always will be the same as outside atmospheric pressure because an auxiliary inlet relief valve to the cabin prevents a negative differential pressure from acting upon the cabin.

It is evident that latter descent from 8000 ft may be uncomfortable to the passengers if the aircraft descent rate is very high, although this is advantageous from the standpoint of quick schedules, particularly if the aircraft is approaching a landing field beyond high mountains. Therefore in the Lockheed system when the field barometer reading is received by radio, the aircraft operator can set the "altitude" adjustment knob to the exact barometric pressure of the landing field long before the descent is started. The cabin pressure then immediately starts to increase at the comfortable rate of 200 fpm, for example, to landing field pressure. There is no danger of the aircraft landing with a plus pressure in the cabin as might otherwise be the case without the "altitude" adjustment.

■ Altitude Warning Light

An "altitude" warning light is located in the center of the symmetrical control adjustment panel. The pointers of all knobs are directed at the warning light during average flight operations. The warning light consists of a piston-borne bezel-ring light bulb normally held out of electrical contact by the control line vacuum at the discharge of the panel reducing valve. If, for any reason, the cabin pressure falls below the setting of the "altitude" control knob, the resultant overtravel of the "altitude" control beam valve breaks the vacuum supporting the piston and the light becomes energized. In this manner duplication of expensive aneroid elements is avoided as well as attendant problems in obtaining calibration between two separate aneroids.

The supercharger load adjustment, constituting an air bleed line into the flow-measuring venturi throat of the supercharger flow controls, is employed as an emergency inflow increasing device in the event of serious leakage in the cabin. It should be mentioned, however, that the superchargers also are overloaded automatically by con-

tection of suitable air bleed valves operated by the cabin outflow valves in case a severe leak has occurred in the cabin, causing complete closure of the outflow valves. Furthermore, the flow into the cabin can be adjusted conveniently to suit weather conditions. In extremely cold weather the superchargers can be overloaded to provide a greater quantity of warm compressed air. During unusually hot weather the inflow may be decreased so that the load upon the cabin air cooling system will not be too severe.

In general, pneumatic controls for pressure cabins may employ valve motor pistons having an area sufficiently large to overcome forces which act on the valve, so that the same air pressure differential acting across the valve can be employed in the motor cylinder to operate the valve positively. Controls also can be constructed along this principle for engine supercharger boost regulation, to avoid hydraulic complication.

The motor piston attached directly to a poppet valve by a stem in a bushing may be spaced from the cylinder wall by a fraction of a thousandth of an inch, eliminating diaphragms or rings. Pistons of this type are more compact than diaphragms because the entire area is effective in moving the valve motor. Air leakage may be approximately 0.15 cfm, for example.

In order to avoid flutter, a follow-up air feed mechanism should be provided to operate the motor piston. In the case of very light motor pistons and valve assemblies, it may be advisable to increase effective inertia and dynamic friction by causing the stem to engage its bushing with a spiral thread. Rapid upward or downward motions are thereby damped due to rotary drag in the bushing, and introduction of angular accelerations of the moving assembly about the stem axis lowers the frequency below the resonant range.

■ Eliminating Whistling

Helmholz' resonance effects sometimes occur in castings surrounding the valve. However, whistling generally may be eliminated by stiffening of the valve rim, by slightly changing the casting volume, by coating the inner surface of the casting with deadening material or, better still, by eliminating the housing altogether if possible.

Air motor parts may be operated without lubrication. However, cooperative metal combinations must be selected carefully.

Thus, the piston-cylinder combination must be made of materials of suitable differential expansion characteristics so that leakage will not change with change of temperature. Non-galling metal combinations should be used. Pistons spun from sheet metal should be shaped so that they do not tend to "pan" or "hook" the cylinder walls. Stresses should be relieved prior to grinding the rim; otherwise warpage may cause wall friction.

All air entering the piston cylinder or other part of sensitive controls should be filtered through screens of fine mesh – 250, for example. It is necessary that these screens be seated properly in the valve motor housings.

Corrosion may be avoided on dry surfaces by chromium plating bronze or steel parts, and by applying anodized or dichromate coatings to light metals. Expansion of air in the controls can cause the condensation of moisture.

Supercharger flow can be measured by a venturi to effect constant-flow control. The venturi preferably is constructed as a venturi within a venturi, in order to obtain a large working depression with a small pressure drop through the flow system.

The development of an efficient compounded venturi of this type can be quickly accomplished from fabricated dural cones lined inside with plaster of paris. Optimum shape is arrived at easily by filing or scraping the interior surfaces between trial-and-error flow tests. The production venturi assembly may then be produced accurately and inexpensively by the Antioch casting process which yields accurate and smooth surfaces without profile machining.

It is sometimes essential to employ two depression reference tubes extending into the throat of the inner venturi, one to effect the desired flow control, and the other for a flow-measuring gage, so that a true indication of the flow always may be obtained at the gage, regardless of action of flow overloading controls, or of changes in leakage in control parts with clearances.

A flow-measuring venturi can be made to maintain constancy of flow through a supercharger, either by correct positioning of control valves, or by indirect control of impeller speed through suitable governing means.

Surge Valve Provided

In conjunction with pressure-side cabin supercharger controls it is essential to provide a surge valve in the duct between the supercharger and cabin. Cessation of supercharger flow into the cabin should cause the surge valve to open. The adjustment of the valve must be accurate; otherwise it might tend to open when it should not, as, for example, when the supercharger overloading control comes into operation. In a typical case the supercharger overloading control functions by bleeding air into the flow-measuring venturi throat, thereby decreasing the vacuum slightly and raising the flow control setting. If the setting of the surge valve blow-off point is too close to rated cabin flow value, the surge valve may open when it should not. A spring-loaded pressure relief valve is not satisfactory as a surge valve since it only partially limits surging tendency; that is, it merely chops the tops off the surge peaks. A thermally responsive bi-metal coil may be installed in the surge valve to act as an emergency opening device in case of failure of the surge flow control action.

The proper operation of surge valves is important when the pressure side throttle is closed. Failure of a surge valve to open at a time when the cabin inflow valve is completely closed manually might result in damage to the supercharger discharge duct under the influence of the surging force, which may, for example, be accompanied by air pressure change from 0 to 35 in. of hg gage pressure in a rapid sequence of fluctuations.

Integral supercharger and inflow control units are highly desirable from weight, compactness, and servicing standpoints. It is generally an advantage to join as many units together as is possible, since this facilitates bench-testing of an entire unit and saves weight.

For instance, it is possible to combine cabin supercharger, supercharger controls, accessory gear box, heating system radiator, and heating system controls into one unit which can be replaced quickly in the airplane by a completely serviced unit. The servicing department can apply more readily systematic overhaul methods to the assembly and can mount it on an automobile engine dynamometer for final performance check before placing it on the shelf awaiting installation.

The discussions of this article referring to control of cabin or engine superchargers apply to either radial- or axial-flow velocity type blowers, since these appear to offer most promise when considered from standpoints of high altitude operation in aircraft. It is appropriate to comment especially on the badly neglected axial-flow type blower which can be controlled in somewhat the same manner as centrifugal-type blowers. The axial blower has higher efficiency than the radial type, but the best working range is rather restricted. Therefore variable speed is more positively indicated.

In an exemplary case, at full speed stable operation will only exist down to about 90% volume, beyond which point the compressor will commence surging. This effect occurs at reduced volumes because then the small axial velocity component causes the air to enter the blades at an extremely sharp angle, which results in loss of contact on the backs of the blades and subsequent unstable conditions. The axial compressor may be operated stably at reduced volumes by decreasing the speed.

In contrast with the type of operation exhibited by the axial compressor is that of a typical centrifugal compressor with stable range extending to less than 50% volume at full speed.

The axial blower tends to have a uniform flow characteristic. Considering installation in the case of radial blowers, a large entry section necessarily leads to a still larger impeller diameter; but, in the case of axial blowers, the diameter of the entry section determines the diameter of the blower, unless a final radial stage is used.

The axial blower is longer, but fits into certain types of installations better, particularly those of in-line or V engines. The rammed inlet of the axial blower may be directed more readily into the slipstream without the use of bends. The combination axial and radial blower may be the best compromise for some installations.

It is anticipated that special types of blades operable in supersonic regions with high efficiency will decrease size of blowers of both radial and axial-flow types.

■ Discussion of Construction

Some remarks are in order relative to details of radial-flow supercharger construction which affect controllability and efficiency. It has been aircraft custom to lighten the impeller backplate between the impeller vanes and near the periphery of the impeller by means of circular grooves. The high-velocity air is required to pass over an abrupt change in section. This introduces frictional losses in the impeller and furthermore the air leaving the impeller is so turbulent that the effectiveness of the diffuser may be hampered. The minimum flow value without surging tends to be increased. Therefore, in order to eliminate this rough spot in the air flow course, the backplate should be extended to the periphery and submerged into a circular recess in the housing of the supercharger. If, in certain cases, it appears necessary to reduce the inertia of the impeller, this can be done by reducing the diameter of the backplate and leaving a portion of the impeller vanes unsupported for a short length. However, care must be exercised to avoid blade vibration frequency falling within disturbing force frequency of the operating range, that is the impeller speed times the number of diffuser vanes. The natural frequency of blades can be determined by plucking them and picking up the vibration on an oscillograph.

Another point which should be mentioned is the desirability of using fully shrouded impellers for superchargers of large output. In cabin superchargers, however, it is sometimes questionable whether this more difficult construction is warranted. Fully shrouded impellers sometimes yield as much as 5% gain in efficiency compared to half-shrouded impellers. The presence of the inlet-side shroud tends to reduce verticular motion in the impeller channels by virtue of less unbalanced tangential drag component acting on the air. Then, the diffuser is capable of acting efficiently at lower flow values and the stable flow range is increased.

The importance of good axial air approach to the impeller for the attainment of optimum supercharger characteristics is recognized generally. The essential consideration as regards the axial approach is not so much attainment of a flow direction parallel to the axis as certainty with regard to the flow direction and uniform flow distribution at the impeller inlet. This also reduces turbulence in the impeller channel and tends to increase the stable range of flow in the supercharger. Accordingly, inlet fans should be used in radial-flow superchargers. The fan employs radial blades with generous curvature, better than that provided by curling the leading edges of impeller vanes, in order to reduce shock of the air entering the impeller. This permits the impeller to be supplied with air free from undue turbulence. The importance of inlet losses is great. It should be borne in mind that the effect of loss at the outlet of the supercharger consists of the inlet loss multiplied by the pressure ratio of the supercharger.

Non-uniformity of flow at the inlet may be reflected at the outlet of the supercharger with a considerable loss of efficiency. In engines unbalanced entrance of liquid fuel at the impeller inlet results in unbalanced discharge at the supercharger outlet. In conventional radial engines uniform diffuser action may be interfered with due to induction pulsations occurring at different points around the diffuser.

Improvements in supercharger efficiency and stability through the use of a well-designed inlet fan are substantial. This, together with the installation of a diffuser with vanes, produces a substantially greater pressure ratio in the supercharger. However, a vane-type diffuser introduces a definite minimum flow point for stability. In this regard the "snail" diffuser is superior, for it tends to be stable throughout the flow range, but generally has lower maximum efficiency.

A diffuser may be constructed from a thin steel disc, steel vanes spotwelded to it. The thinness of the vanes constructed of this very durable material eliminates interference effect sometimes encountered with the customarily used thicker dural vanes and is inexpensive from production standpoints.

In conclusion, we should all have complete cognizance of the importance of supercharger development work yet to be done. Airplane manufacturing concerns throughout the country have become more vitally interested in supercharger performance today than ever before, and are therefore assisting in bringing into common practice advances which should be applied more universally.

It should be mentioned that the Lockheed Aircraft Corp. has constructed the two very large pressure cabin research chambers and has set up a well equipped supercharger research laboratory in suitable keeping with its high-altitude aircraft designs and with its record of having produced the first pressure-cabin aircraft in this country.

Development of Wide-Rim Tires for Passenger Cars

by E. A. ROBERTS
The Firestone Tire & Rubber Co.

RESULTS of tests conducted on wide-rim tires announced in this paper show that:

1. There is an average improvement of 20% in non-skid tread mileage, the increase ranging from 5% for easy driving conditions to 80% for tests at maximum speed under hard driving conditions.

2. Stability and cornering power increase with rim width – approximately in the same proportion.

3. The effect of increased stability is very evident in improved cross-wind handling, especially at high speed.

4. From 2 to 4 lb per sq in. reduction in tire pressure was found necessary with wide-rim tires to produce equivalent ride, equal harshness, thump, and so on.

THE wide-base-rim tire program recently presented to the motor-car industry was the result of an analysis made by Firestone engineers looking toward the ever-increasing demands made of the present-day motor-car tire. Two years of development work and testing were engaged in to prove that these ideas and designs were practical and beneficial.

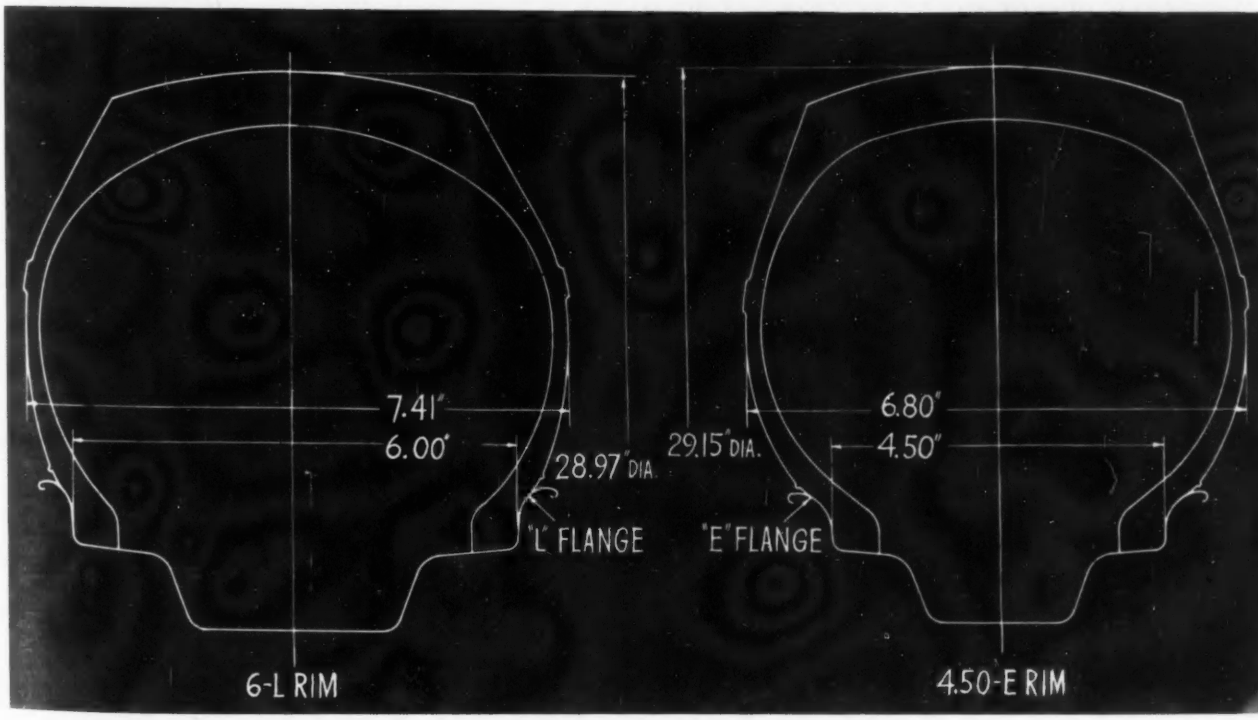
Briefly, let us review the factors in passenger-car tire performance that offer the greatest opportunities for improvement:

First, greater horsepower and increased speed of cars, together with the development of better roads, has made treadwear the outstanding cause of removal of passenger-car tires.

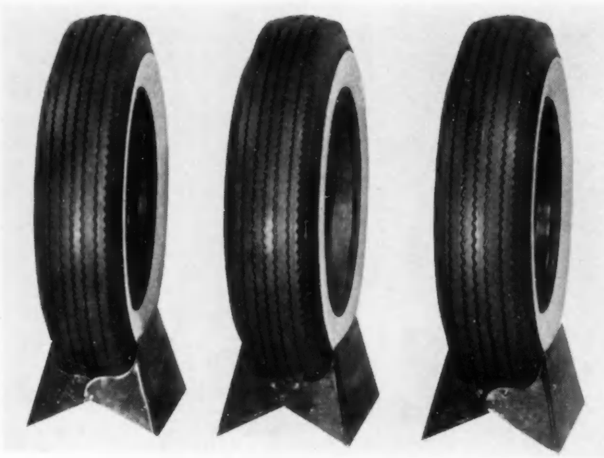
Second, safety at increased speeds demands more car stability, and tires should contribute to the problem of greater safety at high speed.

With these factors in mind it was agreed that a wider foundation for the tire, or a wider rim, offered the greatest possibilities of improvement.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.]



■ Fig. 1 – Cross-section of wide-rim tire compared with that of conventional-rim tire – 6.50-16 size



Tire	6.50-16	6.50-16	7.00-15
Rim	4.50E	6-L	5.00F
Section	6.80 In.	7.41 In.	7.35 In.
Diameter	29.15 In.	28.97 In.	29.34 In.

■ Fig. 2 – Photograph and principal dimensions of (left) 6.50-16 tire mounted on a conventional 4.50E rim; (center) 6.50-16 tire mounted on 6-L wide-base rim; and 7.00-15 tire mounted on 5.00F rim

■ Analysis of Tire Functions

An analysis of the function of a tire is necessary to clarify this problem.

The principal function of a tire is to cushion shocks and provide a flexible contact with the road. However, a typical tire body must have height so that shocks can be cushioned by deflection amounting to 16% to 20% of the tire height. However, this height also gives the tire lateral flexibility which is undesirable because it permits car instability.

The ideal tire, therefore, would have high vertical flexibility and low lateral flexibility, so that a maximum of vertical deflection could be obtained to absorb road irregularities with a minimum of lateral distortion. This would result in better car stability, greater traction, and longer tread life.

Therefore, since the height of the tire could not be lowered without decreasing its carrying capacity and cushioning power, the obvious method was to decrease the lateral flexibility of the tire by giving it a wider foundation or, specifically, a wider rim.

Fig. 1 is a drawing of the conventional tire and a typical wide-rim tire (6.50-16 size) showing the 1½ in. wider rim, or 33% wider foundation.

Another factor of the wide-rim program is the improvement in appearance because of the larger section of the tire on the wide rim.

Fig. 2 shows a photograph and principal dimensions of two identical 6.50-16 tires, one mounted on a conventional 4.50E rim and the other on a 6-L wide rim, together with a 7.00-15 tire of the same design. It will be noted that the dimensions of the smaller tire on the wide rim are practically the same as that of the larger size tire whose weight is 18% greater.

Pressure Reduction on Wide-Rim Tires – As wide rims decreased the deflection of the tire, all tests on durability and car handling were run with 2 lb per sq in. less pressure in the wide-rim tires.

■ Results of Wide-Rim Use

A – Tire Durability

1. Treadwear – The results of over 3,000,000 miles of testing on rims 1½ in. wider than standard show an average improvement of 20% in non-skid tread mileage.

This improvement was found to be greatest under hard driving conditions ranging from 5%, for easy conditions, up to 80% for tests at maximum speed.

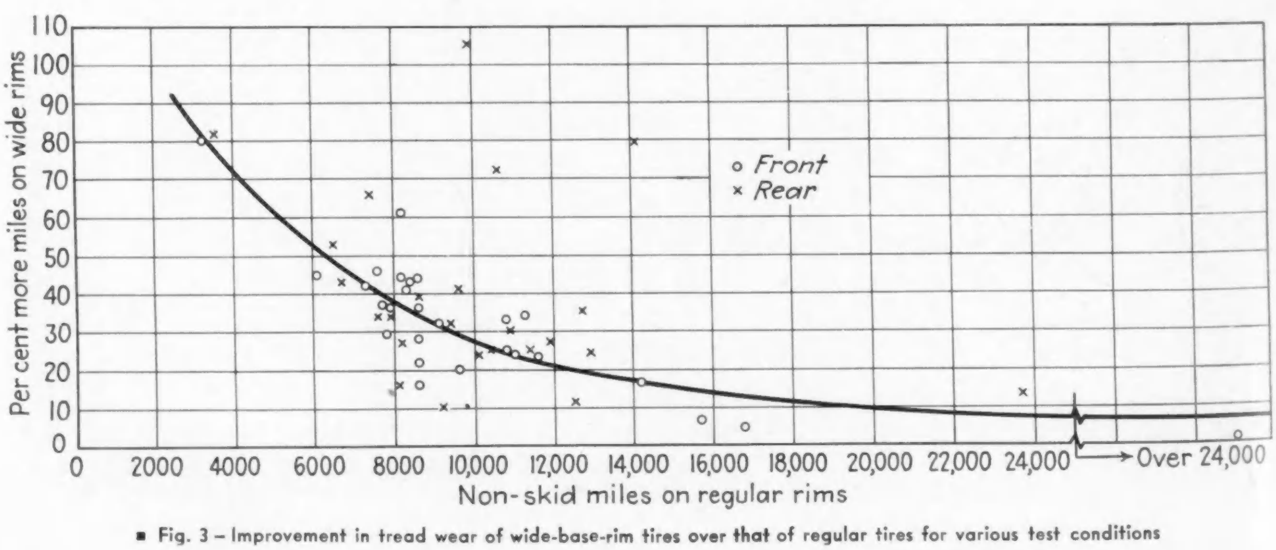
Fig. 3 is a plot showing the effect of the severity of test conditions on improvement in tread life of tires on wide rims versus conventional rims.

2. Uneven Wear, Cracking, and so on – No difference was found in uneven wear, tread cracking, or radial cracking between conventional and wide-rim tires.

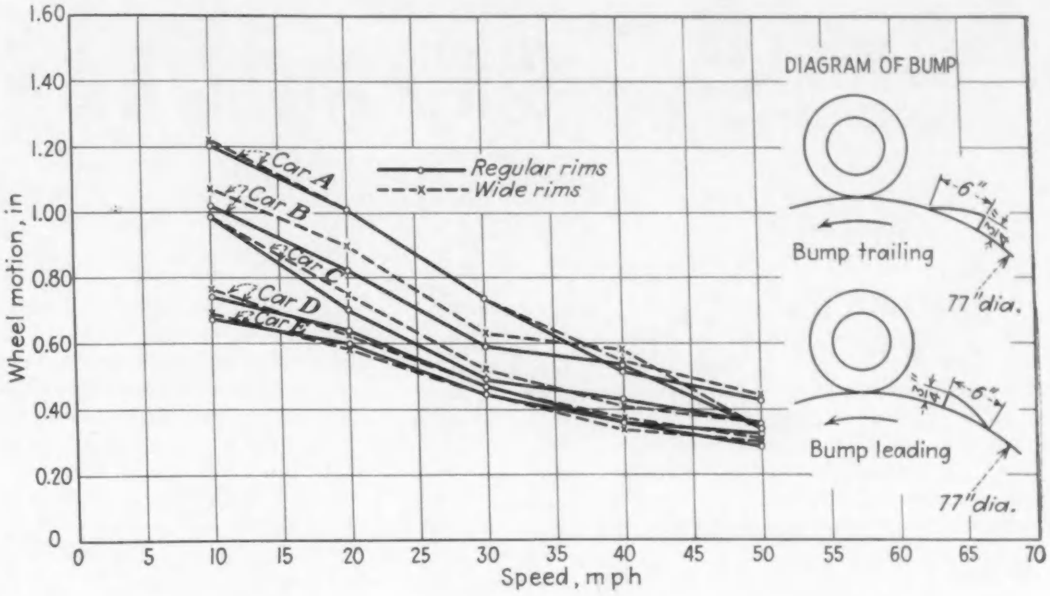
3. High-Speed Separation – Wide-rim tires were equal and satisfactory on both laboratory and road testing at high speed.

4. Fatigue Separation – Wide-rim tires were slightly better than conventional tires on laboratory rivet tests and rough-road tests.

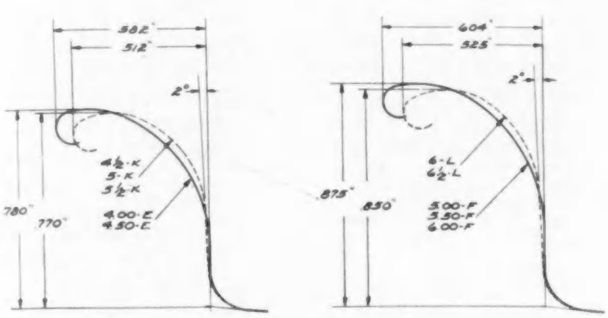
5. Fabric Fatigue Breaks – Fabric life of wide-rim



■ Fig. 3 – Improvement in tread wear of wide-base-rim tires over that of regular tires for various test conditions



■ Fig. 5 – Results of tests comparing riding comfort of regular and wide-rim tires



■ Fig. 4 – Regular versus wide-base rim flanges

tires was equal and satisfactory. Some tests were run at 20% overload and 30% under-inflation. There were fewer impact failures on wide-rim tires than on conventional tires.

6. Beads – Wide rims are a greater strain on beads, and it was found necessary to develop new flanges providing greater support. Fig. 4 is a comparison of the old and new flanges, showing the higher vertical support and narrower width of the new flanges.

7. Curb Blisters – Rim Damage, and so on – It is impossible to blister the sidewall of wide-rim tires and difficult to bruise-break the tire. However, it is easier to damage the rim. An examination of tires on 80 cabs of a taxi fleet in New York, that have used rims equivalent to the full-width wide rim program for several years, shows no complaints of rim or tire damage due to curb-scraping, although some wheels had been used for 200,000 miles.

Our experience on employees' cars also has shown that the driver, after a few experiences, learns to adapt his driving habits toward less abuse from the standpoint of scraping curbs, much in the manner of the user of white sidewall tires.

B. Car Performance

1. Stability – Stability and cornering power increase with rim width and approximately in proportion to increase in rim width. However, actual evaluation of the improvement in stability must necessarily be made by the car engineer on his own particular suspension, tire and rim combination, and so on.

2. Cross-Wind Handling – The effect of the increased stability is very evident in improved cross-wind handling, especially at high speed.

3. Steering and Parking Effort – Slow-speed steering and parking effort is slightly increased with wide-rim tires due to the 2 lb per sq in. reduction in pressure.

4. Traction and Side Skid – Forward traction and braking are improved slightly by wide-rim tires. The improvement in side-skid resistance, however, is considerably better because the tread is held more nearly perpendicular to the road due to the greater inherent stability of the tire.

5. Riding Comfort, Road Harshness, Thump on Expansion Joints, and so on – Riding comfort experience varied considerably with the make and model of car used, but it was found generally that from 2 to 4 lb per sq in. reduction in pressure was necessary in wide-rim tires to produce equivalent ride, equal harshness, thump, and so on.

The only quantitative measurements on riding comfort and harshness were obtained by putting a wedge-shaped bump on a wheel and measuring the axle deflection resulting from hitting this bump. Tests at 2 lb per sq in. differential in pressure on five makes of cars indicated that the wide-rim tires were slightly better when driven into the vertical end of the bump but were slightly worse when dropping off the vertical end. These tests also showed equal wheel motion with wide rims on some cars and slightly greater wheel motion on other cars. Typical results are shown on Fig. 5.

It is also possible that the car engineers can make certain changes in the chassis to improve riding comfort that would not be possible without the greater inherent stability offered by wide-rim tires.

In conclusion, let me express my appreciation for the cooperation already given us by the car engineers in working out this new type of tire construction and express a wish for its continuation in the future.

CRANKCASE OILS

VISCOSITY at starting temperature as well as at operating temperature, pour point, stability or oxidation resistance, and prevention of engine deposits are factors of primary importance in evaluating crankcase lubricants for use under heavy-duty service conditions, Mr. Wolf points out.

As factors controlling the suitability of a lubricant for a specific purpose, he names crude source, refining process, oxidation inhibitor, and detergent or suspension agent. He contends that the heavy-duty lubricants satisfactory for use under all service conditions must contain a liberal amount of both inhibitor and detergent.

This paper presents the results of road and dynamometer tests made to evaluate crankcase oils; appraises the effect of operating conditions; and reviews the causes and cures of failure of copper-lead bearings in heavy-duty service.

■ ■ ■

A PRIVATELY owned and operated passenger car averages from 8,000 to 15,000 miles per year. A heavy-duty commercial vehicle, in intercity truck or coach service, averages from 7,000 to 22,000 miles per month. Consequently, fuel and engine oil costs, which are a relatively small per cent of the total operating expense for a passenger car, become major items in the operating expense in heavy-duty commercial service.

Both the individual passenger-car owner and the commercial operator are interested in and benefit by any new developments in engine lubrication that insure freedom from engine deposits, maintain maximum engine performance over a longer period of time, and increase the overall operating efficiency.

However, since fuel is the major cost item in heavy-duty operations, the commercial operator is interested, in addition, in any changes or improvements in engine design that will result in higher fuel economy. The automotive designer is restricted by space and weight limitation and, while minor improvements may be made by reducing engine friction, any major gain in specific power output and fuel economy must be made by burning more effectively a larger amount of fuel in a smaller combustion chamber. Consequently, any gain in fuel economy obtained by increasing the compression ratio or by employing the diesel cycle results in a higher mean effective combustion temperature level and increases the stress on the engine lubricating oil.

In heavy-duty truck, coach, marine, and stationary installations the modern diesel engine shows an improvement of approximately 40% in fuel economy over a gasoline engine of comparable size, weight, and power output. This is a substantial saving and alone justifies the extensive research and development work which is being carried on by the automotive and petroleum industries on the production of heavy-duty crankcase oils.

■ Suggested Nomenclature

Considerable attention has been given to the nomenclature of this newer class of crankcase oils. Many designations, such as "Truck and Bus Oil," "Diesel Engine Oil," and so on, have been suggested. These terms do not appear to be as desirable as "Heavy-Duty" since they may greatly limit the use of these oils. For example, a passenger-car owner, operating under particularly severe conditions at high speed for long periods of time, may hesitate to use a "Truck and Bus Oil"; likewise, an operator of heavy-duty gasoline equipment may hesitate to use a "Diesel Engine Oil."

Dynamometer and actual field tests have demonstrated that the heavy-duty oils are absolutely necessary for satisfactory operation under the most severe service conditions in both gasoline and diesel engines. Practically all engines operate at least a portion of the time under the more severe service conditions. During these periods, more satisfactory operation will be obtained with the heavy-duty oils, and under the mild or moderate service conditions, the heavy-duty oils will be as satisfactory in performance characteristics as the conventional crankcase oil. Hence, the use of heavy-duty oils should be encouraged even though they may not be required under all operating conditions.

The designation "Heavy-Duty" is thoroughly descriptive of these newer engine crankcase oils and will extend their use in all types of engines and under all service conditions where the conventional crankcase oils are inadequate, rather than limit their use to specific operations.

■ Refinery Inspection Tests

The routine refinery inspection tests, outlined in Table 1, give the petroleum technologist a considerable amount of information regarding an oil and serve as an indispensable guide in refinery operations. In the older models of passenger-car engines these inspection tests also predicated in a more or less general way the performance of an oil under service conditions. With the introduction of the newer methods of subtractive refining, these control or inspection tests no longer serve as identification tests or correlate with service performance in the heavy-duty, high specific power output, modern automotive engines.

In the older engines oil consumption and ease of starting

[This paper was presented at the National Fuels and Lubricants Meeting of the Society, Tulsa, Okla., Nov. 8, 1940.]

for HEAVY-DUTY Service

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were the primary distinguishing differences between oils from different crude bases and processed by different refinery methods. All of these oils formed sludge and all oxidized under severe service conditions but, since only relatively few engines were operated under the more severe conditions, very little trouble was experienced in the field.

In the modern engine, viscosity and pour point remain as important factors in the selection of a satisfactory lubricant. The maximum viscosity at the cranking temperature and the minimum viscosity at the operating temperature, consistent with other properties, determine the ease of starting and oil consumption. A pour point below the cranking temperature is required to insure oil circulation at low temperatures. To these must be added two new requirements, stability or resistance to oxidation and the prevention of engine deposits. Consequently, in evaluating crankcase lubricants for the present-day engines, especially for use under heavy-duty service conditions, the requirements outlined in Table 2 become of prime importance.

New Requirements

Unfortunately, from the standpoint of the laboratory technologist, the less complex inspection tests (Table 1) do not give a clue to the stability or resistance to oxidation, or to the ability of the lubricant to prevent engine deposits. Changes in refinery methods and the use of inhibitors or anti-oxidants have altered the oxidation characteristics of oils to such an extent that products having the same inspection limits may perform in a radically different manner in service. This point was first recognized several years ago, when copper-lead and cadmium-alloy bearing corrosion was experienced with solvent-refined oils. Some of the objectionable sludge-forming materials were removed by solvent refining, but the natural oxidation inhibitors were also removed. Under mild service conditions these oils showed a definite reduction in engine deposits but, at the higher temperatures developed under more severe driving conditions, oxidized rapidly and formed large quantities of acidic compounds. This condition was quickly recognized by the refiners and inhibitors were added to overcome the bearing corrosion troubles that developed in the field. This same point is also illustrated by the use of detergents in diesel lubricating oils. Many of the early detergent compounds were metallic soaps and, under service conditions, acted as catalysts or oxidation promoters. Crankcase oils containing this type of detergent function satisfactorily only if used with non-corrodible bearings, and if drained frequently enough to prevent the accumulation of oxidation products beyond the suspension or dispersion ability of the detergent compound.

Factors Controlling Suitability

In order to meet the more exacting requirements of the present-day engines, especially those operated under heavy-duty service conditions, the refiner must carefully consider the several factors listed in Table 3. This statement does not imply that only a single type of crude or single refining process will produce a satisfactory finished product, but it does infer that the necessary balance must be maintained and that the addition agents are compatible and effectively control oxidation and engine deposits.

The importance of each of these factors depends upon

Table 1 – Routine Refinery Inspection Tests

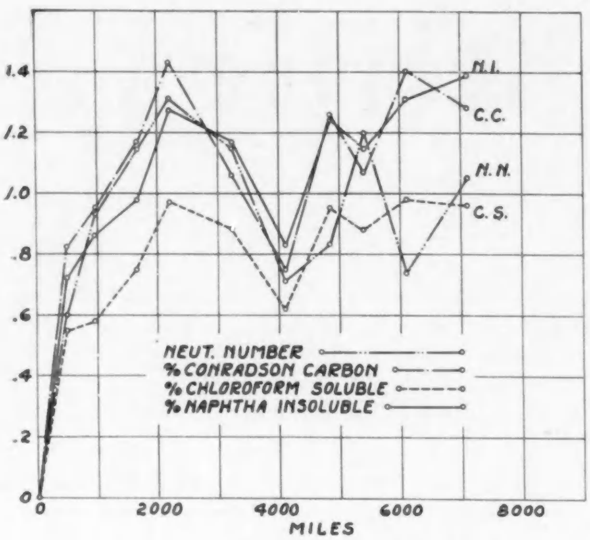
1. Gravity
2. Flash point
3. Fire Point
4. Viscosity
5. Viscosity Index
6. Color
7. Carbon Residue
8. Pour Point
9. Neutralization Number
10. Saponification Number
11. Precipitation Number
12. Emulsion Test

Table 2 – Present Standards for Evaluating Crankcase Lubricants

1. Viscosity,
   a. At starting temperature
   b. At operating temperature
2. Pour Point
3. Stability, or oxidation resistance
4. Prevention of engine deposits

Table 3 – Factors Controlling Suitability of Lubricant for a Specific Purpose

1. Crude Source
2. Refining Process
3. Oxidation Inhibitor
4. Detergent or Suspension Agent



■ Fig. 1 – Analysis of low-viscosity-index 10-W oil taken from crankcase of car in road test

the specific purpose for which the lubricant is intended, and the actual field operating conditions under which the lubricant is to be used.

For use under light or moderate service conditions and at low crankcase oil temperatures, the crude source and the refining process have less influence on performance but, under heavy-duty service conditions, these factors become of major importance.

In the high-speed heavy-duty diesel engine an oxidation inhibitor or anti-oxidant and a suitable detergent or dispersion agent are required. The oxidation inhibitor is needed to retard oxidation of the base oil in the crankcase; the detergent, to control contamination of the lubricant by products of incomplete fuel combustion and decomposition of excess lubricating oil in the combustion chamber, and to prevent the formation of deposits that may interfere with engine operation.

The heavy-duty gasoline truck and coach engine also requires a somewhat similar balance between inhibitor and detergent.

In passenger-car engines operated under average owner-driver conditions, an oxidation inhibitor is far more important than a detergent. However, in passenger car engines operated under extreme high-speed driving conditions, some degree of detergency may be required to minimize engine deposits and to maintain new car performance over a longer period of time.

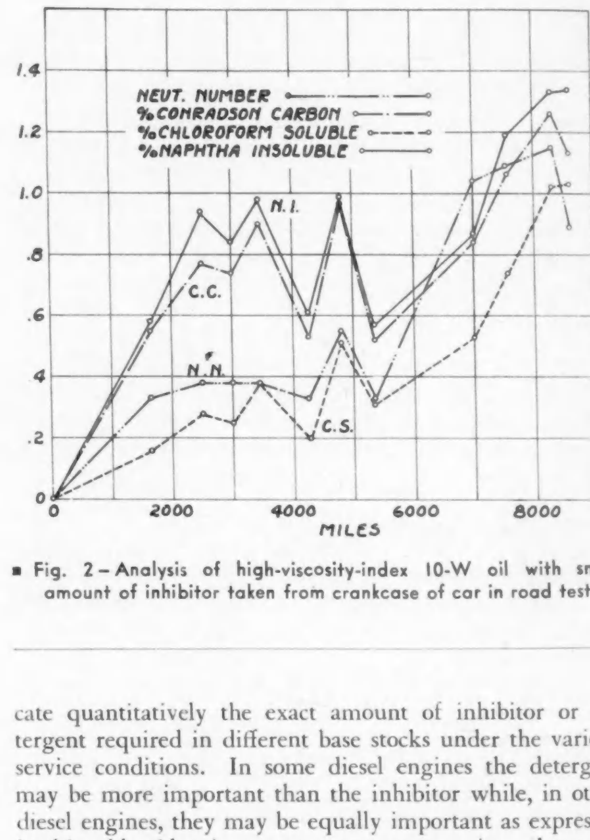
Table 4 – Relative Importance of Oxidation Inhibitors and Detergents in Different Types of Service

	Inhibitor	Detergent
Diesel Engine	XXX	XXX
Gasoline Engine		
Truck and Coach	XXX	XX
Passenger Car		
Severe Service	XXX	X
Average Service	XX	....
Universal Lubricant	XXX	XXX

Table 5 – Operating Conditions Under Which More Frequent Service Complaints Occur

Valve Sticking	– City Driving
Bearing Corrosion	– High Speed Driving
Sludge	
Water	– Low Temperature Winter Operation
Oxidation	– High Temperature Operation (may result from moderate as well as high-speed driving)

The relative importance of oxidation inhibitors and detergents in different types of service is expressed in Table 4. This evaluation is qualitative and is not intended to indicate quantitatively the exact amount of inhibitor or detergent required in different base stocks under the various service conditions. In some diesel engines the detergent may be more important than the inhibitor while, in other diesel engines, they may be equally important as expressed in this table. Also, in some passenger-car engines, detergent may be important even under average service conditions. Regardless of the relative importance of oxidation inhibitors and detergents in the different types of service, the heavy-duty lubricants satisfactory for use under all service conditions must contain a rather liberal amount of both inhibitor and detergent.

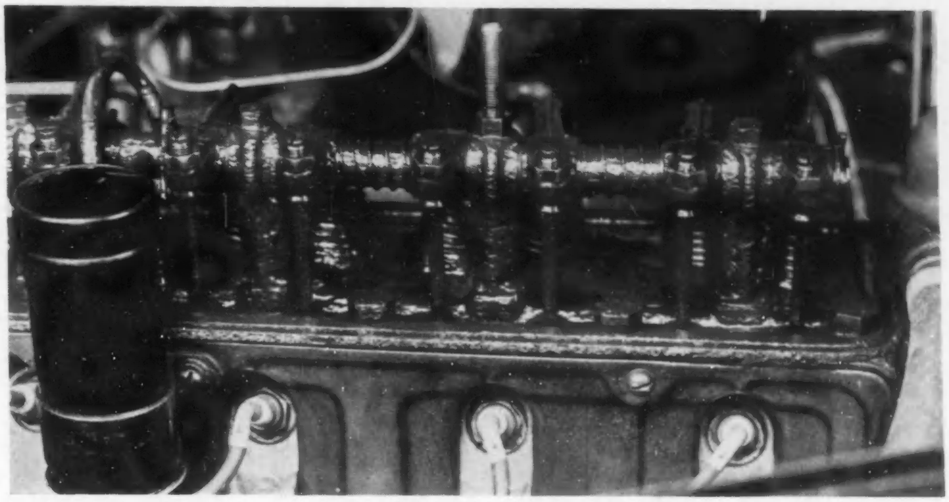


■ Fig. 2 – Analysis of high-viscosity-index 10-W oil with small amount of inhibitor taken from crankcase of car in road test

The light or low-viscosity oils are more subject to oxidation than the more-viscous oils. Consequently, it may be necessary to increase the quantity of the addition agents in the lighter grades in order to maintain the proper balance in oxidation resistance and freedom from engine deposits in all viscosity grades.

The heavy-duty oils will not solve the "water-sludge" problem. The accumulation of water in the crankcase during the winter months in passenger cars driven in in-

▪ Fig. 4 – Precipitation of oxidation products on addition of fresh oil – Deposit on rocker-arm mechanism was formed in 250 miles



termittent city service, and in light commercial cars in door-to-door delivery service, is a low-temperature rather than a heavy-duty problem.

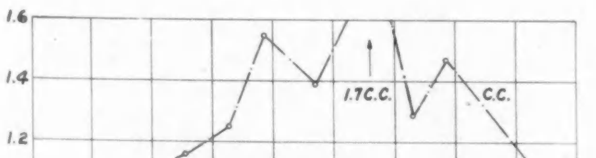
▪ Correlation with Service

The quantitative effects of many of the factors in service that contribute to the deterioration of crankcase lubricants are either not known or recognized. Therefore, until these factors are evaluated and satisfactory laboratory tests are evolved, service tests under actual road conditions must be retained as the final criterion for judging the suitability of a crankcase lubricant.

No one set of service conditions can give a complete evaluation. Most cars operate under all conditions at least a part of the time, and therefore service or road tests must be based on an analysis of service complaints. A partial analysis of the conditions under which the more frequent complaints occur is given in Table 5.

In this connection, attention should be called to the fact that tests designed to determine performance under one set of operating conditions may give misleading results if other operating conditions are permitted to influence the test. For example, a test under city-driving conditions to determine valve-sticking tendencies may be completely upset by intermittent operation at high speed.

Multicylinder engine dynamometer tests operated under any set of conditions (speed, horsepower, output, and temperature) approximate some operation that occurs at some time in service. However, since different service conditions influence oil deterioration differently, it follows that engine dynamometer tests must be made under the several service conditions that have the greatest effect on decomposition and contamination of the crankcase lubricant.



▪ Fig. 3 – Analysis of high-viscosity-index SAE 30 oil taken from crankcase of car in road test

▪ Road Tests

Before discussing specific dynamometer operating conditions for evaluating heavy-duty crankcase lubricants, the performance of several representative oils under service conditions should be considered.

Figs. 1, 2, and 3 give the used oil analyses on three cars operated on the road under comparable driving conditions, over a period of approximately 8000 miles without oil change. The crankcase oil reported in Fig. 1 is a low-viscosity-index 10-W oil. Rapid oxidation occurred during the first 2000 miles. The naphtha insoluble was practically all chloroform soluble.

The crankcase oil used in the second car, reported in Fig. 2, is a high-viscosity-index 10-W oil protected by a small amount of inhibitor. The neutralization number remained quite low during the first 2000 miles as contrasted with the values shown in Fig. 1. The ratio of chloroform soluble to naphtha insoluble also held at a much lower value for the first half of the test. This oil did not, however, contain sufficient inhibitor to control oxidation over a long period of time and all values reached practically the same end point as in the uninhibited low-viscosity-index 10-W oil reported in Fig. 1.

Fig. 3 gives the used oil analysis for the third car, which was operated on an SAE 30 high-viscosity-index oil. With this oil the neutralization number and the ratio of chloroform soluble to naphtha insoluble remained at relatively low values throughout the entire test period.

▪ Resinous Oxidation Products

The precipitation of resinous or varnish-forming oxidation products on the addition of fresh oil is responsible for many present-day service complaints. Road or dyna-

Table 6 – Engine Dynamometer Operating Conditions

	Chevrolet	GM "71" Diesel (3-, 4-, or 6-cyl engine)
Speed	3150 rpm (60 mph)	2000 rpm
Load	30 to 35 hp	27 hp/cyl
Piston head temperature	500 F	.....
Crankcase oil temperature	280 F	230 F
Jacket water outlet temperature	200 F	180 F
Jacket water inlet temperature	.....	12 to 15 F lower than outlet
Fuel temperature	room	120 F
Intake air temperature	room	105 F
Oil consumption	1 qt/500 miles	maximum 0.06 lb/hr/cyl
Exhaust back pressure	nominal	6 in. hg
Length of test	4000 miles (67 hr)	500 hr

mometer tests should therefore be operated under conditions which will produce precipitation deposits with crankcase oils that form this type of deposit under service conditions. Fig. 4 illustrates one phase of this problem. After approximately 4000 miles of operation on a high V.I. high-grade commercial engine oil, the engine was inspected and the valve lash was adjusted. The valve mechanism on this engine was quite clean. After an additional week in service, during which time the car was driven on a 250-mile high-speed run with 2 qt of make-up oil added, the valve mechanism was in the condition shown in Fig. 4.

■ Dynamometer Tests

It has been pointed out that no one set of service conditions can give a complete evaluation of a crankcase oil. Likewise, no one set of dynamometer test conditions can correlate with service. However, since dynamometer tests cannot be conducted to approximate all service conditions, it is necessary to select a set of operating conditions that will approach the conditions on the road under which operating difficulties are most likely to be encountered. An outline of the operating conditions employed in the Chevrolet and GM "71" Diesel Engine Dynamometer Tests is given in Table 6. Attention should be directed to the importance of maintaining the engine in the proper mechanical condition in order to approximate closely the specified oil consumption. This is very important in order to determine the effect of the addition of fresh oil on the precipitation of resinous or varnish-forming products and the formation of coffee-grounds carbon.

Table 7 gives the analysis of the used oil from several Chevrolet engine tests made under the conditions outlined in Table 6, and illustrates the effect of inhibitors and detergents on used oil analyses and the condition of engine parts.

It is interesting to note that, while the engine run on lubricant *B* (a highly solvent refined 10-W oil) was much cleaner than the engine run on lubricant *A* (a medium quality 20-W oil), the pistons in both engines were stuck at the end of 2000 miles. The varnish deposited from the medium-quality oil was quite dark in color while the varnish from the highly refined oil was transparent.

The used oil analyses, without any knowledge of composition of the several lubricants, might be interpreted as predicting a much cleaner engine with lubricant *C* (oil *B* plus an oxidation inhibitor) and a somewhat dirtier engine with lubricant *D* (oil *B* plus a metallic soap detergent) than with lubricant *B*. As a matter of fact, the engines operating on lubricants *C* and *D* were extremely clean and free from varnish on the piston skirts. The metallic soap in lubricant *D* had some catalytic effect on the oxidation of the base oil but prevented the deterioration products from forming engine deposits.

Lubricant *E* (an oil similar to *B* plus both oxidation inhibitor and detergent) gave both a good used oil analysis and a clean engine free from varnish on the piston skirts.

Dynamometer tests made under the conditions outlined in Table 6 evaluate crankcase lubricants in the same general order as road tests but, due to the greater variety of operating conditions encountered on the road, the used oil analyses from the dynamometer tests do not correlate perfectly with the used oil analyses from road test cars.

Table 7 – Used Oil Analysis on Chevrolet Dynamometer Tests

Lubricant	A	B	C	D	E
Type	20-W Medium Quality	10-W Highly Refined	B Plus Inhibitor	B Plus Detergent	Plus Inhibitor and Detergent
Miles	2000	2000	4000	3000	4000
Neutralization No.	0.66	7.70	0.46	9.70	0.22
Naphtha Insoluble	1.07	0.99	0.48	3.41	0.38
Chloroform Soluble	0.87	0.45	0.11	1.75	0.08
Viscosity Increase at 100 F	....	174	12	334	12
Engine Condition	Very Dirty	Fair	Clean	Clean	Clean
Pistons	Stuck	Stuck	Free	Free	Free
Varnish	Dark	Transparent	Nil	Nil	Nil

On the dynamometer, the neutralization number is generally higher and the naphtha insoluble and chloroform soluble are generally lower than obtained on the same oil on the road when operated for approximately 4000 miles without crankcase oil changes. The Conradson carbon residue values generally check very closely. The ash values are usually higher on the road.

This difference in used oil analyses, between dynamometer and road tests, is illustrated in Table 8. Lubricant from the same source was used in both engines, and the tests were of approximately the same duration.

The heavy-duty diesel engine operates under more severe oxidation conditions than does the heavy-duty gasoline engine. In addition, the crankcase oil in a diesel engine is further contaminated by the products of incomplete fuel combustion and decomposition of excess lubricating oil in the combustion chamber. Consequently, engine tests to determine the suitability of a crankcase lubricant in heavy-duty diesel service must be made under laboratory conditions that correlate with field service.

Table 8 – Used Oil Analyses on Same Oil from Dynamometer and Road Tests

	Dynamometer	Road
Miles	4000	4770
Neutralization No.	4.38	0.83
Naphtha Insoluble	0.94	2.11
Chloroform Soluble	0.18	0.38
Conradson Carbon	2.61	2.65
Ash	0.50	0.65

Experience in the field has shown that crankcase lubricants that will not successfully pass the 500-hr test outlined in Table 6, will not perform satisfactorily under the more severe service conditions under which many diesel engines are operated. Experience in the field also has shown that crankcase lubricants that are outstanding in heavy-duty diesel service, are likewise outstanding in heavy-duty gasoline engine truck and coach service.

The used oil analyses on six oils run in the GM "71" diesel engine in substantial accord with the conditions outlined in Table 6 are given in Table 9. The tests on oils *F* and *G* were stopped at 116 and 91 hr respectively, due to excessive bearing corrosion and engine deposits.

Oil *H* did not oxidize as badly as oils *F* or *G* but the engine failed at 96 hr because of excessive piston deposits. In the tests with oils *I*, *J*, and *K*, the neutralization number remained at relatively low values, the copper-lead bearings were not corroded, and the engines were clean and free from injurious piston deposits. In the test on oil *K* the engine was operated for the first 144 hr without a sludge filter. This accounts for the high naphtha insoluble reported on the 96-hr sample.

■ Effect of Operating Conditions

Analysis of the available data on crankcase oil deterioration indicates that oxidation and decomposition occur both in the combustion chamber and in the crankcase. The severity of the oxidation conditions in these two locations depends on engine design and operating conditions.

The effect of some of the differences in operating conditions on the used oil analysis is well illustrated by the data given in Fig. 5. The engine operating conditions are given in Table 10.

The operating conditions in Run No. 2 are not ordinarily considered as severe as in Run No. 1. However, the used oil analysis, Fig. 5 (2), shows more rapid crankcase oil deterioration than in Run No. 1, Fig. 5 (1). It is apparent in Run No. 1 that most of the oil that works past the pistons is burned in the combustion chamber and that the major portion of the oxidation of the crankcase oil occurs in the crankcase. However, in Run No. 2, the engine was motored, without firing, at high speed with the throttle closed and a large portion of the oil thrown onto the cylinder walls was partially oxidized in the combustion chamber and returned to the crankcase, further contaminating the crankcase oil.

Run No. 3 represents operating conditions where practically the entire oxidation process occurred in the crankcase. Run No. 4, employing the same operating conditions as Run No. 3, illustrates the effect of oxidation inhibitors in controlling or preventing oxidation.

■ Bearing Failures

During the past year attention has been focused on the failure of copper-lead bearings in heavy-duty service. These failures may be classified as loss of lead due to the corrosive effect of the lubricant, and mechanical failures due to fatigue. Corrosion and fatigue are greatly accelerated by an increase in bearing pressure; consequently, tests made in full-size multicylinder engines more closely approximate service conditions than miniature single-cylinder engine tests or laboratory bearing corrosion tests.

Loss of lead may be caused by the solvent action of acidic products formed on oxidation of the base oil. The

Table 9 – Analysis of Oils Run in GM "71" Diesel Engine

Oil	Hours	Neutralization No.	Naphtha Insoluble	Chloroform Soluble
F	8	0.05	0.17	0.06
	55	0.82	0.34	0.16
	116	2.14	3.06	2.26
G	25	0.64	0.45	0.24
	68	2.58	3.18	2.83
	91	3.44	6.98	6.24
H	24	0.11	0.33	0.07
	72	0.38	1.07	0.40
	96	0.43	3.04	1.60
I	100	0.16	0.19	0.05
	205	0.26	0.18	0.03
	305	0.42	0.52	0.1[illegible]
	501	0.54	0.11	0.03
	757	0.59	0.16	0.05
J	97	0.16	0.30	0.08
	206	0.22	0.38	0.05
	353	0.16	0.33	0.08
	530	0.21	0.44	0.09
K	96	0.11	1.02	0.54
	192	0.16	0.10	0.05
	312	0.16	0.09	0.04
	408	0.22	0.12	0.05
	500	0.22	0.28	0.13

Effect of operating conditions on used oil analysis

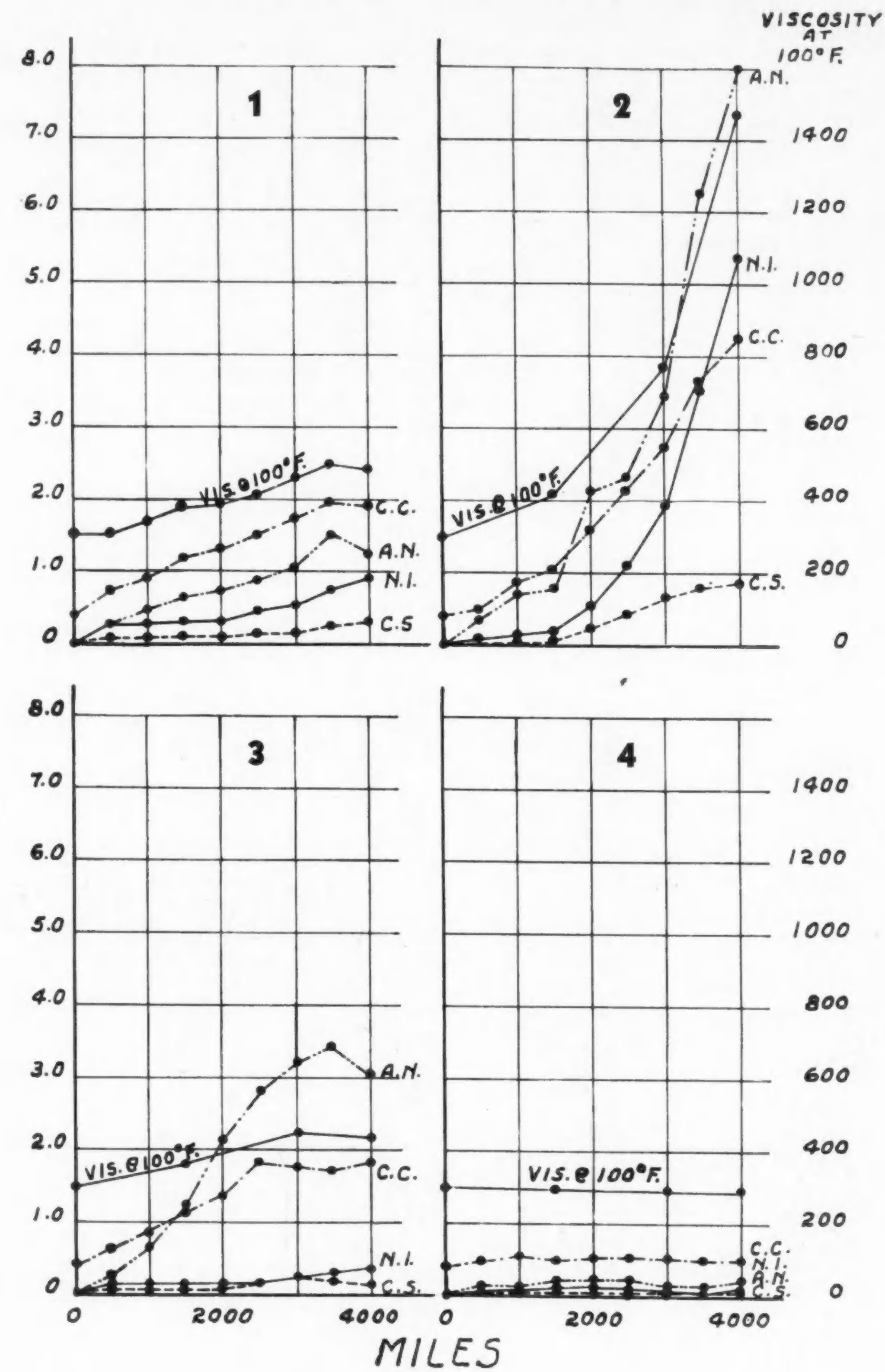
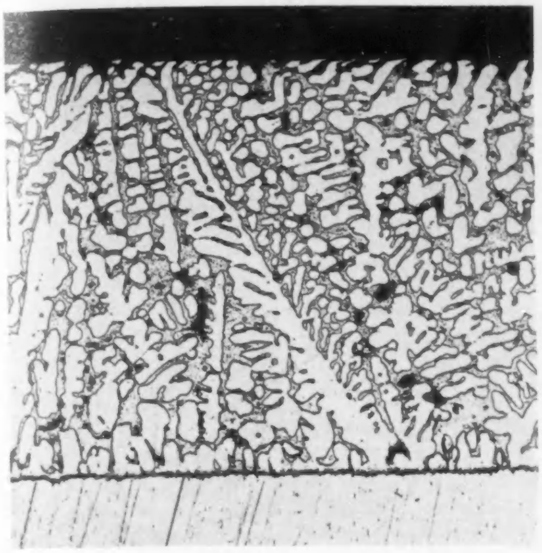
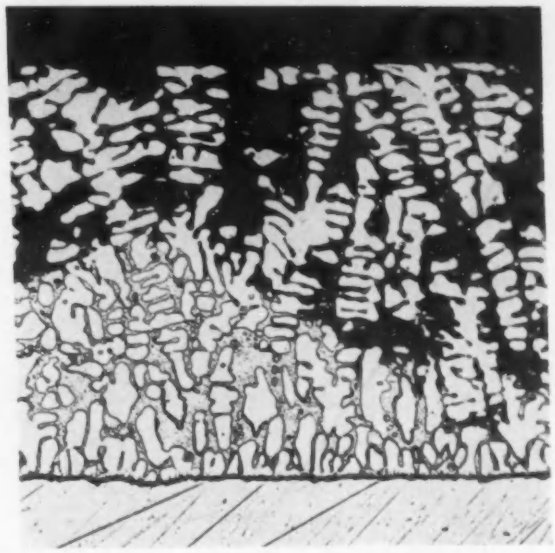


Fig. 5



■ Fig. 6 – Section from unloaded area (X 100)



■ Fig. 7 – Section from loaded area (X 100)

■ Figs. 6 and 7 – Sections from same connecting-rod bearing shell from 500-hr diesel engine test – The loss of lead in Fig. 7 is due to action of detergent compound

oxidation of the base oil may be greatly accelerated by the catalytic effect of metallic soaps used as detergents.

The loss of lead may also be caused by the action of metallic soaps under pressure in the heavily loaded bearing areas. This is a relatively new phase of the bearing corrosion problem.

Figs. 6 and 7 show the microstructure at 100 diameters of the upper half of a connecting-rod bearing shell from a 500-hr diesel engine test. These sections were prepared for micro examination by a special polishing process[1]. The copper appears as a bright constituent, the lead as a gray constituent. The areas in which the lead has been removed by the corrosive action of the acidic oxidation products or by the action of the detergent compound under pressure appear black in the microphotographs.

Fig. 6 is typical of the unloaded and moderately loaded areas of the bearing. Fig. 7, showing loss of lead extending in one section to the steel backing, is typical of the highly loaded area of the same bearing. The neutralization number of the crankcase oil at 500 hr was 0.11. The maximum neutralization number during the test was 0.21.

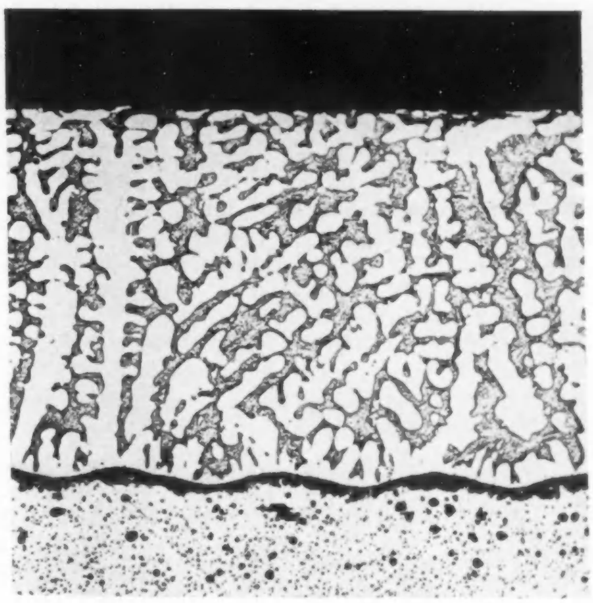
The corrosion shown in Fig. 7 was due entirely to the action of the particular metallic soap or detergent compound used in this lubricant. Tests made with other lubricants, inhibited with similar anti-oxidants but without the metallic soap detergent, showed no loss in lead from the bearing surface.

Metallic soaps, such as calcium naphthenate, aluminum stearate, and so on, appear to decompose under the pressure and temperature conditions existing in the high-pressure areas of heavily loaded bearings, yielding free acids. The free acids, so formed, attack the lead in copper-lead bearings in the same manner as the acidic products formed on oxidation of the base oil.

Metallic compounds other than metallic soaps, which either do not break down under the conditions existing in the high-pressure bearing areas or do not form free acids or acidic compounds on decomposition, are available as detergent compounds. This type of metallic detergent compound does not cause bearing corrosion.

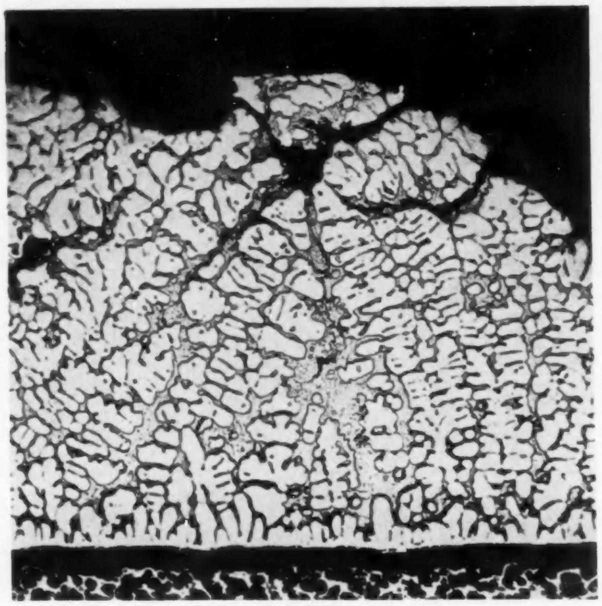
Fig. 8 shows a section from the loaded area of a connecting-rod bearing at the completion of a 500-hr GM "71" diesel engine test. The oil in this test contained both an inhibitor and a detergent. This particular detergent compound was not catalytic and did not decompose under the conditions existing in the high-pressure bearing area. No loss of lead occurred during the 500-hr test. The original copper-lead structure extends to the bearing surface.

Copper-lead bearing failures due to fatigue can be distinguished readily from failures due to loss of lead. In

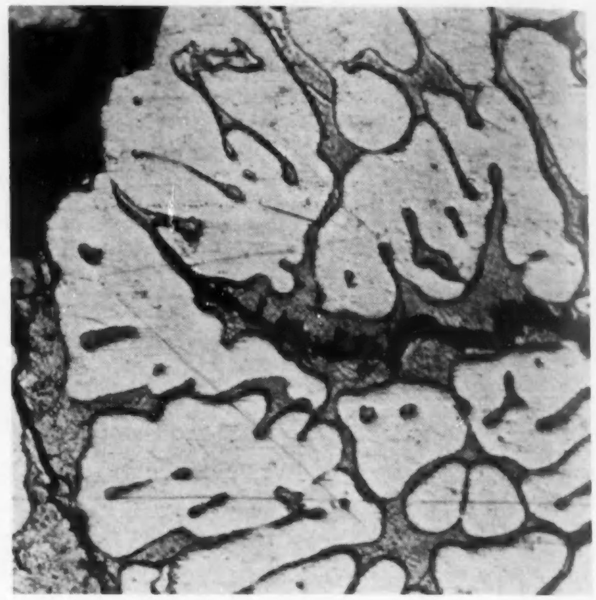


■ Fig. 8 – Copper-lead bearing shell from 500-hr test – High-pressure area – No loss of lead (X 100) – Oil contains inhibitor and detergent

[1] See *Metal Progress*, Vol. 30, No. 5, November, 1940, pp. 674-676: "Metallographic Preparation of Copper-Lead Bearings," by H. L. Grange.

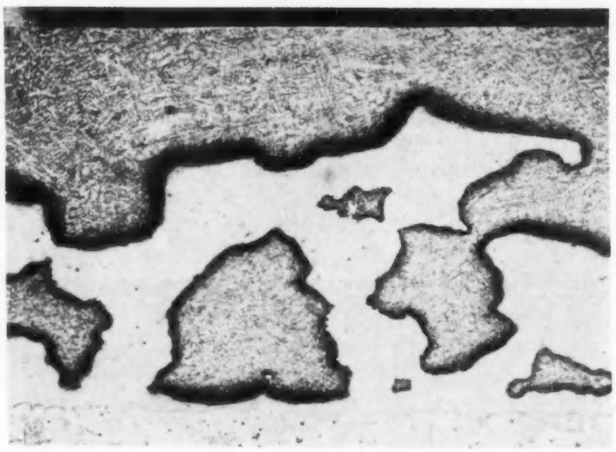


■ Fig. 9 – Fatigue failure – Note that lead extends to the edge of the fatigue crack (X 100)



■ Fig. 10 – Fatigue failure – Note the lead adhering to the copper along the path of the fatigue crack (X 500)

fatigue failures the lead in the copper-lead structure extends to the surface of the fatigue crack. This condition is illustrated in the micrograph at 100 diameters shown in



■ Fig. 11 – Buick-type bearing – Hard matrix with high-lead babbitt bearing surface (X 100)

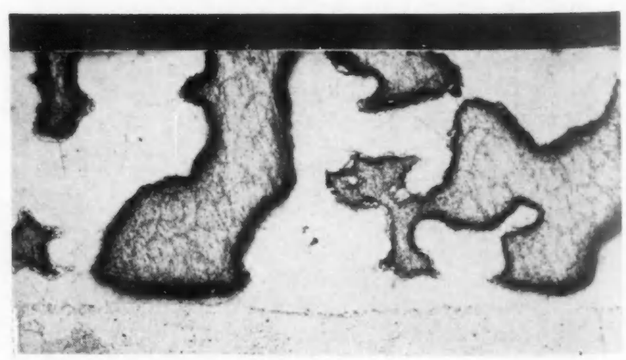
Fig. 9. In the 500-diameter micrograph shown in Fig. 10, lead can be seen adhering to the copper along the path of the fatigue crack.

■ Solution of Heavy-Duty Problem

The stress on the crankcase oil under heavy-duty operating conditions can be lowered by reducing the crankcase oil temperature. This, however, is only a temporary solution of the problem. The trend in engine design is definitely toward higher specific output. Therefore, in any present design where temporary relief is obtained by lowering crankcase oil temperature, the designer immediately takes advantage of this reduction in temperature and increases specific power output.

Bearing manufacturers are making considerable progress in the development of corrosion-resistant bearings. A recent development, shown in Fig. 11, is now in production on Buick 1941 main crankshaft bearings. This bearing is similar in structure to the conventional copper-lead bearing. It has a hard strong matrix filled with a high-lead corrosion-resistant babbitt. If the matrix is completely covered with a thin layer of babbitt, Fig. 11, the bearing may be considered as an extremely thin babbitt bearing possessing all of the frictional characteristics of babbitt and many of the structural characteristics of a copper-lead bearing.

For extremely heavy-duty service where higher fatigue properties are desired, the bearing surface may be lowered, as shown in Fig. 12, to expose the hard matrix. This type of bearing is being tested under heavy-duty operating conditions. If the service tests now in progress indicate that this type of bearing is satisfactory in fatigue resistance and bearing properties and if it can be produced commercially, it will solve the bearing corrosion problem. However, this statement does not mean that the petroleum industry could then furnish crankcase



■ Fig. 12 – Experimental heavy-duty bearing – Hard matrix exposed on bearing surface (X 100)

Table 10 – Engine Operating Conditions for Tests Reported in Fig. 5

Run No.	Crankcase Oil, F	Jacket Water, F	Speed, rpm	Temperature of Underside Piston Crown, F	Operating Conditions
1	280	200	3150	500-510	As outlined in Table 6. Load to maintain piston temperature, approximately 25 hp.
2	280	200	3150	300-315	Engine motored, gasoline line disconnected, throttle closed. High combustion-chamber temperature due to heat of compression.
3	280	200	3150	275-280	Same as Run No. 2, except exhaust valves cut off to avoid compression of air in combustion chamber.
4	280	200	3150	275-280	Same as Run No. 3, except 0.50% inhibitor X and 0.50% inhibitor Z added to crankcase oil.

oils that oxidize in service. While this new type of bearing will not be affected by acidic compounds, the oxidation products that result in resinous or varnish-forming materials, coffee grounds, and other deleterious decomposition products, will continue to be objectionable under heavy-duty service conditions.

The further development of heavy-duty lubricants containing suitable inhibitors or anti-oxidants and detergent compounds, remains as the only permanent and satisfactory solution of the problem.

Diesel Engines for the Navy

THE layman, in following the progress of the war, observed the working of the military axiom of surprise. There is no need to go into examples as to tactics here. It shall be simply cited that the counter is eternal vigilance and instant readiness. A man of war must be ready when the need arises. The engineering plants must be capable of quickly being applied to the screws and to supply any demand from the batteries and those auxiliaries essential for battle. In searching for systems and devices which are capable of satisfying this demand, the sales argument for the diesel engine is attractive. It is capable of quick starting – of quickly developing its full rating. It can run with a smokeless exhaust; it consumes a fuel of low volatility; and it is the most efficient power generator now available. The suggestion of utilizing the diesel engine in heavy ships for propulsion has appeared in the literature. The most ambitious step in this direction was in the German "Deutschland" class to which belonged the "Graf Spee" of Montevideo fame.

The United States Navy has availed itself of these advantages not only in submarines but for propulsion in tugs, auxiliary and patrol craft and in motor boats. Further, as the diesel grows, so will its application in the main propulsion field. For auxiliary purposes, the quickly available feature is of use for stand-by units, while at the time its dependability has been sufficiently recognized to assign it to ships service duty. As to the types and kinds of engines in the naval service, the Navy has used about all those known.

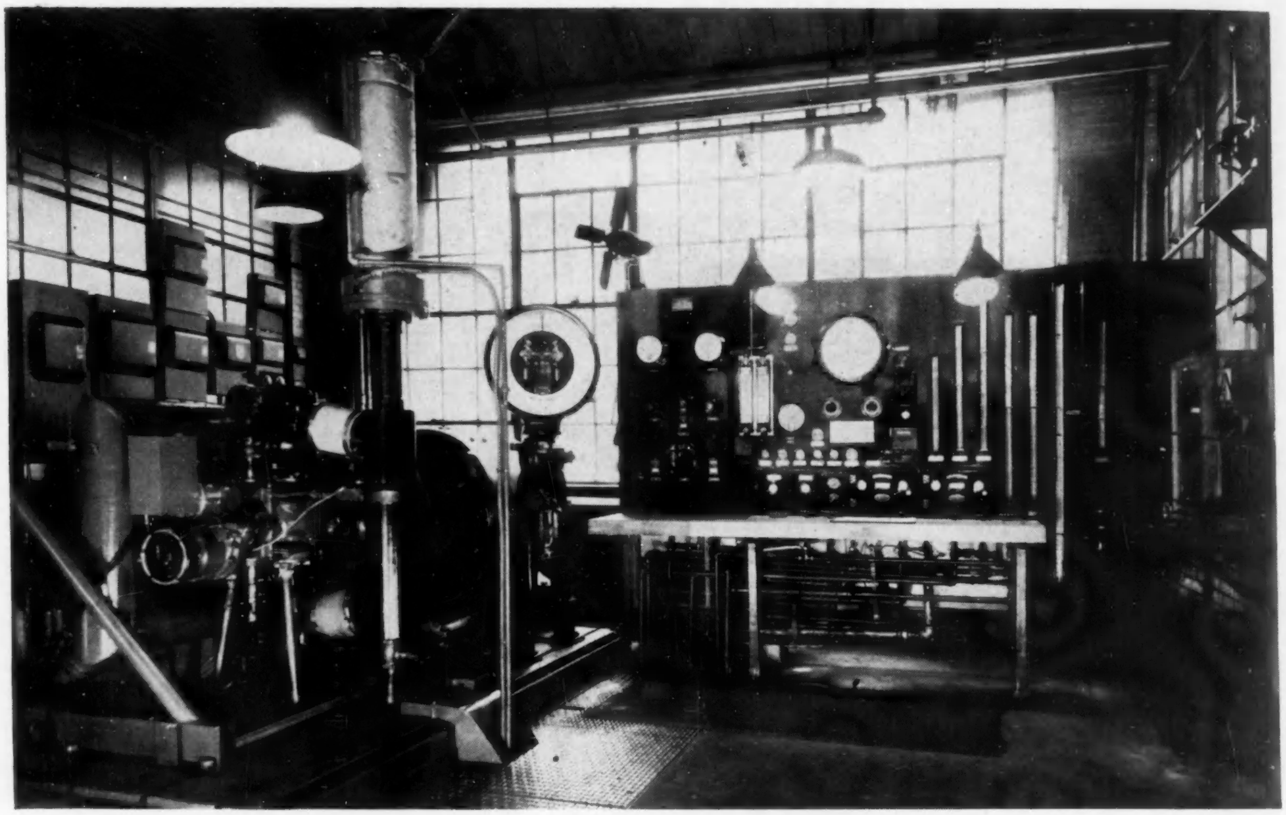
As it is the duty of the Internal Combustion Engine Section of the Bureau of Ships to develop sources of engines as well as engines themselves, I am glad to have this opportunity to place before you the Navy's design requirements and to explain the reasons for them. The present emergency is creating demands for engines far beyond the normal requirements and it now becomes necessary to develop sources which have not previously supplied the Navy.

■ Navy Requirements in General

The Bureau of Ships strives to express in the specifications it writes the cumulative experience of the Service as a whole. It does not express preference on debatable points such as two-stroke cycle vs. four-stroke cycle; single acting vs. double acting. The policy has been to develop designs which are commercially sound and which will find a demand commercially to the end that, as the emergency arises, the Navy's demands will be met with less adjustment than would otherwise be the case. A healthy rivalry results, which indicates its worth in steady progress. This progress demonstrates itself in higher specific outputs, in more compact designs, in lowered fuel consumption, and in higher reliability. One indication of progress concerning which much discussion takes place is the decreasing of specific weights of engines. Is this paralleled by decreasing reliability? The answer is that weight is not a criterion of reliability. Let us quote Admiral Robinson, the Chief of the Bureau of Ships, to convince the skeptic of results of this progress of twenty-five years: "The modern diesel engine, weighing less than one-third of the engine used in the World War, has a reliability of performance in the submarine service such that today it may be considered fully equal to any other propelling plant of similar characteristics." The Admiral was taking a great deal of satisfaction in expressing that opinion, as he has done much to further the advance of the diesel engine.

Excerpt from the paper of the same title by Lieut.-Com. Marshall M. Dana, U. S. Navy, in charge of Internal Combustion Engine Section, Bureau of Ships, presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.

ALTITUDE PERFORMANCE of



■ Fig. 1 – Lycoming single-cylinder engine laboratory

Introduction

WE have been investigating the altitude performance possible of attainment with liquid-cooled engines of fairly high specific output and consequent high groundboost. It is thought that the methods used may be of interest and applicable, with certain obvious revisions, to other types.

The use of the conventional centrifugal supercharger, with efficiencies as obtained on equipment in current production, has been assumed. The advantages to be obtained by improving the efficiency of the centrifugal compressor are well known and need not be discussed here.

■ Engine Assumptions

An engine of 1500 cu in. displacement, with rated crankshaft speed of 3000 rpm and desired rated power of 1000 hp will be assumed for example. The desired take-off power is 1200 hp.

■ Cylinder Performance

As the first step in the investigation of altitude performance, the power of the engine cylinder under all possible conditions of manifold pressure, manifold temperature, and exhaust pressure must be determined. These tests should be conducted with the specified fuel and at speeds including the proposed rated, take-off and military speeds. It is advisable to include a very extended range of conditions in the cylinder calibration, as extrapolation of the performance curves is of doubtful value. Single-cylinder laboratories are available for this work at every major engine plant. Our laboratory, which has been in operation on liquid-cooled cylinders for six years, is shown in Fig. 1. It is possible here to produce any desired exhaust suction; manifold pressures up to 150 in. hg; inlet air temperatures up to 400 F; and to handle engine speeds to 5000 rpm.

A characteristic cylinder calibration is shown in Fig. 2. It will be noted that output is expressed in indicated mep. This is the sum of the brake mep, and the friction mep as obtained by motoring the engine under the specified conditions for the power run. Each power run, according to our practice, is followed immediately by its friction run, the friction beam being read about 6 or 7 sec after the last power stroke, without changing speed. Indicated power, as thus obtained, has agreed with full-scale engine results quite closely. Air temperatures are measured ahead of the

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 10, 1940.]

High-Output Aircraft Engines

by R. N. DuBOIS
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THIS paper deals principally with a description of the methods used in, and a discussion of the results of, an investigation of the possible altitude performance with liquid-cooled engines of high output and consequent high ground boost.

As the first step in the investigation of altitude performance the power of the engine cylinder under all possible conditions of manifold pressure, manifold temperature, and exhaust pressure is determined. The single-cylinder laboratory employed can produce any desired exhaust suction; manifold pressures up to 150 in. hg; inlet air temperature up to 400 F; and handle engine speeds up to 5000 rpm.

Maintenance of power at altitude by means of the exhaust-driven turbo compressor, Mr. DuBois believes, affords a most convenient way out for the engine manufacturer; the troubles, if any, are transferred to the builder of the turbo and of the airplane. He considers "worthy of considerable attention" the use of a two-stage supercharger with an auxiliary engine to drive the first stage. He explains that this arrangement is especially applicable to large airplanes.

Other supercharger arrangements discussed are the single - stage geared compressor; two - speed drives; continuously variable drives; two - stage geared superchargers; and modified two-stage installations, such as two-speed both stages, by-passed first stage, and the two-speed first stage.

The two-speed first-stage method of using two stages is the most satisfactory yet considered, Mr. DuBois believes, since three steps of supercharging reduce the amount of throttling required and, therefore, improve the overall efficiency of the system.

■ ■ ■

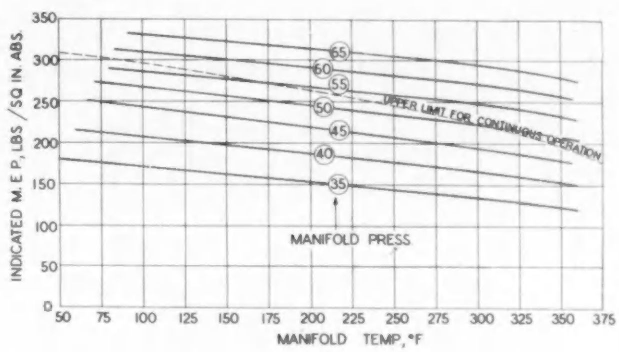
fuel-air mixture. The dotted line indicates the approximate upper limit of continuous operation with 100-octane fuel. Use of fuels of higher antiknock rating will raise these boundaries and also will straighten the high-temperature portion of the curves.

The calibrations, of which Fig. 2 is an example, are also repeated at exhaust pressures corresponding to all altitudes for which the power is to be computed. The range of manifold pressures covered is chosen, at each altitude, to include the manifold pressures computed for the range of impeller drive ratios as described in the following paragraphs.

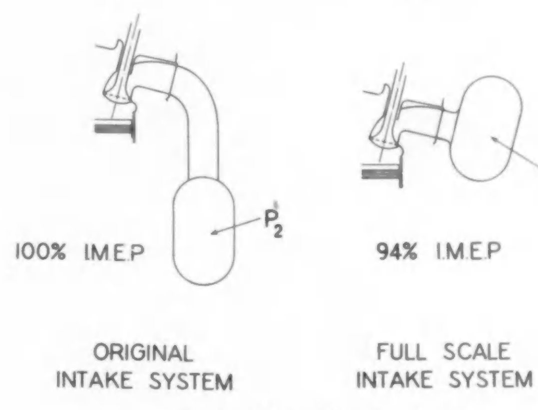
It has been found exceedingly important to maintain, in single-cylinder testing, the length and area of the individual intake pipes or manifold passages of the full-scale engine. Neglect of this detail may lead to misleading power results, as is shown in Fig. 3. The length of exhaust stack also should be kept equal to that of the full-scale engine, the stack then discharging into a large diameter pipe or "manifold." This method does not reproduce the pressure pulsations of the multicylinder intake or exhaust manifold, but these items do not seem as important as the maintenance of correct individual pipe lengths.

■ Single-Stage Geared Compressor

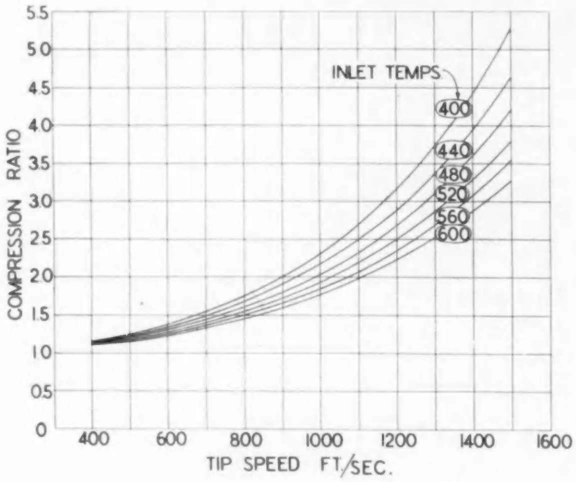
The impeller diameter is dictated by available space and by the airflow required for the desired engine horsepower.



■ Fig. 2 – Cylinder calibration – 3000 rpm – Exhaust pressure, 29.5 in. hg



■ Fig. 3 – Effect of intake pipe



■ Fig. 4 – Supercharger compression ratios versus impeller tip speed (pressure coefficient = 0.65)

For single stage, a 10-in. diameter impeller will be assumed. A series of impeller drive ratios is selected for investigation and computation of brake horsepower at altitude. At each altitude it is necessary to compute:

(a) Indicated horsepower;
(b) Supercharger horsepower;
(c) Engine friction horsepower;

from which the brake horsepower may be obtained by subtracting (b + c) from (a).

(a) Indicated Horsepower – The manifold pressure,

$$p_2 = Rp_1 = R\ (p_o - p_d) \qquad (1)$$

where R = supercharger compression ratio
p_o = atmospheric pressure
p_d = pressure drop between engine air inlet and supercharger inlet

Fig. 4 shows supercharger compression ratios versus impeller tip speed for supercharger inlet temperatures from 400 to 600 F (absolute). The pronounced influence of inlet temperature on compression ratio at high tip speeds is to be noted. Fig. 5 shows the results of a test of a supercharger inlet model, in which p_d is plotted against airflow. The inlet screen is in position on the model. The air consumption in pounds per hour has been found to be 5.7 to 5.8 times the ihp. A trial air consumption is assumed, p_d obtained from the curve, and the computation carried through ihp, being repeated if the trial air consumption is appreciably in error. At altitude, the values of p_d for sea level must be corrected by the altitude density ratio. The supercharger inlet temperature will be higher than atmospheric temperature by an amount which may be estimated from similar installations. The atmospheric temperature may be taken directly from the standard altitude[1], or with any predetermined margin of safety added to it. In these computations, the atmospheric temperature will be assumed to be 30 F higher than the values given by the standard altitude. Having determined R for each drive ratio at the altitude, the manifold temperature, T_2, may be computed from the formula for polytropic compression:

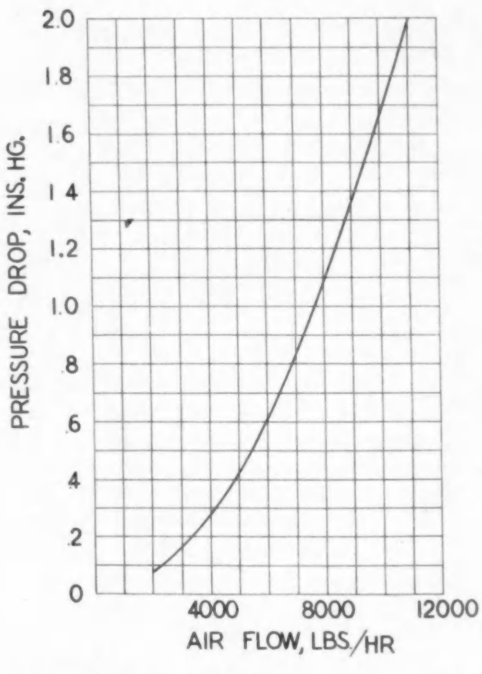
$$T_2 = T_1 R^{\frac{n-1}{n}} \qquad (2)$$

where T_1 is the supercharger inlet temperature and n is an exponent which has a value of 1.68. Reference to the cylinder calibration curve sheet for the proper engine speed will give the imep to be expected at the computed values of P_2 and T_2. The ihp may then be tabulated as shown in Table 1, column 17.

(b) Supercharger Horsepower – The fundamental equation for supercharger horsepower is

$$S = \frac{\frac{1}{k-1} WrT_1 \left(R^{\frac{k-1}{k}} - 1\right)}{33{,}000 \times E} \qquad (3)$$

where S = supercharger horsepower.
W = air flow in lb/min
r = gas constant (53.496 for normal air)[2]
E = efficiency of supercharger as a compressor (adiabatic shaft efficiency)



■ Fig. 5 – Supercharger inlet pressure loss versus airflow (density = 0.0729 lb per cu ft)

Dr. Sanford A. Moss and Chester W. Smith[2] have computed the values of $\left(R^{\frac{k-1}{k}} - 1\right)$ or "Y" for normal air at all useful supercharger compression ratios. The air consumption of the engine is directly proportional to the indicated horsepower and, as already stated, is 0.0966 lb/min, or 5.8 lb/hr per ihp. The supercharger efficiency, E, is 66%, at load coefficients in the range we are considering.

The constants of Equation (3) may be combined to

$$S = 0.000838\ IT_1Y \qquad (4)$$

[1] See NACA Technical Report No. 218, 1925: "Standard Atmosphere – Tables and Data," by W. S. Diehl.
[2] See ASME Transactions, Vol. 52, 1930, APM-52-9, pp. 93-102: "Engineering Computations for Air and Gases," by S. A. Moss and C. W. Smith.

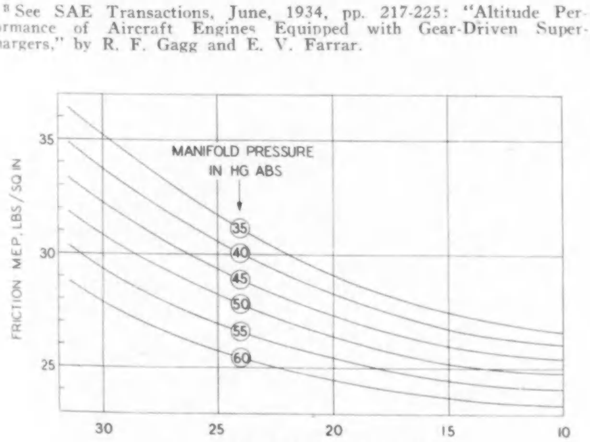
Table 1 – Power versus Altitude Computations – 1500 Cu In. – 3000 RPM

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
		Atmos. Temp., T_0					Manifold Temp., T_2				Inlet Press. Drop										
Impeller Drive Ratio	Impeller Tip Speed, ft/sec	F absolute	F	Super. Entrance Temp., T_1, F absolute	Supercharger Compression Ratio, R	γ	F absolute	F	Atmos. Press., p_0, in. hg	Estimated Air Flow, lb/hr	Sea Level p_d, in. hg	Altitude p_d, in. hg	Supercharger Entrance Press., p_1, in. hg	Manifold Press., p_2, in. hg	Imep, lb per sq in.	Ihp.	Air Flow, lb/hr	Supercharger Hp, S	Engine Fhp	Total Fhp	Bhp
Sea Level																					
6.0	785	548	89	553	1.480	0.11734	649	190	29.92	6900	0.79	0.75	29.17	43.2	210	1193	6920	65	124	189	1004
7.0	916	548	89	553	1.690	0.16009	684	225	29.92	7580	0.96	0.91	29.01	49.0	243	1335	7750	99	115	214	1121
8.0	1047	548	89	553	1.960	0.20979	726	267	29.92	8550	1.22	1.16	28.76	55.9	262	1490	8640	145	103	248	1242
9.0	1178	548	89	553	2.300	0.26581	775	316	29.92	9700	1.57	1.50	28.42	65.4	291	1654	9600	210	87	297	1357
10.0	1309	548	89	553	2.750	0.33147	832	373	29.92	10270	1.76	1.67	28.25	77.7	308	1750	10200	268	68	336	1414
5,000-Ft Altitude																					
6.0	785	531	71	536	1.504	0.12243	632	173	24.89	5700	0.54	0.59	24.30	36.6	177	1010	5858	56	112	167	843
7.0	916	531	71	536	1.727	0.16722	669	210	24.89	6580	0.72	0.79	24.10	41.6	201	1141	6620	86	106	192	949
8.0	1047	531	71	536	2.024	0.22084	713	254	24.89	7480	0.94	1.04	23.85	48.3	229	1300	7540	131	96	227	1073
9.0	1178	531	71	536	2.375	0.27736	761	302	24.89	8450	1.20	1.32	23.57	56.0	258	1466	8500	185	86	271	1195
10.0	1309	531	71	536	2.825	0.34165	816	357	24.89	9400	1.48	1.63	23.26	65.7	280	1590	9250	244	73	317	1273
10,000-Ft Altitude																					
6.0	785	513	53	519	1.528	0.12747	615	156	20.58	4600	0.35	0.46	20.12	30.7	142	807	4680	45	106	151	656
7.0	916	513	53	519	1.757	0.17292	652	193	20.58	5350	0.48	0.62	19.96	35.1	166	944	5480	71	101	172	772
8.0	1047	513	53	519	2.067	0.22812	697	238	20.58	6250	0.65	0.84	19.75	40.9	192	1090	6320	108	95	203	887
9.0	1178	513	53	519	2.440	0.28716	744	285	20.58	7100	0.84	1.08	19.50	47.6	219	1244	7210	156	88	244	1000
10.0	1309	513	53	519	2.910	0.35295	800	341	20.58	8000	1.07	1.38	19.20	55.8	242	1375	7980	211	78	289	1086
15,000-Ft Altitude																					
6.0	785	495	36	501	1.550	0.13205	598	139	16.88	3600	0.23	0.35	16.54	25.7	111	632	3670	31	101	132	500
7.0	916	495	36	501	1.785	0.17818	634	175	16.88	4340	0.32	0.49	16.39	29.3	137	778	4290	58	98	156	612
8.0	1047	495	36	501	2.116	0.23629	679	220	16.88	5100	0.44	0.66	16.22	34.3	159	904	5250	90	94	184	720
9.0	1178	495	36	501	2.510	0.29750	728	269	16.88	5890	0.58	0.87	16.01	40.2	185	1050	6100	131	88	219	831
10.0	1309	495	36	501	3.010	0.36590	783	324	16.88	6500	0.70	1.06	15.82	47.6	209	1188	6900	183	81	264	924
20,000-Ft Altitude																					
6.0	785	477	18	484	1.575	0.13718	582	123	13.75	2810	0.14	0.26	13.49	21.2	85	483	2800	20	100	120	363
7.0	916	477	18	484	1.825	0.18560	618	159	13.75	3500	0.21	0.38	13.37	24.4	110	624	3620	47	97	144	480
8.0	1047	477	18	484	2.174	0.24579	664	205	13.75	4220	0.31	0.54	13.21	28.7	134	741	4410	74	93	167	574
9.0	1178	477	18	484	2.590	0.30907	712	253	13.75	4840	0.39	0.70	13.05	33.8	155	887	5150	113	89	202	685
10.0	1309	477	18	484	3.110	0.3786	766	307	13.75	5600	0.52	0.93	12.82	39.9	172	1013	5870	155	84	239	774

where I = indicated horsepower. The supercharger horsepower for each drive ratio is then computed and tabulated as in Table 1, column 19.

(c) Engine Friction Horsepower – This is considered, in our computations, to be the power required to overcome the mechanical friction and cylinder pumping losses of the engine *at the manifold pressure and exhaust pressure corresponding to the altitude for which the brake horsepower is to be computed.* Its value will change with change in these pressures, and it is possible to imagine manifold pressures high enough, and exhaust pressures low enough so that the engine friction horsepower becomes zero and the engine, acting as an air motor between these pressures, overcomes its own mechanical friction. Under these conditions, the supercharger horsepower, only, would be subtracted from the indicated horsepower to obtain the brake horsepower. Fig. 6 shows the change in friction mep with change in exhaust pressure from single-cylinder tests at different manifold pressures.

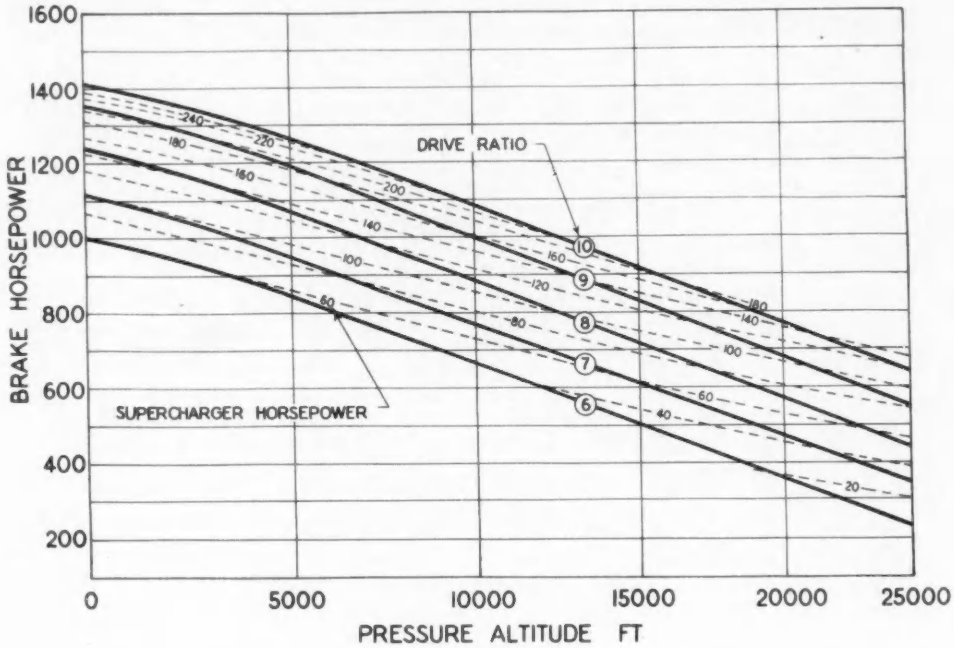
[8] See SAE Transactions, June, 1934, pp. 217-225: "Altitude Performance of Aircraft Engines Equipped with Gear-Driven Superchargers," by R. F. Gagg and E. V. Farrar.



■ Fig. 6 – Friction mean effective pressure versus exhaust pressure – 3000 rpm

Where a full-scale engine is available, the total friction horsepower is obtained by motoring the engine under any convenient conditions of manifold pressure and temperature. The supercharger horsepower is then computed for these conditions and deducted to obtain the engine friction horsepower. This value may be corrected to any desired altitude and manifold pressure by use of the curves of Fig. 6 and entered in column 20 of Table 1. If a full-scale engine has not been built, it is necessary to estimate the overall mechanical efficiency based on previous experience and, using the resulting estimated total friction, proceed as just outlined.

(d) Brake Horsepower – The resulting brake horsepower is now tabulated in the last line of Table 1 for each altitude and drive ratio. Fig. 7 shows the results plotted against pressure altitude. The shape of the power versus altitude curve is determined chiefly by that of the cylinder calibration curve. Gagg and Farrar[3] have shown that the altitude performance of engines tested up to 1934 follows a straight line when plotted versus density ratio and that the brake horsepower at 20,000-ft altitude is 47% of the sea-level power. It will be noted in Fig. 7 that this per-



■ Fig. 7 – Single-stage supercharger – Brake horsepower versus altitude

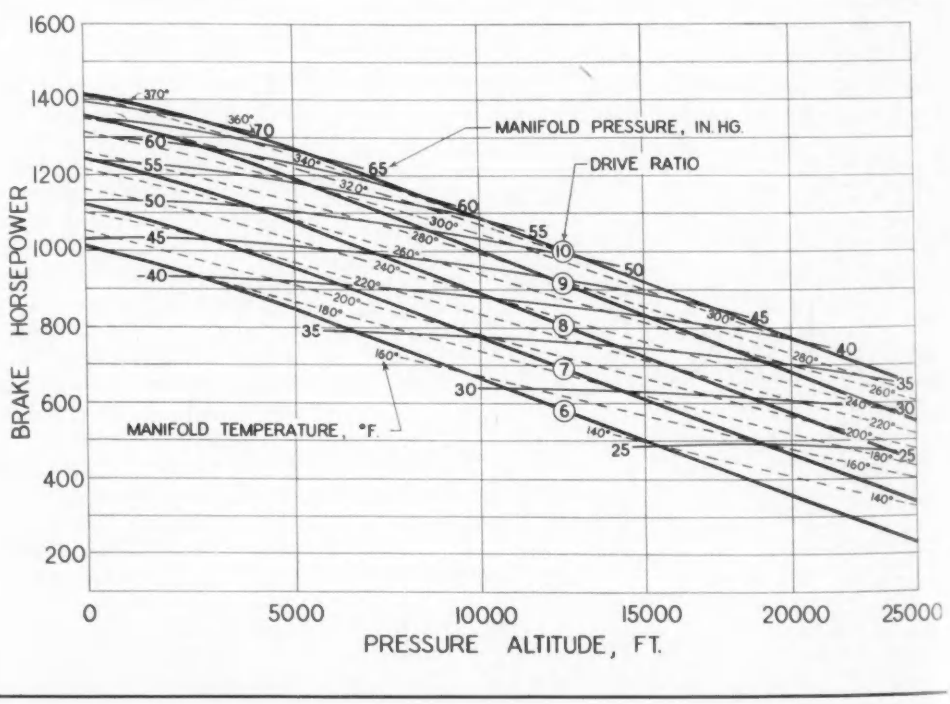
centage appears valid for the 8:1 drive ratio only. At 6:1, the figure is 36% and at 10:1 the power is 55% of the sea-level value.

■ Supercharger Horsepower Lines

As an aid to design, it is desirable to plot lines of constant supercharger horsepower on the drive ratio lines, as shown in Fig. 7. It is interesting to note that the supercharger horsepower required to develop 1000 bhp at 12,500-ft altitude is double that required at 4300 ft. Also that, at 12,500 ft, 600 bhp is obtained with 40 supercharger hp, whereas 200 supercharger hp is required to raise this to 1000 bhp.

■ Limiting Conditions

It is convenient to plot lines of constant manifold pressure and temperature on the drive ratio lines, as in Fig. 8. By reference to the cylinder calibration chart for permissible manifold pressures and temperatures, the possibility



■ Fig. 8 – Single-stage supercharger – Brake horsepower versus altitude with lines of constant manifold pressure and temperature

of continuous operation at any desired output and altitude may be ascertained.

■ Effect of Inlet Restriction

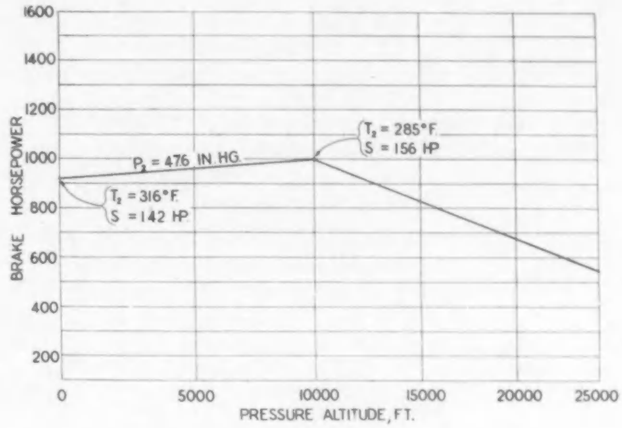
The reward for making easier the passage of the engine air from the atmosphere to the supercharger inlet is greatly increased with altitude. This is especially true of highly ground-boosted engines such as our hypothetical example. The effect of a restriction in the inlet causing an increase of 0.3 in. hg in the inlet pressure drop, p_d, at sea level (and this is a reasonable drop for an 8-mesh screen) is such as to reduce the full-throttle bhp at 10:1 drive ratio and 25,000-ft altitude from 648 to 578. This is a loss of 70 hp or almost 11%.

■ Use of Chart

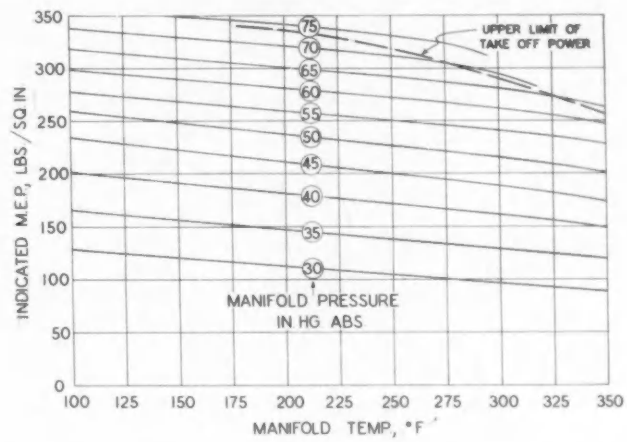
The power chart of Fig. 8 is basic for all single-stage operation at that speed. Examples of the use of these charts in selection of drive ratios and in constructing model specification power curves, follow:

Example (a): Maximum Single Speed Normal Critical Altitude for 1000 Bhp – The 1000-bhp line is followed to the right until the upper boundary of the region for safe continuous operation is approached. The drive ratio there is found to be 9:1, at 10,000-ft altitude. If this ratio is not feasible from a gear design standpoint, the nearest lower acceptable ratio is used. The manifold pressure is found by interpolation to be 47.6 in. hg. Following this drive ratio line back to sea level, it is found that the manifold temperature there is 316 F. This is not affected by throttling to 47.6 in. hg manifold pressure, at which the engine will be operated between sea level and the critical altitude of 10,000 ft. Reference is now made to the sea-level cylinder calibration chart of Fig. 2, to make sure that 316 F and 47.6 in. hg are within the safe region for continuous operation. Should this not have been the case, it would be necessary to use a lower drive ratio and lower critical altitude. The imep for the foregoing manifold condition is obtained from Fig. 2 and converted to indicated horsepower. The supercharger horsepower and engine friction horsepower are computed in the manner already described. The resulting bhp of 920 is the normal sea-level rating of the engine. The power above critical altitude follows the 9:1 drive ratio line from Fig. 8, so that the normal altitude power curve is as shown in Fig. 9.

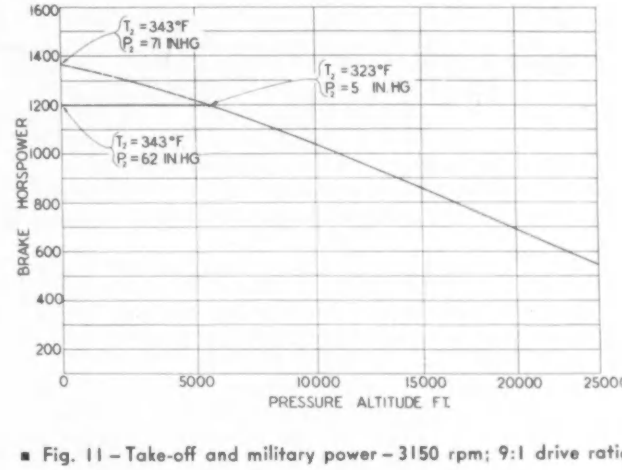
Example (b): Take-off and Military Power – Having selected the supercharger drive ratio in Example (a) the take-off speed is chosen and the full-throttle brake horsepower computed as described for the results tabulated in Table 1, but for 9:1 drive ratio only. To do this requires a cylinder calibration chart, Fig. 10, for the take-off speed, which will be assumed to be 105% of rated speed, or 3150 rpm. Fig. 11 shows the power curve. It is seen that, at sea level, 1360 hp is developed, at 71 in. hg manifold pressure and 343 F manifold temperature. As this is greater than the required 1200 hp, the engine will be throttled to that value. The full-throttle power curve reaches 1200 hp at 5500 ft, indicating that take-off power may be maintained up to that level. The required take-off manifold pressure must next be found, to determine if it lies within the region for safe take-off operation on the cylinder calibration chart. To do this, it is first necessary to compute the imep required to produce 1200 bhp. The procedure is simplified by the justified assumption that, at constant tip speed and inlet temperature, the supercharger horsepower



■ Fig. 9 – Normal power – 3000 rpm; 9:1 drive ratio; single stage



■ Fig. 10 – Cylinder calibration – 3150 rpm; exhaust pressure, 29.5 in. hg



■ Fig. 11 – Take-off and military power – 3150 rpm; 9:1 drive ratio

is proportional to the air flow and hence, to the indicated horsepower. The engine friction horsepower is greater than the full-throttle value, due to the reduced manifold pressure.

where

B_F = full-throttle brake horsepower.
B_T = required take-off brake horsepower.
I_F = full-throttle indicated horsepower.
I_T = required take-off indicated horsepower.
S_F = full-throttle supercharger horsepower.
S_T = supercharger horsepower at take-off power.
F_F = engine friction horsepower at full throttle.
F_T = engine friction horsepower at take-off power.

$$I_T = B_T + S_T + F_T$$

and

$$S_T = S_F\left(\frac{I_T}{I_F}\right)$$

substituting and combining

$$I_T = B_T + S_F\left(\frac{I_T}{I_F}\right) + F_T$$

I_F and S_F may be obtained from the computed values for Fig. 11. F_T may be estimated for a trial solution by correcting F_F to an assumed take-off manifold pressure, using curves of fmep at 3150 rpm versus manifold temperature and exhaust pressure, computed as was done for Fig. 6.

$$I_T = 1200 + S_F\left(\frac{I_T}{I_F}\right) + F_T \quad (4)$$

= 1530 hp, corresponding, at 3150 rpm to 257 lb per sq in. imep.

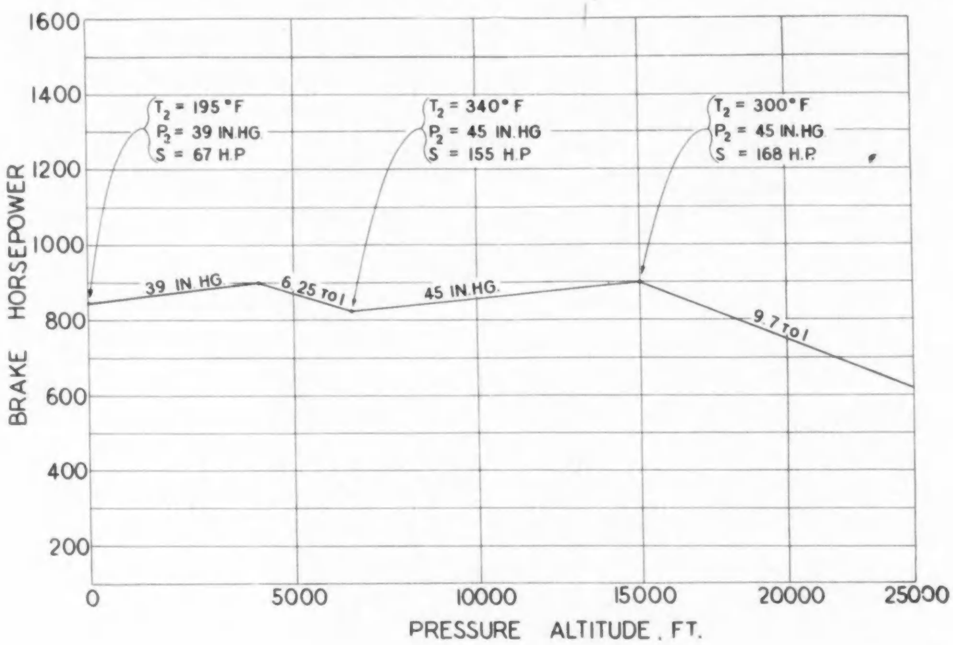
Referring back to the sea-level cylinder calibration chart, Fig. 10, it is seen that, at manifold temperature of 343 F, 257 lb per sq in. imep is developed at 62 in. hg manifold pressure which is within the region for safe take-off operation.

Should the first take-off speed result in manifold temperature too high for safe operation, the speed may be reduced, with resultant increase in manifold pressure required to obtain the desired power.

If military speed is to be the same as take-off speed, 5500 ft becomes the military critical altitude, and the military power then follows the 9:1 drive ratio line as shown in Fig. 11, reaching 700 hp at 20,000-ft altitude.

Example (c): Selection of Two-Speed Drive Ratios – These are governed by the limiting manifold conditions at sea level and at the engagement of second speed. Reference to Fig. 8 shows that, to obtain a desired 900 hp at 15,000 ft, a drive ratio of 9.7:1 is necessary and the manifold pressure will be 45 in. hg. The engine will be throttled to 45 in. hg in second speed, and the altitude below which second speed may not be used is that at which the manifold temperature approaches the limit for safe operation at that manifold pressure. Fig. 2 is consulted and 340 F taken as the limiting manifold temperature. This is approached with 9.7:1 drive ratio, at approximately 6500-ft altitude. The imep corresponding to 45 in. hg and 340 F is found by reference to the proper cylinder calibration and converted to ihp. The supercharger horsepower and engine friction horsepower are computed as already explained, and the resulting bhp obtained. It is found to be 823 hp. The altitude of 6500 ft and power of 823 bhp locate the point of engagement of second speed, which is marked on Fig. 8 and found to lie on the 6.25:1 drive ratio line. This becomes the first speed ratio. To find the first critical altitude, at which 900 bhp is required, the 6.25:1 drive ratio line is followed down until the required horsepower is reached, at 3600 ft.

The manifold pressure at this point is seen to be 39 in. hg. The manifold temperature at sea level is found, at 6.25:1 drive ratio, to be 195 F. The brake horsepower at 39 in. hg and 195 F is now computed as described in Example (a) and is 846 hp. The pressure and temperature are checked against the cylinder calibration chart and are



■ Fig. 12 – Normal power – 3000 rpm; two-speed single stage; first ratio, 6.25:1; second ratio, 9.7:1

well within the region for continuous safe operation. Fig. 12 shows the resulting power versus altitude curve.

The conditions for take-off and military operation may now be determined and checked against the cylinder calibration, using the methods of Example (b).

Example (d): Continuously Variable Drive – Referring again to the drive ratio chart for rated speed, Fig. 8, the range of drive ratios required from a continuously variable supercharger drive may be determined. It is seen that at sea level, 6:1 ratio is required to develop the rating of 1000 hp. The manifold temperature is 190 F, instead of 316 F at sea level for the single-speed engine with 10,000-ft rating, and 340 F for the two-speed engine at 6500 ft in second speed. The permissible altitude rating is limited by the manifold temperature and pressure at rated altitude, rather than at sea level in the case of the single-speed engine or just after engagement of second speed, in the two-speed engine. The supercharger horsepower required may be obtained from the curves of Fig. 7. A further advantage of the continuously variable drive is the reduction of supercharger power in cruising operation, by lowering impeller speed to obtain the required manifold pressure, instead of by throttling. The foregoing considerations explain the continued effort to obtain a satisfactory variable-speed drive.

■ Two-Stage Geared Compressor

The simplest application of multi-staging to the geared centrifugal compressor involves two impellers mounted on the same shaft, geared at a fixed drive ratio to the engine crankshaft. An intercooler is placed after the first stage, and the carburetor, if any, is located between the intercooler and the second stage. Such an arrangement is shown diagrammatically in Fig. 13. It is possible to construct charts of brake horsepower versus altitude at various drive ratios for the engine equipped with two-stage supercharging, although the process is necessarily more complex.

For an example, the diameter of both impellers will be assumed to be 10 in.

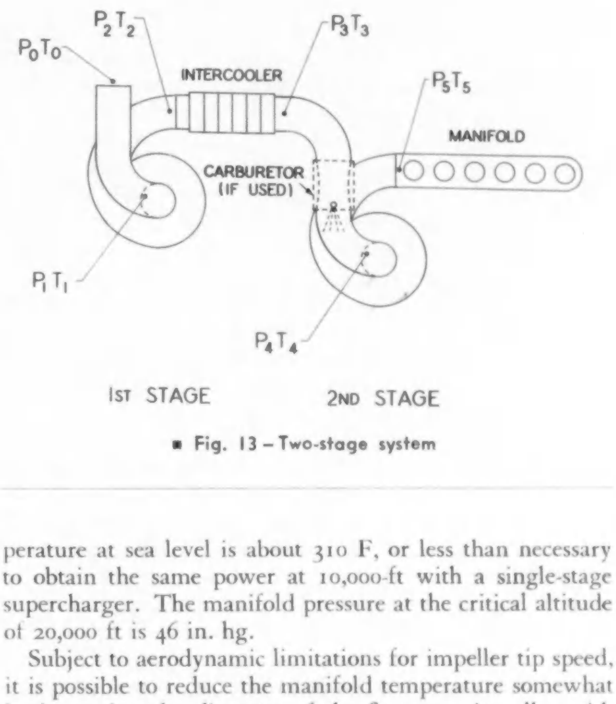
Indicated Horsepower – To obtain the indicated horsepower, it is necessary to compute the manifold pressure P_5 and manifold temperature T_5 for each drive ratio and altitude within the range of the chart. Referring to Fig. 13, it is obvious that P_2 and T_2 may be computed as for the single-stage compressor. The intercooler pressure drop ($P_2 - P_3$) and temperature drop ($T_2 - T_3$) may be obtained from tests of the intercooler, should such information be available. Since such is very seldom the case, the intercooler performance will be assumed. The pressure drop, $P_2 - P_3$, will be assumed to be 1 in. hg at sea level, which is the goal that all good intercooler designers are approaching. The temperature drop will be taken to be $\frac{1}{2} \Delta T$ or $T_2 - T_3 = \frac{1}{2} (T_2 - T_1)$. Centrifugal compressor characteristics are such that, at constant tip speed, the temperature rise, ΔT, is constant at any inlet temperature. The same consideration holds for the second stage, so that the overall temperature rise $T_5 - T_1$, for each drive ratio is constant, at all altitudes if our assumed intercooler performance is valid. If a carburetor is used between intercooler and second stage, the vaporization drop, $T_3 - T_4$, will be approximately 40 F, and the manifold temperature, T_5, be 40 F lower than if air only were compressed. However, since the cylinder calibration curves used in our computations are plotted versus manifold temperatures measured *before* the introduction of fuel to the charge, the vaporization drop will not be considered in computing T_5 for reference to these curves for indicated mep.

The pressure loss, $P_3 - P_4$, must be estimated or obtained from flow tests of the second-stage carburetor and/or air inlet, as was done in the single-stage computations. The supercharger compression ratio and manifold pressure, P_5, are computed, using the curves of Fig. 4. The vaporization drop when using a carburetor will result in a higher value of P_5 than when compressing air, due to the increased supercharger compression ratio.

Supercharger Horsepower – The supercharger horsepower for each stage is computed as for the single-stage system. T_4 is used as the inlet temperature of the second stage.

Engine Friction Horsepower – The engine friction horsepower may be computed exactly in the same manner as the single-stage operation, using the manifold pressure P_5 and the altitude exhaust pressure for correcting from the sea-level friction values.

Brake Horsepower – Fig. 14 shows brake horsepower at rated speed for the chosen range of drive ratios, plotted against pressure altitude. It is seen that rated power of 1000 bhp at 20,000-ft altitude may be maintained with a manifold temperature of only 260 F at 7.9:1 drive ratio. Using this ratio as a single-speed drive, the manifold temperature at sea level is about 310 F, or less than necessary to obtain the same power at 10,000-ft with a single-stage supercharger. The manifold pressure at the critical altitude of 20,000 ft is 46 in. hg.

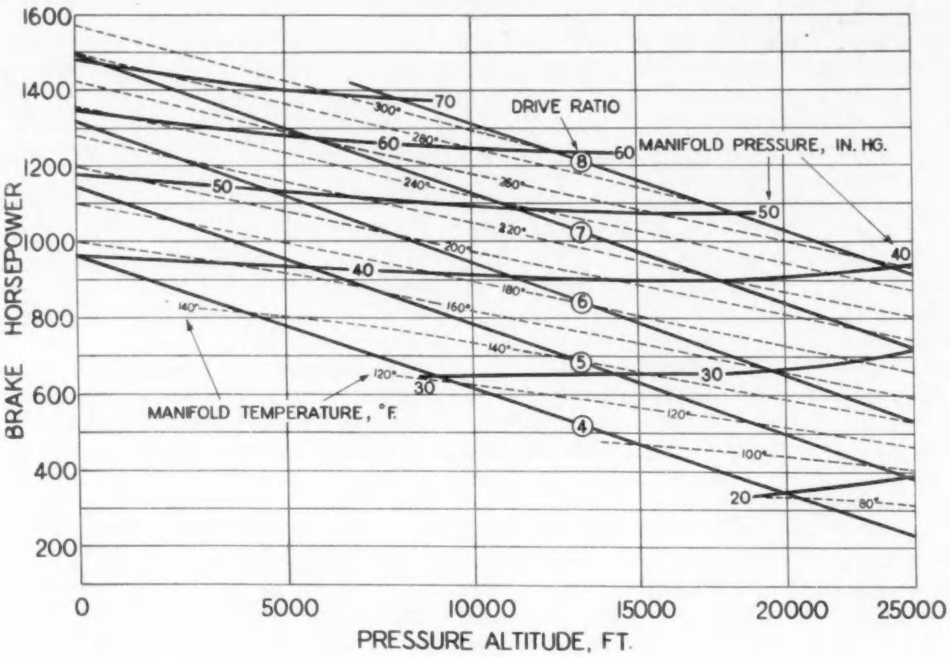


■ Fig. 13 – Two-stage system

Subject to aerodynamic limitations for impeller tip speed, it is possible to reduce the manifold temperature somewhat by increasing the diameter of the first-stage impeller with corresponding reduction in diameter of the second-stage impeller. This increases the intercooler entrance temperature and heat absorbed accordingly. At 7:1 drive ratio on our hypothetical engine at sea level, a decrease in manifold temperature of approximately 13 F is obtained by increasing the diameter of the first-stage impeller to 11 in. and reducing the second stage to 9 in.

■ Modified Two-Stage Installations

Two-Speed, Both Stages – From Fig. 14, it is seen that by using a two-speed drive, the critical altitude may be



■ Fig. 14 – Two-stage supercharger – $1/2\ \Delta T$ intercooler; 3000 rpm

pushed still higher without causing excessive manifold temperatures at sea level. It is believed that, with most engines, the critical altitude is limited by the horsepower requirements of the second-speed clutch, rather than by manifold conditions at second-speed engagement. At 10,000 ft and 8:1 drive ratio, the supercharger power for both stages totals 336 hp, which is a fair size job for any clutch. Curves of altitude power for such an arrangement may be made in the same manner as for the two-speed single-stage installation.

Bypassed First Stage – A further modification of the original two-stage arrangement involves a positively driven second stage and a clutch by which the first stage may be disengaged at low altitudes. For this operation the inlet air is introduced between intercooler and "second" stage. The power below clutch engagement follows the drive ratio lines of Fig. 8, and on engagement rises to that shown in Fig. 14 for the same drive ratio. It is obvious that this method reduces the clutch power by approximately 50%, or from 336 hp to 169 hp in the example of the preceding paragraph.

The same results may be approximated by omitting the clutch and "smothering" the first stage at low altitudes by blocking its exit. Enough air must be allowed to pass to prevent overheating of the impeller and associated parts. The reduction of first-stage supercharger power is here inversely proportional to the percentage of the normal air flow necessary for cooling. Pulsations in the first stage are a possible hazard due to the low load coefficient.

Two-Speed First Stage – The first stage may be equipped with a two-speed drive, including a neutral position. The inlet air when the first stage is inoperative is introduced between intercooler and "second" stage. The power at neutral first stage and high first stage speeds is obtained as already described. Computation of power in low first stage speeds involves a second set of two-stage computations using two different drive ratios instead of one.

This method of using two stages is the most satisfactory yet considered, since the three steps of supercharging reduce the amount of throttling required and, therefore, improve the overall efficiency of the system.

■ Exhaust Turbo-Compressor

Maintenance of power at altitude by means of the exhaust-driven turbo-compressor affords a most convenient way out for the engine manufacturer. The troubles, if any, are transferred to the builder of the turbo and of the airplane. The operation of the device has been so excellently described in papers before this Society by Berger and Chenoweth[4], that further explanation will not be attempted here.

In applying the turbo-compressor to our hypothetical engine, we assume an engine inlet temperature of 90 F, and select a drive ratio sufficient to develop rated power at sea level. Take-off power will be developed by increasing the engine inlet pressure by means of the turbo, to produce take-off manifold pressure at the take-off speed. The altitude power curve becomes a straight line at rated power up to the altitude rating of the turbo-supercharger, after which the power may be considered to decrease at the normal rate of the drive ratio line selected.

■ Auxiliary-Powered First Stage

This arrangement is especially applicable to large airplanes and resembles the just-mentioned turbo-compressor two-stage system, except that the first-stage compressor is powered by an auxiliary engine instead of an exhaust turbine. A schematic diagram is shown in Fig. 15. One of the fundamental principles of this method is that the engine take off without the use of the auxiliary-powered first stage. The drive ratio of the second-stage supercharger is therefore selected to develop take-off power at

[4] See SAE Transactions, November, 1938, pp. 472-484: "Supercharger Installation Problems," by A. L. Berger and Opie Chenoweth.

sea level. The auxiliary engine is idling during take-off and is gradually speeded up by the inlet pressure control, which is adjusted to maintain sea-level pressure at the engine inlet. This control would function in the same manner as the turbo-supercharger control, except that it would operate the auxiliary engine throttle instead of the waste gate. The direct air inlet for take-off would be closed by the control as the auxiliary engine was opened up.

It will be seen in Fig. 15 that the auxiliary engine is supplied with air at sea-level pressure up to the critical altitude of the system. Analysis of this method of supercharging results in the following conclusions:

1. One auxiliary engine and supercharger may be connected to several engines, since failure of the auxiliary plant will only result in reducing the available power to that of sea-level engines operating at altitude.

2. The air requirements of the auxiliary engine do not increase the total air supply over that of other arrangements since, at the same brake horsepower, the indicated horsepower and air supply of the main engine is reduced by the amount required for the auxiliary.

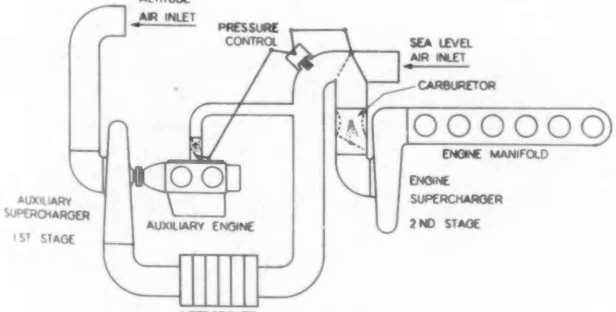
3. The specific weight of the combination of auxiliary and main engines, in pounds per horsepower at the propeller shaft, will not exceed that of the main engine when driving both stages in the geared two-stage arrangement. This is due to the fact that the brake horsepower of the main engine is increased by the power formerly required to drive the first-stage supercharger. If the weight per horsepower of the auxiliary engine exceeds that of the main engine, there will be a minor increase in specific weight of the combination.

4. In comparison with the geared two-stage compressor, the indicated horsepower of the main engine is less, for a given brake output, by the indicated power formerly required for the first-stage drive, or, as just explained, for a given indicated horsepower, the brake horsepower is correspondingly increased. The advantage of reducing the required indicated mean effective pressure, when operating near the limiting output of the cylinder, is obvious.

The auxiliary engine as a drive for first-stage supercharging is believed to be worthy of considerable attention.

■ Conclusion

It is believed that the high-output engine, as such, is capable of satisfactory altitude performance, and that the continued development of intercoolers and methods of supercharging will allow this position to be maintained in the face of future requirements.



■ Fig. 15 – Auxiliary powered first-stage system

Structural Developments in Motor Coach Design

THE typical transit bus of 1941 is constructed so that the maximum dimensions of the whole bus are incorporated into a box-girder structure to provide the maximum section modulus, thereby minimizing unit stress, and is built of forms and shapes especially designed to work harmoniously in such construction. There is a growing tendency to impart to the various portions of this structure functional formation, and to give as many of them as possible multiple purposes, thus gaining simplicity, rigidity, improved appearance, greater stamina, lighter weight and more economical manufacturing.

In the development of these structures a great deal of attention has been paid to the proper distribution of stresses and strains to avoid their localization.

It is only fair to confess that the modern motor bus owes much to the parallel development which has taken place in airplane structures, but it would be a mistake to assume that buses have merely followed airplane developments. At the time buses were built with wood frames with steel gussets and sheet-steel panels, airplanes were being made of wood, wire, and cotton. The later development of all-aluminum planes was paralleled closely by early attempts to employ this useful metal effectively in bus body construction. The alloy-steel tubing skeleton with non-stressed aluminum skin construction was developed experimentally on both bus bodies and planes at about the same time; while modern stainless-steel structures were fairly well developed for truck, bus, and railcar bodies before they were seriously considered in airplane design.

Each of these materials has had a controlling effect upon the type of design employed with them as well as methods used in joining the parts. Each material has presented its own problems arising from their wide divergences of relative physical properties and also has presented a different problem from the standpoint of cost of the material itself, its fabrication, and the extent of tooling required.

So long as wood, strap iron, and malleable castings were the principal materials of construction, it was necessary to rely upon bolts, screws, and glue to hold the parts together. With the shift to all-metal construction, rivets assumed a more important role. The development of gas and arc welding broadened the possibilities of aluminum and high-tensile steel, but stainless-steel progress was forced to await the development of spot and shot resistance welding and recently high-frequency flash welding.

Naturally, in an industry in which developments have been rapid and in which competition is keen – particularly so in view of the comparatively small number of operators and the comparatively large investments in equipment made by each – it is to be expected that there should be some divergence of procedure, but all builders seem to be seeking the common goal of stiffer, simpler, and lighter structures. In the speculative realm of future development are such possibilities as monocoque structures and sectional units made of synthetic-resin plastics, high-tensile or stainless-steel construction employing paper-thin sheets and shapes, and aluminum or even magnesium alloys in various forms.

Excerpts from the paper by the same title by Merrill C. Horine and Harry S. Bernard, Mack Mfg. Corp., presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 6, 1941.

An Instrument for Measuring the IGNITION QU

THIS paper is a report of the work done on the development, by Caterpillar Tractor Co., of the cetane valve, an instrument used for the measurement of the cetane ratings of diesel fuels by throttling the intake air of a one-cylinder test engine. The author traces the history of the valve from its conception at the time of rating diesel fuels for altitude work to the present design, which has been used very successfully on a 5¾ x 8-in. engine, a 3¾ x 5-in. engine, and is being adapted to the CFR test engine.

Ratings made with the instrument, consisting of both reference blends and commercial fuels, are compared with results obtained by the coincident flash method. The advantages of the cetane valve as enumerated by the author are:

1. Simplicity of instrument and method;
2. Ease and quickness of operation;
3. Reproducibility and accuracy of results; and
4. Adaptability to engines of various strokes and bores.

THE purpose of this paper is to describe the development and the use of the Caterpillar cetane valve, and to present some ratings made with the instrument. The development of the valve was carried out in the San Leandro and Peoria Research Laboratories of the Caterpillar Tractor Co.

The work was undertaken because it was felt that the use of the cetane number as a measure of the ignition quality of diesel fuel required a simple, rapid, accurate, and reproducible method and instrument for its determination.

History

The application of reduced intake pressure to ignition quality rating was first conceived when an attempt was made to rate diesel fuels for altitude operation. It was found that, as the intake pressure was reduced, the ignition lag increased until a point was reached where the engine misfired.

The first use of this principle was made when a butterfly valve, Fig. 1, was placed in the intake of a Caterpillar one-cylinder test engine. The valve was equipped with a graduated quadrant to determine the position of the butterfly in degrees.

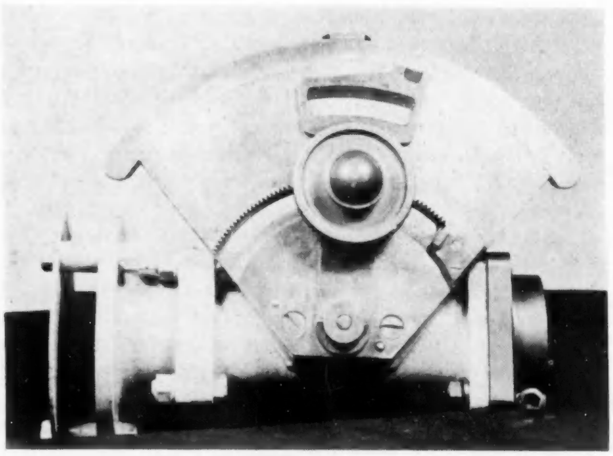


Fig. 1 – Butterfly valve

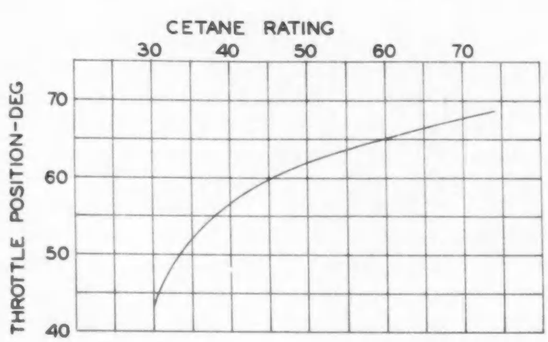


Fig. 2 – Calibration curve for butterfly valve

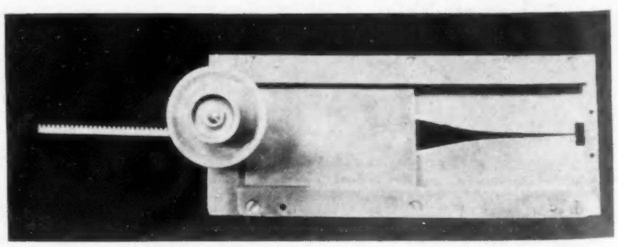


Fig. 3 – Hand-operated gate valve

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.]

QUALITY of DIESEL FUELS

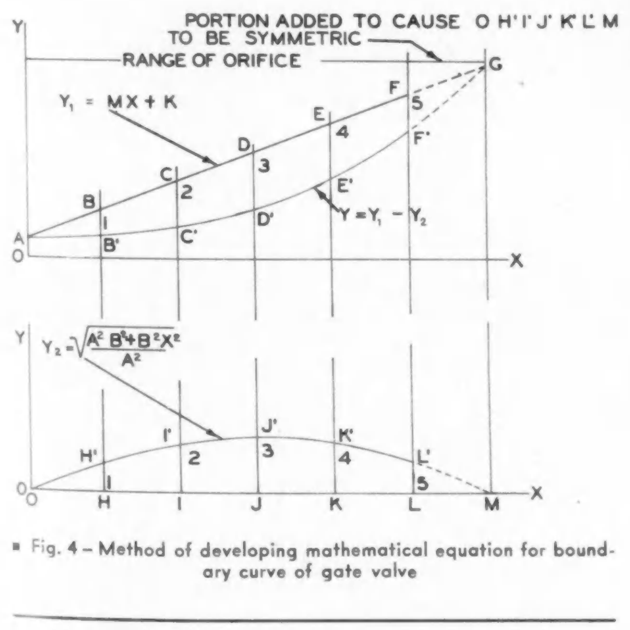
by WILLIAM H. BROWNE
Caterpillar Tractor Co.

The procedure was to close the butterfly slowly until a misfire of the engine occurred as signaled by a puff of smoke from the exhaust tap (a small tube connected to the exhaust pipe about 6 in. from the head). The position of the butterfly at this instant, when referred to a calibration curve, gave the cetane rating of the fuel under test. This procedure has been generally adhered to with the later developments of the valve, although some changes in the operating conditions of the engine were made for simpler testing.

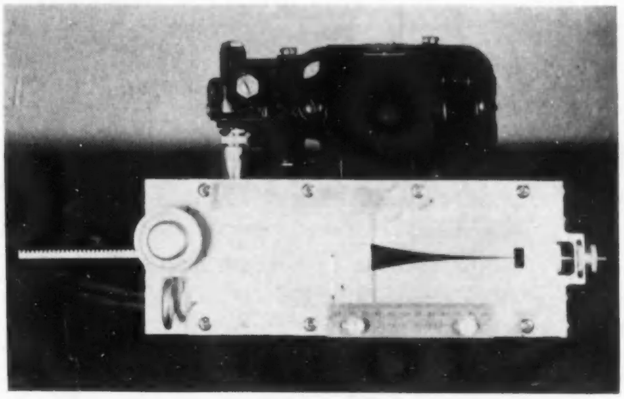
The calibration curve, shown in Fig. 2, for this type of valve requires the use of six or more reference fuels for its determination. As it had to be checked every day, the result was loss of time which might otherwise have been spent in rating test fuels. It was believed that this difficulty could be overcome by developing an instrument that would have a straight-line calibration curve and thus require only two reference fuels for standardization.

This result was accomplished by the valve of Fig. 3. This design had a straight-edged gate moving across an orifice of varying cross-section. The first approximation for this orifice was formed from the data gathered with the butterfly valve. To bring it to its final form was a cut-and-try process, each change in shape being tested by plotting cetane rating as a function of gate position at misfire, as measured in inches. Standard fuels were used for this work. When the plotting resulted in a repeatable straight line with less than one number deviation, the dimensions of the orifice were recorded and a mathematical equation developed to express the shape of the boundary curve as a function of gate position.

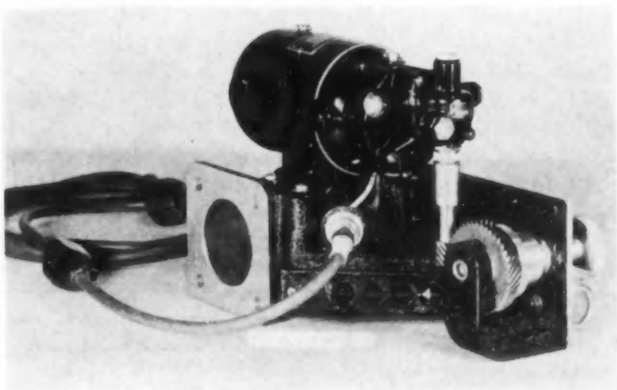
The method of developing this equation is shown in Fig. 4. The curve $AB'C'D'E'F'$ represents one boundary curve of the orifice. A straight line $ABCDEF$ was drawn and extended to G. (The point G was chosen by trial and error to cause the vertical differences between the straight line and the orifice curve to be symmetrical about a verti-



■ Fig. 4 – Method of developing mathematical equation for boundary curve of gate valve



■ Fig. 5 – Front view of cetane valve



■ Fig. 6 – Rear view of cetane valve

Table 1 – Cetane Numbers, Room Air Conditions, Compression Conditions, and Air-Fuel Ratios at Various Positions of the Cetane Valve

Cetane No.	Room Air Conditions			Compression Conditions			Air-Fuel Ratio
	P_r	V_r	T_r	P_c	V_c	T_c	
25	14.55	0.0714	540	395	0.00827	1700	32.9
30	14.55	0.0625	540	345	0.00827	1695	28.8
35	14.55	0.0555	540	305	0.00827	1688	25.5
40	14.55	0.0501	540	275	0.00827	1688	23.1
45	14.55	0.0461	540	251	0.00827	1678	21.3
50	14.55	0.0424	540	233	0.00827	1690	19.6
55	14.55	0.0393	540	218	0.00827	1702	18.1
60	14.55	0.0375	540	205	0.00827	1680	17.3
65	14.55	0.0360	540	197	0.00827	1680	16.6
70	14.55	0.0350	540	193	0.00827	1692	16.3

cal axis when plotted as in the lower diagram.) The equation of the straight line was written in the familiar form $Y_1 = MX + K$. The difference between this straight line and the orifice curve was then plotted as a function of "X" and an equation written for this curve, of the hyperbolic form $Y_2 = \sqrt{(a^2b^2 + b^2x^2)/a^2}$. The equation of the orifice curve then becomes $Y = Y_1 - Y_2$. This holds true only within the limits of the intersections of the straight line and the orifice curve. However, this is the only portion desired because it is sufficient to measure cetane ratings from 23 to 76, the range within which most diesel fuels fall.

The resulting equation is:

$$Y = \mp (0.1161 \times + 0.1025 \sqrt{(X - 2.1)^2 + 0.5230} - 0.2160)$$

During the work of shaping the orifice, two problems became apparent. The first was that the results attained with the valve were somewhat dependent upon the speed with which the gate was moved, it being important that it be moved at a uniform speed. The second problem was the necessity of using a calibration curve, even though it was a straight line, because of daily engine variations which influenced its slope. These two problems led to the third design of the valve.

In this design, Figs. 5 and 6, an electric motor and appropriate drive mechanism were added to move the gate at a slow, constant speed across the orifice. This addition effectually solved the first problem. The second problem was answered by the addition of a calibrator. This is a small shutter which varies the total area of the orifice. At the lower limit of the instrument, its effect is negligible but, as the upper limit is approached, its influence increases. The calibrator, therefore, varies the slope of the calibration curve, thus allowing a scale of cetane number to be placed on the instrument. The scale was made adjustable to allow calibration at the lower limit of the instrument. To set the instrument for use, it is but necessary to fix the scale in position with a low rating fuel blend and set the calibrator with Shell Secondary Reference Fuel, a job of about fifteen minutes. Cetane ratings can then be determined directly by the instrument.

At this point, it was decided to investigate the effect of the cetane valve on the ignition of the fuel. This took the form of observing the changes in the compression conditions at various positions of the gate.

The quantity of air flowing to the engine through the cetane valve was measured by nozzles made from the information supplied in the paper by Sanford A. Moss: "Measurement of Flow of Air and Gas with Nozzles," submitted at a meeting of the American Society of Mechanical Engineers in December, 1927.

The compression pressure was measured by an indicator of the balanced-diaphragm type. During the readings taken with this instrument the fuel was shut off and the engine motored for as short an interval of time as was possible in order to approximate closely the actual running temperatures of the cylinder.

The results are tabulated in Table 1, the temperatures and pressures being in absolute units. Column 7 of the table shows a maximum deviation of 1% from the average, which would indicate that the compression temperature is not affected by the cetane valve, this being based upon the assumption that the error in the observations might lead to the 1% discrepancy. Therefore, the cetane valve must affect the fuel ignition by varying the compression pressure and the air-fuel ratio. The latter was found to play no determinable part in the ignition quality rating. This was investigated by varying the amount of fuel injected and noting any change in the rating of the fuel. Finding no definite influence, it was concluded that the air-fuel ratio was a very minor factor at best. The conclusion to be drawn is that the cetane valve rates diesel fuels solely by virtue of its control over the compression pressure. The compression pressure is plotted as a function of cetane rating in Fig. 7. The curve can be interpreted as representing the compression pressures at which the fuels of various cetane ratings will begin to misfire.

This conclusion led to the belief that the valve could be adapted to engines of different strokes and bores. Such an adaptation required that the orifice be capable of variation. To accomplish this, the equation of orifice area as a function of gate position was obtained by integrating the previously found equation for the orifice boundary curves.

$$\text{Area} = 2\int (0.1161 \times + 0.1025\sqrt{(x - 2.1)^2 + 0.5230} - 0.2160)\, dx$$

1. $\int 0.1161x\, dx = 0.1161x^2/2$

2. $\int 0.1025 \sqrt{(x - 2.1)^2 + 0.5230}\, dx$

Let $z = (x - 2.1)$ Then $dz = dx$

We have then

$$\int 0.1025 \sqrt{z^2 + 0.5230}\, dx = \frac{0.1025}{2}\left(z\sqrt{z^2 + 0.5230} + 0.523 \text{ Sin } h^{-1}\frac{z}{0.5230}\right)$$

Substituting $(x - 2.1) = z$ we get

$$\frac{0.1025}{2}(x - 2.1)\sqrt{(x - 2.1)^2 + 0.5230} + 0.0536 \text{ Sin } h^{-1}\frac{x - 2.1}{0.724}$$

3. $-\int 0.2160\, dx = -0.2160x$

Adding the three integrals, we arrive at the expression for the area.

$$A = 0.1161x^2 - 0.4320x + 0.1025\,(x - 2.1)\sqrt{(x - 2.1)^2 + 0.5230} + 0.0536 \text{ Sin } h^{-1}\frac{x - 2.1}{0.724} + C$$

The value for the constant of integration C was found to be 0.5723 when applying the condition of Area $= 0$ when $x = 0$.

This equation, when divided by a constant K represents the contour of a gate moving across a rectangular orifice of width K, the area varying with respect to gate position the same as for the case of the straight-edged gate and curved orifice. The advantage gained by this design, Fig. 8, was the use of an orifice which could be easily adjusted to suit the conditions of various size engines.

In order to test the applicability of this design to an engine of smaller bore and stroke, a test was carried out on the 3¾ x 5-in. Caterpillar test engine. Satisfactory operation was attained at a speed of 1600 rpm and a compression ratio of 13¼:1. Under these conditions, Shell Reference Fuel blends gave a variation of less than one number over the entire range of the instrument.

The decision was then made to test the operation of the cetane valve on the CFR test engine. Here the problem of definitely fixing the misfire point arose, a difficulty which is not present in the larger bore engine. In this engine the misfire point is definite, being signaled by a puff of visible smoke accompanied by an audible noise from the exhaust tap. However, in the CFR engine, the throttling of the cetane valve causes combustion to take place intermittently for a considerable period around the misfire point, thus eliminating either means of determination. However, work is being done on this problem.

Table 2 – 1936 CFR Fuel Ratings and Caterpillar Ratings Made on 6⅛ x 9¼-In. Test Engine

Fuel Designation	CFR Rating	Caterpillar Rating
R	29.6	31.3
S	30.5	29.3
T	39.7	41.5
U	37.7	36.9
V	37.4	38.0
W	51.6	51.3
X	51.1	50.2
Y	61.1	61.1
Z	68.7	66.0

Discussion of Ratings

The fuels listed in Table 2 were rated by the thirteen laboratories participating in the work of the CFR Committee in 1936. The ratings as made by the Caterpillar laboratory were with the butterfly valve on a 6⅛ x 9¼-in. test engine. It was largely on the basis of these results that further work was done on the throttle valve.

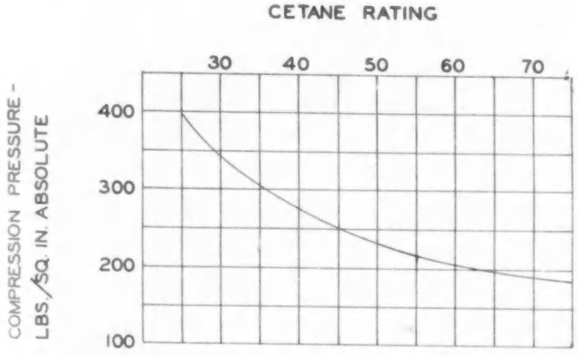


Fig. 7 – Compression pressure versus cetane rating

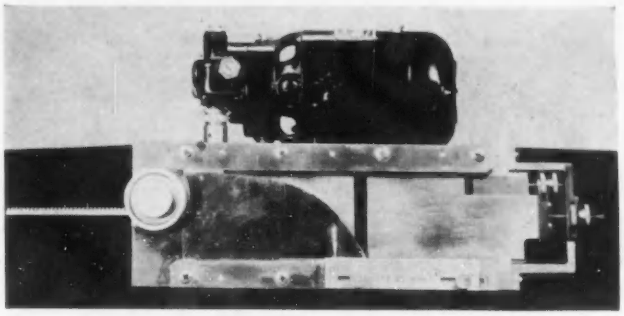


Fig. 8 – Master cetane valve

The first 12 fuels of Table 3 will be recognized as those submitted by the CFR Committee in 1937. The remaining fuels in the table are monthly exchange samples which were secured for rating by the Caterpillar Laboratory from one of the laboratories participating in this work. All the ratings by the cetane valve were made on a 5¾ x 8-in. test engine from August, 1939, to May, 1940, using the valve shown in Figs. 5 and 6.

On Table 4 are listed the fuel ratings made with the cetane valve of Fig. 8 used with the 5¾ x 8-in. test engine. The first four fuels are those used by the Full-Scale Engine Committee, the rest being monthly exchange samples.

Conclusions

Analysis of the tables of ratings shows that the correlation between the cetane valve and the coincident flash method is very good. The ratings by the cetane valve are higher by approximately one number. The ratings of the pure cetane blends show the standardization of the instrument to be very good. The average arithmetical deviation for these blends is 0.30, while the algebraic deviation is +0.05.

The reproducibility of the instrument is shown by the 0.06 algebraic deviation and 1.43 arithmetical deviation of the two series of ratings with the cetane valve. These ratings were made at different laboratories by different operators.

The simplicity of the instrument is quite obvious. It is of simple design with a minimum of adjustments required. The construction is rugged, which reduces greatly any possibility of operation failures.

The speed and ease of operation can best be realized by the fact that a single operator can easily rate from six to eight fuels an hour.

As far as the accuracy of the instrument is concerned, some absolute standard must first be agreed upon before any method can be discussed from this standpoint. All the present methods of ignition-quality rating can be standardized with reference fuels and still give differing results for commercial fuels.

At present the cetane valve can be applied to the 5¾ x 8-in. Caterpillar test engine with most satisfactory results. Although a number of applications on other engines have not been made, the work on the 3¾ x 5-in. engine would indicate that it can be adapted very easily. As far as the CFR engine is concerned, further work on a method of

Table 3 – CFR Ratings and Caterpillar Valve Ratings Made on 5¾ x 8-In. Test Engine

Fuel Designation	CFR Rating	Caterpillar Valve Ratings: Caterpillar Laboratory	Caterpillar Valve Ratings: Other Laboratories
1	55	55.4	
2	55	56.9	
3	68	68.4	
4	41	43.7	
5	54	57.6	
6	47	48.5	
7	43	44.3	
8	43	45.0	
9	34	34.7	
10	62	64.1	
11	63	65.0	
12	51	49.6	
D-26	53.5	56.5	55.9
D-27	36.8	40.2	40.1
D-28	43.1	43.2	43.4
D-29	55.8	55.0	56.0
D-32	48.7	50.1	52.1
D-33	46.3	46.9	48.4
D-34	69.2	69.6	71.3
D-35	42.3	43.3	44.5
D-36	49.1	49.5	51.4
D-37	29.5	29.6	30.9
D-38	31.9	34.4	34.9
D-39	49.4	51.2	54.8
D-40	44.5	47.8	48.9
D-41	46.3	46.9	46.2
D-42	46.5	47.7	48.2
D-43	46.9	48.0	46.3
D-44	49.9	50.9	48.7
D-45	42.5	44.5	43.4
D-46	64.3	65.1	60.4
D-47	60.6	60.8	59.5
D-48	42.7	43.6	43.0
D-49	36.6	38.2	37.5
D-50	56.1	57.1	54.0
D-51	33.2	34.4	33.2
D-58	72.3	72.6	
Pure Cetane Blends			
35.0	....	....	35.8
45.0	....	....	44.4
55.0	....	....	54.8
65.0	....	....	64.95
70.0	....	....	69.4
72.5	....	....	72.85

Table 4 – Caterpillar Ratings Made With Cetane Valve of Fig. 8 on 5¾ x 8-In. Engine, and CFR Ratings

Fuel Designation	CFR Rating	Caterpillar Rating
A	56	56.0
B	40	41.8
C	33	34.0
D	41	44.0
D-62	36.2	38.8
D-63	56.6	56.5
D-64	53.2	52.5
D-65	44.4	47.0
D-66	63.1	61.0
D-67	53.1	51.5
D-68	51.4	52.0
D-69	55.9	57.5
D-70	56.7	57.0
D-71	46.1	47.0
D-72	56.3	55.0
D-73	43.9	45.0

determining the misfire point with reasonable accuracy may allow the use of the valve with this engine.

APPENDIX

Testing Procedure for the Caterpillar Cetane Valve

INSTALLATION:

Fig. 9 shows the manner of connecting the sample tanks to the fuel injection pump.

OPERATING CONDITIONS:

1. Engine Speed .... 900 ± 3 rpm
2. Jacket Water Temperature .... 175 ± 1 F
3. Crankcase Lube Oil .... SAE 20 (approved oil)
4. Lube Oil Pressure .... 25-30 lb per sq in.
5. Valve Clearance .... 0.010 in. intake, 0.015 in. exhaust
6. Injection Timing (cut-off) .... 17.5 deg BTC
   Lift – 0.1575 in., Tappet Height – 3.7106 in.
7. Fuel Injection Quantity .... See (b) "Procedure"
8. Engine .... Standard 5¾ x 8-in. Caterpillar one-cylinder diesel test engine.
9. Speed Control .... A direct-connected 15-hp induction motor is advocated because of its inherent speed-regulation qualities.

PROCEDURE:

(a) Starting and Stopping the Engine – Motor the engine at 900 rpm until the lubricating-oil pressure exceeds 25 lb per sq in. Give the engine compression and admit fuel, allowing the motoring unit to remain active to maintain a close control of the speed.

The engine should be run for 1 hr with the jacket-water temperature at 175 F before any tests are made.

Caution: In stopping the engine, shut off the fuel before stopping the motoring unit.

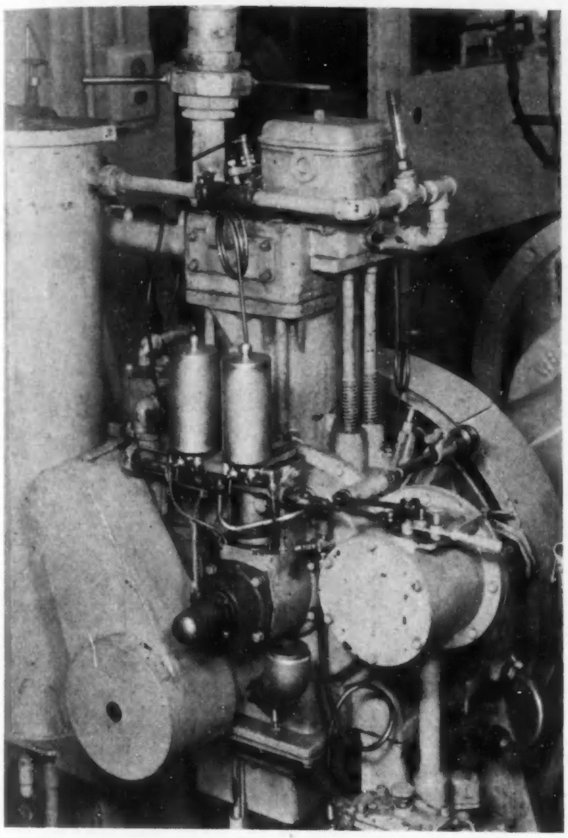
(b) Injection Pump – Set the rack by means of the micrometer so the rate of injection is 37 to 37.5 cc/min of Shell high-cetane reference fuel at an engine speed of 900 rpm. This rack setting is maintained for all tests. (This setting can be made by using a burette.)

(c) Procedure in General – The general procedure in using the instrument is to move the gate across the orifice by means of the electric motor until a misfire occurs, as evidenced by the first puff of smoke from the exhaust tap (an opening in the exhaust stack). When the instrument is calibrated, this misfire point gives the cetane rating of the fuel.

(d) Calibrating the Instrument – The first time the instrument is calibrated, cetane and alpha methylnaphthalene should be used and the secondary reference fuel checked. Subsequent calibrations may be made using the secondary reference fuel.

1. Low Cetane Number Point – With the engine operating on a reference blend of 30 cetane number and the calibrator at mid-position (open 5/32 in.), make ten gate travels to misfire and determine the average of the ten readings. Move the gate until the pointer corresponds to this average and then move the scale until 30 is opposite the pointer. The position of the scale is now fixed.

2. High Cetane Number Point – With a reference blend of from 70 to 75 cetane number in the engine, open the



■ Fig. 9 – Sampling tank installation

calibrator to its maximum position, move the gate to the correct cetane number (the cetane number of the reference blend being used) and close the calibrator until misfire occurs. Fix the calibrator in place at this point, return the gate to open position and make ten travels, recording the corresponding cetane number at each misfire. If the average falls within $\pm$ 0.5 cetane numbers of the correct rating, the calibrator setting is sufficiently accurate.

3. Intermediate Point – As a further check, although not absolutely necessary, make ten travels of the gate with a fuel of about 50 cetane number. If the average falls within $\pm$ 0.5 cetane number of the correct rating, the two previous points were correctly fixed. If the average does not fall within this limit, recheck the calibration of all three points.

(e) Testing – Rinse the two sample tanks thoroughly from the main fuel supply before introducing the fuel to be tested. Two tanks are provided so that one may be draining while the other is in use.

After the fuel under test has been placed in the tank and admitted to the injection pump, allow 2 min before operating the instrument to be certain that the system is free from the previous fuel.

The gate is operated by pressure on the button engaging the clutch. At the first puff of smoke signaling a misfire, remove the pressure on the button immediately, record the corresponding cetane number, and return the gate to its open position.

Repeat this procedure at 10-sec intervals for ten readings and average them, reporting this average to the nearest 0.5 cetane number.

Truck and Trailer Brake Equalization

BECAUSE of the tonnage of freight hauled over the highways by truck and semi-trailer combination units, the subject of brake balance and adjustment of the entire combined unit is a matter of utmost importance from the standpoint of operating efficiency. A study of operating units has shown three highly important considerations:

First and primarily, a braking system which is adequate in design and capacity for the rated load of each individual vehicle unit.

Second, the proper adjustment and maintenance of the brakes of the entire train so that each bears its proper share of the braking load.

Third, operation control so that concentration of the brake load is not too heavy upon any individual drum or axle in relation to the braking efforts on other units.

Neglect of any one of these three considerations will nullify the benefits of the other two.

It is possible to have a train which is properly balanced and equalized throughout but, because of the tendency of the operator possibly to apply the power brake system of the trailer for the major part of his braking requirements, through independent means, excessive concentration of braking is carried by the trailer axle and drums. Consequently, an adverse set of circumstances is set up which creates overheating and excessive wear.

In exploring this subject, we have conducted considerable field research which clearly indicates the reasons for troubles where they have existed and the cures for these difficulties. Where the recommendations resulting from these studies are carried out, brake difficulties have been corrected in many cases and, in others, completely eliminated, and all of the added efficiency which should be expected by the full utilization of the brake equipment available on both tractor and trailer has been gained.

The conditions under which trucks and trailers are operated at the present time make it imperative that certainty and ease of control be developed to the maximum degree possible.

The functions of control are divided as to: (a) the application of power for acceleration and sustaining speed against the forces acting against the vehicle; (b) a braking system to reduce the speed, to stop the vehicle and to hold it at rest; (c) a steering system to enable the operator to pilot the vehicle safely along the road.

It seems peculiar that, of these three functions, between which it is hard to distinguish as to importance from the standpoint of safety, the one that has received most consideration from regulatory bodies is the function of braking. Furthermore, it seems equally peculiar that all regulations are pointed toward requiring the maximum obtainable braking capacity. This is in spite of the fact, which is well-established, that extremely effective braking will cause serious interference with the function of steering and thereby will of itself adversely affect safety on the road. There is no function of the truck and trailer which operates under such widely and constantly varying conditions as the brake system.

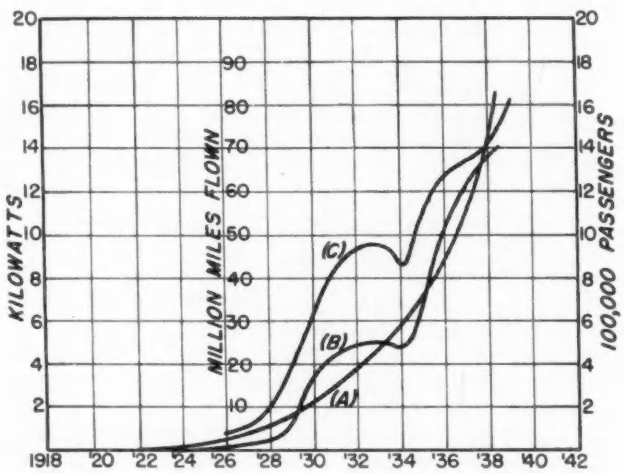
Excerpts from the paper: "Brake Equalization Between Truck-Tractors and Trailers," by John W. Votypka, chief engineer, Fruehauf Trailer Co., and E. Vance Howe, Bendix-Westinghouse Air Brake Co., presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 7, 1941.

AIRCRAFT ELECTRICITY as the

THIS paper shows that the growth of electrical demands on planes is due mainly to the problems encountered in operating the planes and, hence, electrical systems are the chief concern of the plane operators. Existing electrical systems are discussed from the standpoint of the operator, under the headings of "trouble," "weight," and "facilities."

A new system is described for the plane of the future, which is assumed to have a gross weight of about 100,000 lb, and a power demand of 30 kw. By means of variable-frequency systems a considerable weight saving is forecast.

DURING the first World War the makers of the Liberty airplane engines, unable to secure satisfactory magnetos for ignition, resorted to so-called battery ignition and installed a small storage battery and suitable generator. Electricity supplied by this means had as its sole purpose the operating of the plane, but later there were added a few instrument lights. Electricity used for this latter purpose



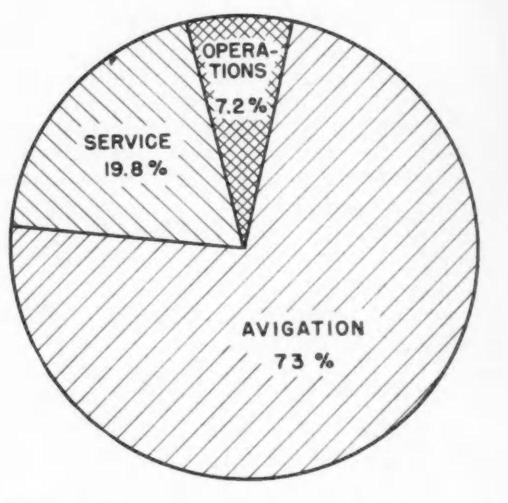
■ Fig. 1 – Trends over the last twenty years of:

A – Kw of power used per plane
B – Passengers carried per year
C – Miles flown per year

assisted the plane in its movement between two points. Later planes again made use of magneto ignition but the demand for electricity on board the plane continued to grow. The growth of this demand is shown on curve *A* of Fig. 1. Curve *A* is not intended to show the kilowatts of power actually drawn, but the total electrical power that would be drawn if all electrical loads on an airplane were simultaneously connected to the electrical power source. Considering only the years of commercial flying between 1929 to date, it will be seen that total electrical loads on the plane grew from 2 to over 15 kw.

■ Electricity on a Modern Plane

As previously stated, the plane's electrical system is no longer used for ignition; however, electrical power is still required in order for the plane to demonstrate successful flight. Electricity used for this purpose is termed "operations" electricity. The various apparatus which consume



■ Fig. 2 – Use of power on DC-3 plane (based on 5-hr flight)

electricity for operational purposes on the Douglas DC-3 plane are listed in Table 1, together with the estimated amount of power they would consume on a typical 5-hr flight. It can be seen that, while the starters take more current than other devices, the total power taken by them during a typical flight is not high. However, some planes also use electric landing gear and flap motors which would make the total power of Table 1 greater, but it so happens that the DC-3 performs these functions hydraulically. High drain for short periods characterizes this type of load, so this perhaps accounts for Douglas' choice.

Table 2 shows the electrical loads which a DC-3 uses to

e Airline Operator Sees It

by P. C. SANDRETTO

Superintendent, Communications Laboratory, United Air Lines Transport Corp.

■ Fig. 3 – Reconstructed regulator unit using selenium rectifiers

provide means for successfully avigating between two points. Electricity used by devices in this class is termed "avigation" electricity. While the instantaneous drain of these devices is not great they are in continuous operation during a trip and hence they are electrical high power consumers.

Table 3 shows the apparatus installed on board plane for convenience to passengers. Electricity drawn by these is designated as "service" electricity. Like the "avigation" loads, these "service" loads are characterized by long periods and low drain.

On Fig. 2 is shown how the total electrical power is utilized during this sample flight. Only 7.2% of the electricity is used for actually operating the plane while 73% is used to enable the plane to reach its destination. Even "service" electrical demands are greater than those of "operations."

Referring back to Fig. 1, curves *B* and *C* show the passengers carried per year and the miles flown per year. The relation between these factors and the electrical demand is evident. Since the largest portion of the power is used to make the airplane travel between two points, the increase in miles flown can be equated directly to the "avigation" electrical demand while the increase in the number of passengers carried accounts for the "service" electricity.

During the period from 1929 to 1938 the main engine horsepower increased by a factor of about 14 and plane gross weight increased about the same factor, but the "operations" power hardly tripled. It follows, then, that, since flying these passengers and planes is the business of the airlines, the plane's electrical system is primarily their concern. A plane's lift, airworthiness, and speed, primarily the concern of the plane manufacturer, can easily be demonstrated with only 7.2% of the electrical power carried.

Of course, every airplane manufacturer maintains an electrical department, but the interest of this department centers chiefly around the execution of customer demands as contrasted with the airline interest to be discussed later.

■ Airline Interest

The interest of the airlines in electrical systems can be classified under three headings. These are: troubles, weight, and facilities. The reason for their interest in the first two phases is evident, but the reason for interest in the last may not be so evident. As might be expected their interest in "facilities" ties in closely with the demand for the all-important avigation instruments which signify to the airline operator those two vital elements of his business – *schedules completed* and *safety*.

These complicated and ingenious apparatus have utilized practically every physical phenomena known to man in order to extend his senses so that he may know his position, attitude, the safe path to follow, and the condition miles away from him when all about him there are conditions which restrict his physical sight and hearing. It is these instruments that have demanded both a-c and d-c voltages varying from 1.5 to 1000 v. Further, a glance at the equip-

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 31, 1940.]

Table 1 – Operation Electricity Douglas DC-3 Plane

	Amp	Amp-Min
Warning lights	1.2	380
Landing-gear warning lights	0.7	210
Engine starters	260	520
Fuel and oil pressure warning lights	1.1	66
Booster coils	3	6
Total	......	1162

ment not only in the laboratory, but which engineers are anxious to place on the planes, shows demands for 10,000 v.

We will now pass from this subject and consider the electrical systems actually used.

The DC-3 Electrical System

While the Douglas DC-3 is considered by the public as the plane of today, to the engineers it is the plane of the past because it rolled out of the factory in 1936.

The 28 items of apparatus drawing current and the value of this current for this plane has been described earlier, but little has been said regarding the source and character of the electrical supply. This plane, today's most common United States commercial transport, is all-metal, with 21 seats and a provisional gross weight of about 24,800 lb. Its usual source of electrical power consists of two 12-v d-c generators, each capable of 50 amp continuous load and two 65-amp-hr storage batteries. The generators are geared directly to the engines and run at 1½ times engine speed. They feed two independent electrical buses, and by means of a series of switches either of the generators or batteries can be connected to either or both buses. In the usual circuit, the generators are not operated in parallel. These storage batteries are rated on the basis of a 5-hr discharge rate; hence, on this basis the power supply produces 126 amp or at 12 v 1.5 kw for a period of 5 hr.

Controlled voltage output is obtained by the use of a Tirrell-type regulator which has a current-controlled relay in addition to a voltage-controlled relay. This current relay limits the maximum drain. In the same container with these two relays is a third unit that serves as an inverse current relay which prevents the battery from discharging through the generator when the generator voltage is low.

Except for the use of two generators, batteries, and buses, the system as described is essentially that which was designed and put into use by the Air Corps in about 1925. The old Liberty system had used 8 v, but the desire to lighten conductor weight occasioned by landing lamps installed for night flying prompted an increase in voltage. Subsequently most aircraft electrical units were designed for 12 v, and this standard was in use at the time of design of the DC-3. The systems designed prior to the advent of the DC-3 had made use of a single generator and battery, but the increased electrical demand of the DC-3 made another generator and battery necessary.

If discussion of airline electrical-system interest is begun by asking any airline maintenance man about his electrical troubles, he will answer, "regulator." The vibrating-contact type of voltage regulator which has seen extensive use in the past suffers continually from contact points which either "freeze" or fail to make good connection. High-current, low-voltage contacts mounted on a vibrating structure are as certain a source of trouble as any electrical engineer can conceive.

Regulators are usually changed at 100-hr intervals. United Air Lines has been attempting to make them operate without adjustment for one engine change period (625 hr) and has made several changes in their design. One of these changes consisted in replacing the inverse current relay with selenium rectifiers. A reconstructed regulator unit using those rectifiers is shown on Fig. 3. Note the shock-mounted current and voltage actuated regulators. The unit of Fig. 3 is used for controlling two generators. Assured of an inverse current device which cannot have frozen contacts, the two bus systems were paralleled with the drop through the rectifiers serving as an equalizer bus. This system has been in operation for about six months and has given fair service. The heavy load demanded

Table 2 – Avigation Electricity Douglas DC-3 Plane

	Amp	Amp-Min
Windshield defroster fan	1.0	300
Argon dynamotor and/or instrument spotlight	1.5	450
Pilot heater	14.0	4200
Running lights	2.5	750
Compass, gyro and radio panel lights	1.0	300
Instrument panel lights	1.0	300
Radio - receiving dynamotor and receiver filament	7.3	2190
Electrical instruments	1.0	300
Wing de-icer	1.5	450
Transmitting dynamotor	60	900
Transmitting filaments	15	450
Landing lights	70	1050
Baggage pits	2	90
278-kc relay	0.4	2
Total	......	11732

from these generators accounts in a measure for the regulator trouble. The parallel system assuring approximately equal load division helps to minimize the individual generator drain. It is not believed, however, that this system can, by our standards, yet be classified as trouble-free. Generators have been satisfactory in the past, but they have never been entirely free from brush troubles.

Any system which uses a storage battery is heavy because the storage battery is inherently a low-efficiency device. The generators described weigh 30 lb each, while the batteries weigh 65 lb each. Without considering installation weight or regulators, the total supply source weighs 190 lb, or about 127 lb per kw on a 5-hr continuous-drain basis. The use of low voltage requires high current in order to transmit appreciable power. High currents, in turn, require large-diameter conductors in order to keep voltage drop along the conductors small. Fig. 4, computed for a plane about the size of a DC-4, shows conductor weight as a function of voltage. The curves consider 100% weight as that of a 12-v system, and it can be seen that the weight would fall to 35% of the 12-v value if 40 v were used.

There is hardly a more inflexible power system than that utilizing 12 v d-c unless it is one making use of lower d-c voltage. Their high voltage drops preclude the use of thermionic devices with this supply, and voltage changing

must be accomplished either by heavy rotating machinery or power-consuming resistors. Occasionally vibrators have been employed as an a-c source, but the filters necessary to keep the "hash" they generate out of audio circuits, together with their required maintenance, have not made them a very satisfactory device.

In fairness to storage-battery systems, their virtues must be described. It is obvious, of course, that a storage battery makes electrical power available without the main engines operating and offers a supply of the power for a limited period. It also greatly acts as a reservoir for momentary loads, thereby reducing the generator capacity required. The total electrical load which is listed in Tables 1, 2, and 3 amounts to 5.6 kw, yet the actual electrical power supplied is only 1.5 kw on a continuous basis. The reason for this is that a large portion of the load, while it has heavy demands, has short-time demands and a storage battery can supply a tremendous amount of power for a very short period. If 5.6 kw of power were to be supplied by generators similar to those used in the DC-3, the generator weight would be about 270 lb as contrasted with a system weight of 190 lb. Of course, this weight could be reduced appreciably by limiting the operation of starters to only one at one time. It should be understood that this is purely a theoretical discussion to show the facilities of a storage battery without attempting to describe how the generators would be operated when the engines are idle. Of course, hydraulic means could be used instead of the storage battery, but no attempt will be made in this paper to discuss the relative merits of a hydraulic versus an electrical system.

Aside from the foregoing advantages offered by the storage battery, the DC-3 electrical system can be characterized by stating that it is troublesome, heavy, and inflexible.

■ The DC-4 Electrical System to Be

When the Douglas DC-4, a four-engined plane, makes its appearance on the commercial airways of the United States sometime during the spring of 1941, it will have a provisional gross weight of 55,000 lb and will carry 40 passengers and a crew of 5. An earlier model of this plane was test-flown in 1938, so this is really the airplane of today. It has many more electrical units than the 28 listed for the DC-3. For example, 11 radio units will replace the 4 used on the DC-3 and the total of its electrical loads amounts to about 15 kw. For power, it will use four 24-v generators operated in parallel, each having a power capacity of 3 kw. A single 24-v 65-amp-hr battery is used to start the engines and supply radio from the time of loading of the plane to the time of its take-off. Generators are designed to operate from the main engine with a 3:1 gear ratio.

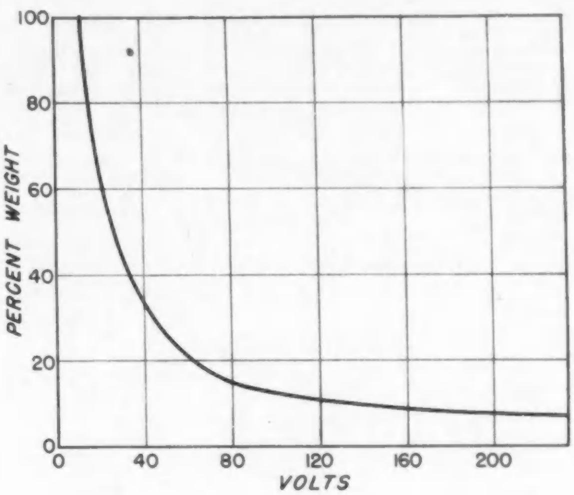
Table 3 – Service Electricity Douglas DC-3 Planes

	Amp	Amp-Min
Cabin side lights	6.2	1860
Buffet lights	1.0	300
Cabin warning lights	2.5	100
Seat belt warning	1.2	360
Companionway domelight	1.0	45
Stewardess call light	1.0	5
Cabin light relay	0.5	30
Cabin dome lights	6.2	372
Entrance door light	1.7	105
Total		3177

Reference to Fig. 4 shows why a higher voltage than 12 was chosen for the DC-4 plane. A 24-v system, however, was chosen because:

1. Military interests began using this voltage and certain 24-v d-c apparatus became available.
2. The Radio Technical Subcommittee No. 3 on power supplies adopted this voltage as standard in 1938.

The reason that military interests chose 24 v is probably due to the fact that this is an exact multiple of 12 v; hence,



■ Fig. 4 – Comparison of plane wiring weight/voltage

two of the standard batteries could be used. The reason for choosing a voltage with a magnitude in the vicinity of 24 v is described in a paper by Vernon H. Grant of the Navy and Dr. Melville F. Peters of the Bureau of Standards published in the October, 1939, issue of *Electrical Engineering*. This paper is a classic on the subject of optimum voltages for aircraft. The military services, faced with the problem of selecting an optimum voltage, not for a single plane but for a variety of planes ranging from a single-seat fighter to a twin-engine bomber, had to select a voltage optimum for the entire range.

Table 4 was compiled from data taken from the paper mentioned and shows that a 24-v system would be only 5 lb heavier than a system employing the optimum voltage when the plane is a single seater, and 70 lb heavier than the optimum system when the plane is a large, twin-engine type.

Since the system designed for the DC-4 has not been flown, we can only forecast its possible troubles. An attempt was made (within the limits of the 24 v) to provide a voltage regulator free from trouble. The regulator employs a voltage-sensitive element which actuates a motor-driven field rheostat. By this means, the sensitive element is not charged with the necessity for actually doing work; hence, it should be more trouble-free. A small selenium rectifier is used as an element sensitive to inverse current and actuates a large relay which stops reverse current flow. The current limiter is eliminated. It is felt that this regulator design should be much more trouble-free, particularly since the use of a distributed field winding in the genera-

tor supplies regulation for current changes of reasonable magnitude.

Since these generators will be operated at about 100 amp and at twice the speed of the present units, more brush troubles may result; however, imperceptible sparking made possible by the field winding previously mentioned may serve to offset higher current and speed and make for good brush life.

An attempt to reduce weight has been made by paring down battery size to an absolute minimum. The 65-amp-hr batteries may be insufficient, and space has been provided for additional units if they are found to be necessary. By increasing speed, generator weights have been cut to 28 lb each. This makes the total weight of the power-supply system about 242 lb, or 19.8 lb per kw as contrasted with 127 lb per kw for the DC-3 system.

Table 4 – Comparative Weights of Electrical Systems

Weight of Cable and Conduit of 12-V System, lb	Increase in Weight if 24 V Is Used Instead of Optimum Voltage Given, lb	Increase in Weight if 12 V Is Used Instead of Optimum Voltage Given, lb
	Sport Type	
10	5	1
	Single-Seat Type	
20	2	5
	Two-Seat Type	
50	0	25
	Three-Seat Type	
70	0	40
100	3	65
	Small Twin-Engine Type	
150	10	110
200	15	155
	Large Twin-Engine Type	
300	30	245
400	50	340
500	70	435

In Table 4 it will be seen that, as the size of the plane increases, the conduit and wiring weight for a 24-v system becomes appreciably heavier than the weight of conduit and wire for a system utilizing the optimum voltage. The weight of conduit and wires for the DC-4 plane is 542 lb. Using the methods of Grant and Peters, it may be computed that this system is 170 lb heavier than the optimum system, but 740 lb lighter than a 12-v system.

The fact that 24 instead of 12 v are used makes little difference in the system flexibility. The increased use of Autosyn and Telegon indicators, as well as fluorescent lighting, will require many rotating machines or equivalent converter.

To summarize, the DC-4 has an electrical system which should give less trouble and weighs proportionately much less than the DC-3 system. It, however, is still inflexible and increased weight of the distribution system will probably prohibit its use on larger planes.

■ A-C Systems for Aircraft

Beginning about 1934, both military and commercial interests considered the use of high-frequency, high-voltage a-c power for use on aircraft. Apparatus was designed and installed on certain military crafts, as well as on the experimental model of the DC-4.

A Technical Subcommittee was formed under the then-existing Bureau of Air Commerce for the purpose of standardizing on a single a-c power system. During the many stimulating meetings of this Subcommittee every possible reason why high-frequency a-c power supplies would not be practical was discussed. These reasons included the possible heating of conduits, the low efficiency of motors, and the supposedly unsurmountable hum in radio equipment. Everyone felt that the use of small constant-speed gasoline engines would constitute an entirely satisfactory prime mover for the alternator.

Apparatus for two systems was developed. One of these systems used 800 cycles, single-phase while the other used 400 cycles, three-phase. Each employed 115 v. High frequency was decided upon because transformer weight decreases with frequency and, since motor speeds increase, lower motor weight also results. After several years of discussion the Subcommittee never came to an unanimous agreement on a standard.

A single-phase, 800-cycle system consisting of two 15-kva alternators driven from 30-hp, 4000-rpm gasoline engines was installed and tested in the DC-4. The electrical apparatus performed very well. It was found to be both light and trouble-free. At first, however, considerable trouble was experienced with the gasoline engines. This trouble, due partially to a cramped installation, was cleared up later in a satisfactory manner; however, it became apparent from the standpoint of airplane operations that:

1. It was unreasonable to use a *four-motored* plane for safety in flight then predicate its safety of travel between two points on only *two* gasoline engines.

2. The cost of maintaining a small high-speed gasoline engine equipped with the necessary automatic mixture controls, speed controls, and supercharger is almost as much as the cost of maintaining a large 1400-hp engine; hence, engine maintenance cost has increased for this four-motored plane to a figure nearly equivalent to a six-motored plane.

A constant-speed hydraulic drive was designed by the Douglas Co. This drive consisted of four hydraulic pumps, one mounted on each engine, supplying hydraulic fluid under pressure which actuated two constant-speed turbines. The alternators were then driven by these turbines. Such a system might be successful; however, it is unduly complicated and would require considerable development before it would be trouble-free. Experience would indicate that constant-speed drives and hence constant-frequency a-c systems for commercial aircraft of the size now visioned are not practical.

■ System for the Plane of the Future

The plane that will supersede the DC-4 is visioned as a four-motored plane with a gross weight of 100,000 lb. It probably will carry 100 passengers and a crew of 7. Its power demand will be about 30 kw. It is suggested that four alternators, each delivering 7.5 kw of power at 115 v, be mounted directly on each engine. They should be driven at about 4000 rpm during cruise and deliver power at a frequency ranging from 300 to 900 cycles. No attempt will be made to maintain this frequency constant, but it will vary as the speed of each engine varies, except that an alternator will be disconnected when the speed of its driving engine falls below a predetermined minima. In order to eliminate the necessity for synchronizing the alternators, the total electrical load will be divided into five groups.

Each one of four groups of loads will be connected to an individual alternator bus. The fifth load group will be connected to a 24-v battery with a 65 amp-hr capacity. This battery will be charged by four rectifiers, one connected to each of the four alternator buses. This fifth load group will consist in the main of those loads indicated on Fig. 2 as operations loads, and hence constitute only 7.2% of the total electrical power demand. Certain of the loads can be connected by means of a switching system which will automatically connect them to a new bus if the power fails on their normal bus. All motors should be connected to the d-c system.

The alternators should be designed for an exciter voltage of about 500 v. This voltage will allow the use of trouble-free thermionic voltage regulators. The brushes on the exciter commutator will be for very low currents and medium high voltage, so they should not give trouble. The main power take-off has no brushes; hence, no trouble from this source.

Basing weight estimates on the equipment that was actually constructed for the DC-4, the total weight of the power-supply system should be about 450 lb or 15 lb per kw, as contrasted with 19.8 lb per kw for the DC-4 system and 127 lb per kw for the DC-3 system. There should be a saving in wiring and conduit weight over a 24-v system of about 350 lb. Transformer weights will be about 70% of those for 60-cycle transformers and the weight for auxiliary power supplies should not be more than one-half of that for rotating machinery. Maintenance cost will be lower for these power supplies and efficiency will be higher than for rotating machines.

From a facility standpoint, this system has everything to offer. Transformers can be designed easily to operate over the range of 300 to 900 cycles and supply any voltage desired. Engines can be started on the ground from the battery which could supply the minimum essential loads in event of failure of all four alternators. A-c switches can be made small to handle large power load without arc-backs. D-c power is available for variable speed and high torque motors as well as for solenoids.

Conclusion

It is understood that the Navy has done some development work with variable-frequency systems and has found them satisfactory for practically everything except motors. Other than this work, however, all information available indicates that no development is under way on the system described, but it is believed that manufacturers of planes and equipment should consider this for the electrical system of the future.

Waste-Heat Recovery with Vapor-Phase Cooling

IN the vapor-phase cooling system for internal-combustion engines the circulation is maintained through the jacket of the engine and steam is removed at a central point from where it is condensed and returned to the system. Most of the heat of the engine is carried out in the form of water and minute steam bubbles, and the heat is subsequently removed from the system in a condenser or heat exchanger. The name vapor phase comes from the fact that the heat is removed in the "vapor phase" of the fluid by changing the state of the fluid from vapor to liquid.

The possibility of operating at high temperatures opens a new field of usefulness.

The internal combustion engine is a device for converting one form of heat energy to another. The distribution of heat in ordinary internal-combustion engines follows:

Shaft hp	26%
Jackets	28%
Exhaust	34%
Friction	6%
Radiation	6%
	100%

Of the total 100% input, we are at present using only about one-quarter.

Many attempts have been made in the past, and for that matter are being made every day, to use the waste heat from engines. Most of the processes in which we use heat, however, require that the heat be available at a reasonable working temperature. For practically all industrial purposes, heat at less than steam temperature, or 212 F, is useless. However, in the exhaust gases we have temperatures above this point, and it is with this portion of the rejected heat that waste heat recovery systems have in the past dealt. Exhaust-waste-heat boilers can recover between 2¼ and 2½ lb of steam per bhp-hr. However, the apparatus for this purpose is large and costly and, due to the low temperature in certain parts, the corrosion is excessive. So it is that little use has been made of exhaust heat because of the cost and bulk of the apparatus and the relatively low heat recovery. By use of the vapor-phase system of waste heat recovery, however, it is possible with relatively simple equipment to recover approximately 50% of the total input heat in the form of steam. This, coupled with the shaft horsepower, makes a total thermal efficiency of approximately 75% possible. Another way of stating the foregoing figures is that approximately 5½ lb of steam can be recovered per bhp-hr at full load. This rate is much greater at partial loads. This steam can be delivered at pressures of between 5 and 20 lb per sq in. without alteration of the engine equipment as designed today. By taking special precaution in the engine design, it would be entirely possible to deliver steam at 125 lb per sq in. pressure. The major factors concerned in the design of the engine to operate at higher temperatures are the gasketing and the method of packing of the liner. The average liner is not equipped with gaskets to operate under pressure in excess of 20 lb. Liner type engines usually employ rubber rings at the bottom of the liner, and ordinary rubber rings tend to give trouble at temperatures above 225 F. However, by use of neoprene or duprene sealing rings, the temperatures can be raised to 350 F without difficulty.

An interesting figure to remember is that approximately one-half of the total Btu input to the engine can be recovered as waste heat.

Excerpts from the paper: "High-Temperature Cooling by the Vapor-Phase Method and Its Relation to Engine Performance and Wear," by J. H. Wallace, Pacific Enterprise Products, Inc., presented at a meeting of the Tulsa Group of the Society, Tulsa, Okla., Oct. 11, 1940.

A 13-Year IMPROVEMENT in

BETWEEN 1927 and 1940 there has been a considerable improvement in the direction of leaner air-fuel mixtures used in representative cars, amounting to an average of about two ratios at road load, equivalent to a saving of about 18% in fuel consumption, the authors state in reporting data collected in surveys of mixture ratios used in representative groups of cars in 1927, 1933, and 1940. These data, they add, also suggest that there is a possibility of still further gains.

The air-fuel ratios used in automobiles, they explain, represent the engineering compromise that must be made between the relatively lean mixtures which are desirable from the standpoint of economy of fuel, and the richer ones which are necessary because of inherent imperfections in commercial induction systems.

Comparisons of mixtures used by the cars tested in the three years are given on graphs showing average and range of air-fuel ratios plotted against miles per hour at road load and full load, and per cent of energy loss plotted against miles per hour at road and full load.

IT is the purpose of this paper to present the data which were obtained in surveys of the mixture ratios used in representative cars in the years of 1927, 1933, and 1940, respectively. These data seem of interest not only of themselves, but because they indicate a considerable and regular improvement of which automobile engineers may well be proud, and also because they suggest that there is the possibility of still further improvement.

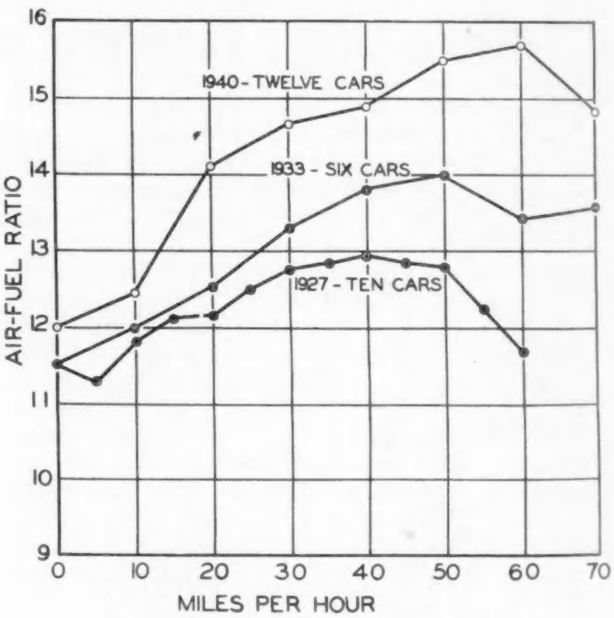
■ Compromise Necessary

The air-fuel mixtures used in automobiles are of special interest as representing the engineering compromise that must be made between the relatively lean mixtures which are desirable from the standpoint of economy of fuel and the richer ones which are necessary because of inherent imperfections in commercial induction systems.

Disregarding the matter of variations between individual explosions – which includes the differences in air-fuel ratios between different cylinders of a multicylindered engine at a given time, the differences between successive charges in a given cylinder, and also stratification of the charge within the cylinder itself – the air-fuel ratios which it is desirable to use in an automobile are fairly easy to evaluate. Under ideal conditions there is considerable evidence that the mixture ratio for maximum power comes close to that for a theoretical mixture. This is the one which burns the fuel substantially completely to carbon dioxide and water, and is about 15:1 for the average commercial gasoline. The maximum-economy mixture is generally taken to be somewhat leaner than this.

In practice, with multicylinder engines with imperfect distribution or even in single-cylinder engines under nonideal conditions, the actual observed maximum power mixture is found to be considerably richer than this, by an amount depending upon the quality of the distribution.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1941.]



■ Fig. 1 – Average mixture ratios at road load

This comes about because, as is well known, the power under ideal conditions falls off very much more rapidly with mixtures leaner than the theoretical than with richer mixtures. Consequently, in actual practice in order to have maximum-power mixtures, it is necessary to maintain the average quite rich in order that the deviations from the average shall be mostly on the rich side of the theoretical mixture and the integrated power from a number of cylinders or a series of explosions may be at a maximum.

Although it has been found quite possible to get a good idle with mixtures approaching and even leaner than the theoretical with good distribution using a gaseous fuel,

in MIXTURE RATIOS

by W. G. LOVELL, J. M. CAMPBELL, B. A. D'ALLEVA, and P. K. WINTER
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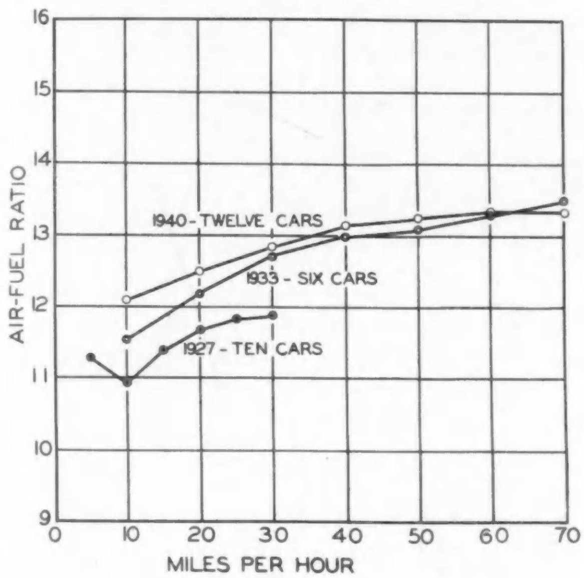
actually in commercial practice such mixtures cannot be used because of occasional missing due to non-ideal distribution. Conditions of exhaust-gas dilution and stratification within the cylinder are also factors of influence under these conditions.

Another important factor contributing to the necessity for the use of rich mixtures is the necessity of avoiding momentarily excessive leaning-out of the mixture during sudden acceleration.

The mixture ratios actually used in practice then represent the necessary engineering compromise with these and other factors, and the use of mixtures more closely approaching the theoretical or even the maximum-economy one may be taken as an indication of better distribution and other engineering improvements.



▪ Fig. 2 – Average mixture ratios at full load

▪ Method of Surveys

These three surveys were made at six and seven-year intervals, respectively, by taking a group of cars in number varying from 6 to 12 and measuring the mixture ratios used in them at both full-load and road-load steady-running conditions at intervals over the speed range.

Due to the length of time covered by the three surveys, it was obviously impossible to make them strictly comparable in all details, since at the time the purpose of making a direct comparison was not in mind. For example, some of the makes of cars in the earlier tests are no longer in production, and some of the makes in the latter surveys were not being produced in 1927. In each case, however, a group of automobiles was used which was considered at the time to be contemporarily representative. The cars were all in good mechanical condition, but no special adjustments were made for the tests. Check determinations were made, of course, on all individual determinations and, in some cases, different individual cars of the same model were used yielding results in what is thought to be good agreement.

In all cases, the determinations of mixture ratio were made by analysis of the exhaust gas. In the 1927 survey, the samples of exhaust gas were collected during actual operation of the cars on the road under carefully controlled full-load and level road-load conditions. In the 1933 and 1940 surveys, the cars were operated on a chassis dynamometer. No reason is known at present why such data as used in this paper should not be quite comparable. Samples of exhaust gas were collected on the road by tapping the exhaust pipe in front of the muffler, and those collected during dynamometer operation were obtained through a length of copper tubing extending about three feet into the tail pipe.

▪ Computation of Data

The samples of exhaust gas were analyzed for carbon dioxide, oxygen, and carbon monoxide in a portable Orsat-type of gas-analysis apparatus. In 1927 and 1933 carbon dioxide was determined by absorption in a solution of sodium hydroxide, oxygen in an alkaline solution of pyrogallol, and carbon monoxide in a cuprous chloride solution. In the 1940 survey the oxygen absorbent was a slightly acid solution of chromous chloride, and the carbon monoxide absorbent a mixture of cuprous sulfate and beta naphthol in sulfuric acid.

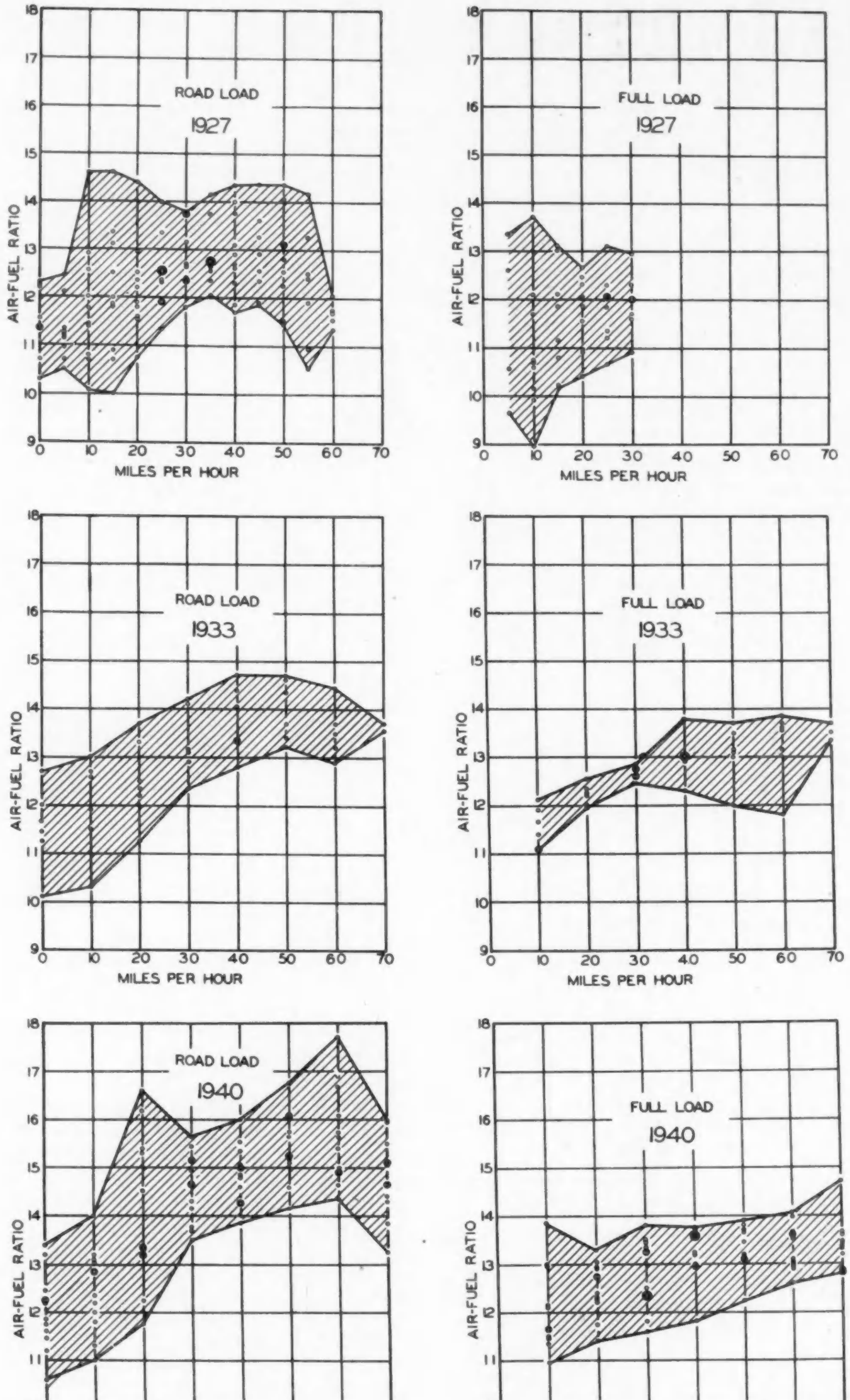
In order to relate the exhaust composition to mixture ratio, use was made of tables based upon the computed chart published by D'Alleva and Lovell some years ago[1], which was found to represent the relation between exhaust composition and mixture ratio as actually measured on several engines and with different types of air-measuring equipment to within 3%. It is thought that any errors introduced from this source are not significant for the purposes at hand.

▪ Discussion of Data

Average Values – From the standpoint of considering the data as indicating trends, it is most convenient to use the average values of the values obtained at different speeds under full and road-load conditions, respectively, for the different surveys.

Such values for road-load average mixture ratios at various speeds for the three surveys are shown in Fig. 1. It is noteworthy that the curves lie consistently one above the

[1] See SAE Transactions, February, 1936, pp. 90-98: "Relation of Exhaust Gas Composition to Air-Fuel Ratio," by B. A. D'Alleva and W. G. Lovell.



■ Fig. 3 – Range of mixture ratios

other, indicating a progressive use of leaner mixtures under these conditions. There is an improvement from 1927 to 1940 of somewhere in the range of two mixture ratios or a very considerable amount. The curve for 1927 drops down towards richer mixtures at lower speeds than do the others, and this is possibly due to the fact that these cars would not go as fast as the others and approached the rich top-speed mixture ratios at lower speeds than the others. It is also of interest that all three curves show somewhat the same general shape, richer at low speeds, as might be expected under conditions where the distribution would not be thought to be so good. These average values also show that, only in 1940, and only at speeds above 40 mph were average mixtures used which were as lean or leaner than the theoretical.

The picture of similar data under full-load conditions is shown in Fig. 2. Here the progressive improvement with time is not so marked as in the data obtained at road load, but is still possibly significant. Here again, leaner mixtures are used at higher speeds, although the 1927 data do not extend to as high speeds as those for the other years. It is only at speeds above 35 mph that even the later cars use mixtures even as lean as 13:1; and under no condition do these average mixtures become leaner than about 13.5:1. They are still quite far from the theoretical ratio of 14.7:1.

Range of Mixture Ratios

The values for individual cars are possibly not especially significant for the purposes at hand, but the range covered by the groups of cars may be of interest. Such data for all the cars incorporated in the surveys have been plotted in the group of curves of Fig. 3, representing the data grouped by surveys and also for road-load and full-load conditions. The shaded bands indicate the range for the individual points as plotted, but the upper or lower points do not necessarily represent the same car in either case.

The most outstanding feature of this group of curves is the width of the shaded area, or the range covered, which is from as little as a half to as much as almost five ratios. There is not much indication of trends in the sense of the range becoming greater or smaller. About all that may be said profitably about these ranges is that, if the worst were as good as the best, a great general improvement would result.

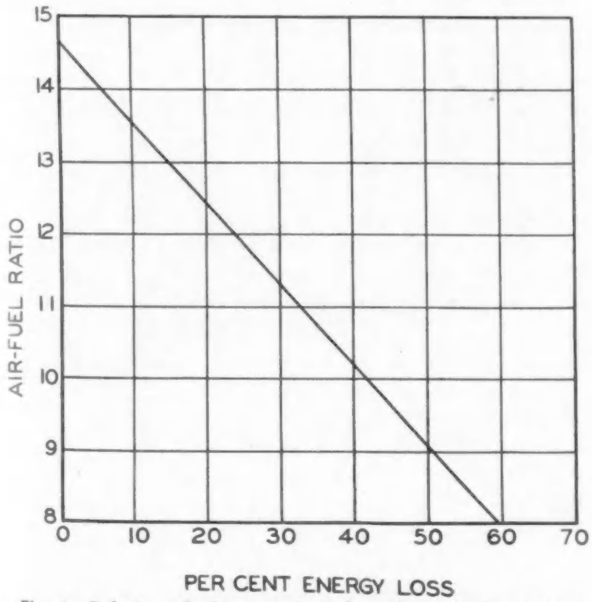


Fig. 4 – Relation of mixture ratio to loss of energy due to incomplete combustion

Losses Due to Incomplete Combustion

In spite of improvements which have been made, the use of mixtures richer than theoretical represents a loss of fuel due to incomplete combustion. This is a loss in the sense that carbon monoxide and hydrogen are present in the exhaust, and they are components which are of themselves combustible, and that more heat would have been liberated as thermal energy from the fuel had more oxygen or air been present. Such losses, as previously mentioned, are to a certain extent unavoidable, but they may be very considerable in amount.

Their magnitude may be computed directly, the details of the computation of which are given in the Appendix. This energy loss represented by unburned constituents of the exhaust may be computed as the per cent of the total potential energy of the fuel. A curve of the per cent of energy thus lost against air-fuel ratio is given in Fig. 4. Thus, while there is no such loss at the theoretical mixture ratio of 14.7:1, the losses amount to as much as 50% of the total heating value of the fuel at mixture ratios of 9:1. The elimination of such losses can thus represent a very great potential gain in economy.

How much these percentage losses actually amount to in these surveys is represented in Figs. 5 and 6. There the percentage of the fuel energy lost because of rich mixtures is plotted against speed, using the average of the different surveys for road-load conditions in Fig. 5, and full-load conditions in Fig. 6.

The road-load data show a progressive reduction in the loss with time. While an average loss over the speed range in 1927 might be taken as about 20%, by 1940 such losses were reduced to less than half of this. In fact, the 1940 average shows that, at speeds above 30 mph, less than 5% of the total potential energy of the fuel is lost due to the use of rich mixtures.

The data of Fig. 6 relating to full-load conditions are not so favorable. To be sure, there is a considerable and impressive improvement with time but, even in 1940, on the average about 15% of the fuel energy is lost due to rich mixtures under these conditions.

It is quite obvious that a very considerable improvement has been made, but that there is still room for further progress. The direct advantage of such further progress is the further reduction of such losses due to chemically incomplete combustion of rich mixtures and better economy or miles per gallon. An additional valuable byproduct of such further improvement is the further reduction of the concentration of carbon monoxide in the exhaust.

APPENDIX

Calculation of Energy Losses

The loss of energy due to incompleteness of combustion is the difference between the energy obtained under a given set of conditions and that which would be obtained if combustion were complete.

The chemical reaction for 'complete' combustion may be expressed by the general equation:

$$CH_{2R} + n\,O_2 = a\,CO_2 + b\,H_2O + c\,CH_4 \qquad (1)$$

and for partial combustion (rich mixtures) by the equation,

$$CH_{2R} + n\,O_2 = a\,CO_2 + (1-a-c)\,CO + b\,H_2O + (R-b-2c)\,H_2 + c\,CH_4 \qquad (2)$$

In these equations the fuel (gasoline) is represented by the simple formula, CH_{2R}, in which R is the molar ratio of hydrogen to carbon in the fuel. Its value has been determined[1] as 1.062 for the average gasoline. A small amount (about 0.3%) of methane, CH_4, is found in the exhaust gas from both rich and lean mixtures, and the value of the coefficient (c) of CH_4 in the chemical equation is nearly constant at about 0.02. Using these values and maintaining a material balance, Equation (1) may be rewritten:

$$CH_{2.124} + 1.492\,O_2 = 0.98\,CO_2 + 1.024\,H_2O + 0.02\,CH_4 \tag{1a}$$

There are two empirical rules[2] which are useful in computing the heat of combustion of hydrocarbons: (1) $\Delta H = -105.4$ cal/mol O_2 consumed, and (2) $\Delta H = -153.5$ cal/CH_2-group. By the first rule and Equation (1a), $\Delta H = -105.4 \times 1.492 = -157.5$ cal. To apply the second rule, we observe that the fuel molecule consists of one CH_2-group plus a little excess hydrogen, $H_{0.124}$. When hydrogen burns to form liquid water, 34.2 cal per atom is evolved. The heat of combustion of the fuel is, therefore, $-153.5 - (34.2 \times 0.124)$, or -157.7 cal. According to Equation (1a), this value has to be corrected for the small amount of residual methane, whose heat of combustion is -210.8 cal/mol, or -4.2 cal for 0.02 mol. The corrected value for the heat of combustion of the fuel by the second rule is, therefore, $-157.7 + 4.2$, or -153.5 cal, and the average value by the two rules is -155.5 cal/mol.

The heat evolved by the combustion of rich mixtures depends upon the relative amounts of fuel and oxygen in the mixture, and the heat of the reaction may be calculated from the molecular heats of formation of all the substances involved in the reaction. From Getman and Daniels[3] the following values for the heats of formation were obtained: CO_2, -94.4 cal; CO, -26.4 cal; H_2O (liq.), -68.4 cal; CH_4, -18.5 cal. The heat of formation of the fuel may be calculated from the Equation (1a):

$$CH_{2.124} + 1.492\,O_2 = 0.98\,CO_2 + 1.024\,H_2O + 0.02\,CH_4,$$
$$\Delta H = -155.5 \text{ cal}$$
$$-X - 0 - (94.4 \times 0.98) - (68.4 \times 1.024) - (18.5 \times 0.02)$$
$$\Delta H = -155.5 = -92.5 - 70.0 - 0.37 + X$$
$$X = 7.37 \text{ cal}$$

The heat of the partial combustion reaction may now be expressed in terms of the coefficients of the chemical Equation (2):

$$CH_{2.124} + n\,O_2 = a\,CO_2 + (1-a-c)\,CO + b\,H_2O + (R-b-2c)\,H_2 + c\,CH_4$$
$$-7.37 \qquad -0 \qquad -94.4\,a - 26.4\,(1-a-c) \qquad -68.4\,b - 0 \qquad -18.5\,c$$

$$\Delta H = -94.4\,a - 26.4 + 26.4\,a + 26.4\,c - 68.4\,b - 18.5\,c + 7.37 = -19.03 - 68\,a - 68.4\,b + 7.9\,c.$$

Substituting 0.02 for c,

$$\Delta H = -18.87 - 68\,a - 68.4\,b,$$

Per cent completeness of combustion

$$= \frac{100 \times \Delta H \text{ for partial combustion}}{\Delta H \text{ for 'complete' combustion}}$$

$$= \frac{100\,(18.87 + 68\,a + 68.4\,b)}{155.5} = 12.12 + 43.7\,a + 44\,b$$

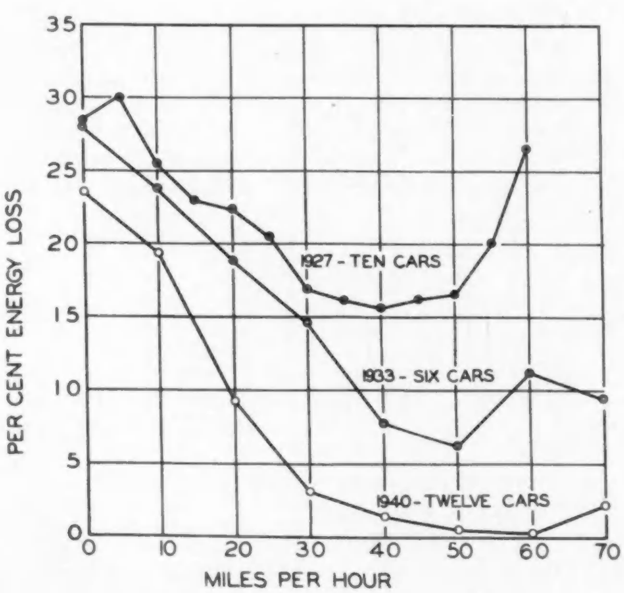
$$\text{Per cent energy lost} = 100 - 12.12 - 43.7\,a - 44\,b = 87.88 - 43.7\,a - 44\,b.$$

The values of a and b corresponding to different air-fuel mixtures have been calculated by a method similar to that of D'Alleva and Lovell[1] and are tabulated below, together with the computed energy losses. This relationship is also shown in Fig. 4.

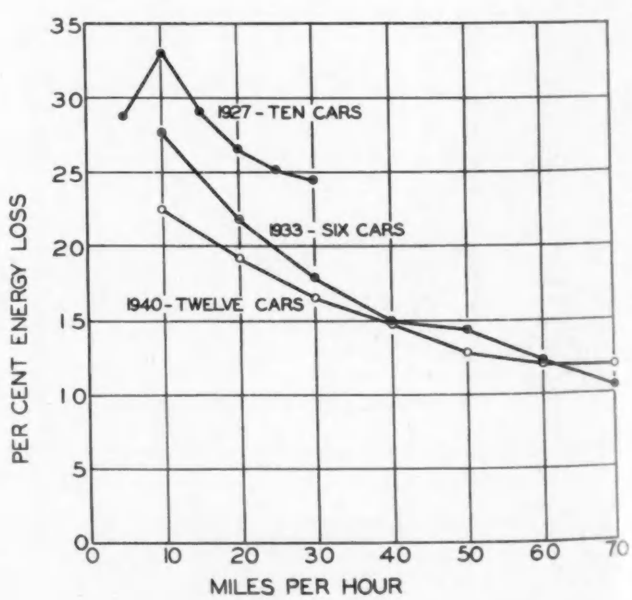
A/F	a	b	% Energy Lost
8	0.178	0.468	59.5
9	0.259	0.590	50.6
10	0.355	0.697	41.7
11	0.464	0.791	32.8
12	0.588	0.870	23.9
13	0.727	0.938	14.9
14	0.874	0.994	6.0
14.5	0.950	1.020	1.5
14.675	0.980	1.024	0.0

[2] See p. 259: "Outlines of Theoretical Chemistry," by F. Getman and F. Daniels, 5th Edition, 1931, John Wiley & Sons, Inc., New York, N. Y.

[3] See p. 251: "Outlines of Theoretical Chemistry," by F. Getman and F. Daniels, 5th Edition, 1931, John Wiley & Sons, Inc., New York, N. Y.



■ Fig. 5 – Average energy losses at road load



■ Fig. 6 – Average energy losses at full load

ENGINE DESIGN Versus ENGINE LUBRICATION

by R. J. S. PIGOTT
Gulf Research & Development Co.

DISCUSSION is given of the increase in severity of conditions for lubrication in the last few years, during which the engine designer has been calling upon the oil chemist for special oils to correct difficulties in operation. The 51% increase in horsepower in 10 years is proportioned between increase of compression ratio, intake system, displacement, and speed. Increase in severity of mechanical and thermal loading is due both to increase of rotative speed and brake mean effective pressure. Cooling done by the oil has been increased, and pistons are hotter than formerly. Crankcase temperatures have risen.

Compounded oils do not eliminate the basic rate of change of deterioration of oil with temperature. Compounding delays the start of deterioration and lowers the absolute rate.

Too many engineering problems recently have been left to the chemist to solve. Several cases of lubrication difficulty are discussed, and a method of analysis is described which shows accurately what any oiling system will do, and can locate most of the troubles. The general fault nowadays is too low an oil flow over the bearings for cooling and unnecessarily high crankcase temperatures.

Compounded oils are bound to increase in the future, but ultimately each must cover a much wider range of usefulness than any present single oil covers. A short discussion is given on means for increasing brake horsepower without increasing the octane demand of the engine.

CONSIDERATION of the lubrication and fuel difficulties with automotive and aeronautic engines indicates plainly that we have reached a badly muddled situation which is hard, to some extent, on the engine manufacturer (who does not appear to realize it), and very much harder on the lubricant and fuel manufacturer. The latter has begun to discover the jam in which he is caught. Economically, the present situation is thoroughly unsound.

For some 60 or 65 years of petroleum lubricants, we were able to lubricate all machinery with reasonable success using nothing but straight mineral oils and fuels, and the various devices were designed to run on these materials. But in the last five years, a very considerable and undesirable change has occurred; mineral oils increasingly showed distress in modern designs, and it apparently was assumed that the only recourse was to "improve" the oil by the addition of various dopes, inhibitors, vegetable and animal oils, and what not. "Oiliness," that ghostly quality, has come in for a considerable play; "film strength," another manifestation from the occult, is in everybody's mouth.

The result is that a considerable number of oils have been developed which show improvement in some desirable characteristic, but by no means in all. Further, these oils generally work in a particular engine, class of design, or service much better than did the earlier oils; but in no case of which we are aware do they fit all cases. For example, an oil with additives to suit one design of diesel engine may not be satisfactory in another design, nor serve for heavy-duty gasoline engines (truck and bus service). At the same time many diesels and many heavy-duty gasoline engines are getting along perfectly well on high-grade straight mineral oils.

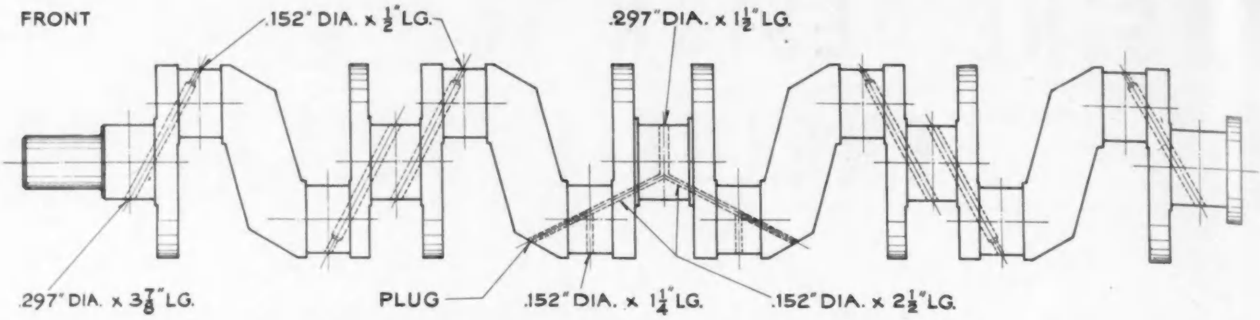
■ Too Many "Prescription" Oils

It looks as if the program is getting to be "prescription" oils for too many cases. Look at the defense situation a moment. The Army and Navy will want not over four oils for *all* automotive purposes, and they would be glad to use less. How can they possibly handle 15 or 20 prescription oils for particular designs? The answer is, of course, that it cannot be done and it won't be done.

When almost any kind of engine trouble occurs, the first thing blamed is the oil; next, the fuel. A long while after the smoke clears away, sometimes the design is reconsidered. The net result of this unthinking procedure is to throw into the lap of chemists many problems that ought to have been handled by the engineers themselves. Usually (and quite naturally) the decision is to call for a special oil if there is trouble; any fault of the design, especially the oiling system, is generally overlooked.

It must be recognized that the engineer, by his design, proposes the lubricating problem; only after the *mechanical* aspects of the problem have been covered are we justified in asking the chemist to solve the problem by a *chemical* change in the oil. Calling in the chemist to

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1941.]



■ Fig. 1 – Crankshaft oil passages of 1935 light 8 engine

solve engineering problems should be a last, not a first, resort.

Why are we in these dilemmas? Easy to answer: increase in output per cubic inch of displacement or per pound of material.

The same thing happened in the steam powerplant with the advent of the turbine. Weight of turbine per horsepower came down to one-sixth or one-eighth that for reciprocating engines, largely due to increase of rotative speed of 9 or 10:1; but life and ruggedness came down at the same time.

Without benefit of advertising, let us examine increase of horsepower for automotive engines in the last ten years. Average brake horsepower of eight makes has increased 51.0% from 1931 to 1941 (inclusive); brake mean effective pressure has come up 25%, of which 11% is due to increase of compression ratio from 5.1 to 6.65, 14% is improvement of volumetric efficiency due to better manifolding, valve timing, and lower carburetor resistance. Increased displacement accounts for 12% more, and increased revolutions per minute at rated brake horsepower for the last 14%. Table 1 shows the data from which the comparison is made, and is limited to the makes using sixes and eights in both years.

At any rate the lubricating problem involves two limits: mechanical loading, due chiefly to rotative speed and brake mean effective pressure, and thermal loading, due to the same causes, increasing the necessary heat-removal rate. Few seem to recognize that increased output has resulted in a greater proportion of cooling being transferred from cooling water to oil. Full-length jacketing already has been adopted, which yields nearly all the relief that can be expected in this direction. Since the high resistance to heat flow to the jacket water is on the gas side, increasing water circulation rate does not increase cooling in proportion. Not enough can be done in this direction. As a consequence, the piston is definitely hotter than formerly and, in spite of full-length jacketing, more heat reaches the crankcase sides via the cylinder wall, and more heat comes down the rod. The rise in crankcase temperatures in many designs of automobile motors during the period considered is a general confirmation of the foregoing conditions.

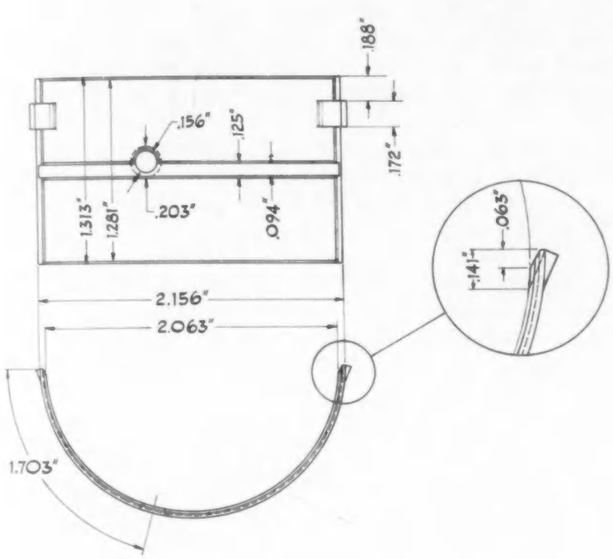
The increase in speed and brake mean effective pressure has naturally increased bearing loads, since the bearings ordinarily cannot be increased in size proportionate to output per cubic inch; but, probably much more important, the increase of temperature, particularly in the crankcase, has made the situation very hard on any oil. The chemists easily prove to us that an increase of 18 to 20 F doubles the deterioration rate of any oil. Inhibitors do not alter this condition; they can only postpone the start of

Table 1—Comparison of Significant Data for 1931 and 1941 – Limited to Models of Sixes and Eights Offered in Both Years

Car	Compression Ratio	Hp	RPM	Cu In.	Weight Sedan, lb	Hp per Cu In.	Car Weight per Hp, lb
			1931				
Buick 50	4.75	77	3200	220.7	3065	0.35	....
Chrysler 8	5.3	88	3400	260.8	3490	0.34	....
Dodge 6	5.2	68	3200	211.5	2820	0.32	....
Oldsmobile 6	5.06	65	3350	197.5	2855	0.33	....
Studebaker 6	5.2	70	3200	205.3	2900	0.34	....
Cadillac 8	5.35	95	3000	353.0	4655	0.27	....
Nash 8	5.25	87.5	3400	240.0	3360	0.37	....
Chevrolet 6	5.02	50	2600	194.0	2500 Est.	0.26	....
Average	5.1	75	3170	235.3	3205	0.32	43.0
			1941				
Buick 50	7.0	125	3800	248.0	3734	0.50	....
Chrysler 8	6.8	137	3400	323.5	3590	0.42	....
Dodge 6	6.5	91	3800	217.8	2995	0.42	....
Oldsmobile 6	6.1	100	3400	238.1	3100	0.42	....
Studebaker 6	6.5	94	3600	226.2	3150	0.42	....
Cadillac 8	7.25	150	3400	346.3	4032	0.43	....
Nash 8	6.3	115	3400	260.8	3450	0.44	....
Chevrolet 6	6.5	90	3300	216.5	2990	0.42	....
Average	6.65	113	3500	258.4	3380	0.44	30

serious deterioration, thus lengthening the time before deterioration to an undesirable degree. It is evident that, if sound results are to be obtained, very serious attention must be given to lowering oil temperatures as much toward the lower limit (about 140 to 150 F) as possible.

Let us look at the record and see what the engineer can do before he has to adopt a prescription oil. Four or five years ago, one of the popular light cars adopted cadmium-silver bearings and ran into bearing trouble in a very small percentage of their output. All the trouble was localized



■ Fig. 2 – 1935 connecting-rod bearing – Light 8 engine

in the Southwest where long, hot, high-speed runs are common. Bearings went out in from 300 to 3000 miles. Soon the idea was being circulated that cheap oils would handle this situation better than high-grade highly-refined oils. Naturally this situation put the makers of high-grade oils in an unpleasant spot, particularly as no one seemed able to remember that cheap, poorly-refined oils would not handle the piston and rings very well. It occurred to us that a check of the lubricating system might disclose a definite source of trouble.

The first move was to check the oil pump (spur-gear rotary). The manufacturer's test curve (delivery versus speed) gave a delivery at 2400 rpm that exceeded the displacement by 8% – obviously impossible. Our own tests showed the delivery curves at constant pressure, paralleling the displacement line as they should, and below the manufacturer's curve in delivery all the way. It was found further that the pump capacity could be altered nearly 25% by going from minimum to maximum manufacturing tolerances. A change of gasket from 0.008 in. to 0.004 in. increased delivery 9%. Apparently not much attention had been directed to the great importance of clearance in a capillary-sealed pump. Many engineers seem to be unaware that slip in such a pump varies directly as pressure, inversely as viscosity, and as the *cube* of clearance dimensions. However, the foregoing situation did not turn out to be controlling; the main trouble was elsewhere.

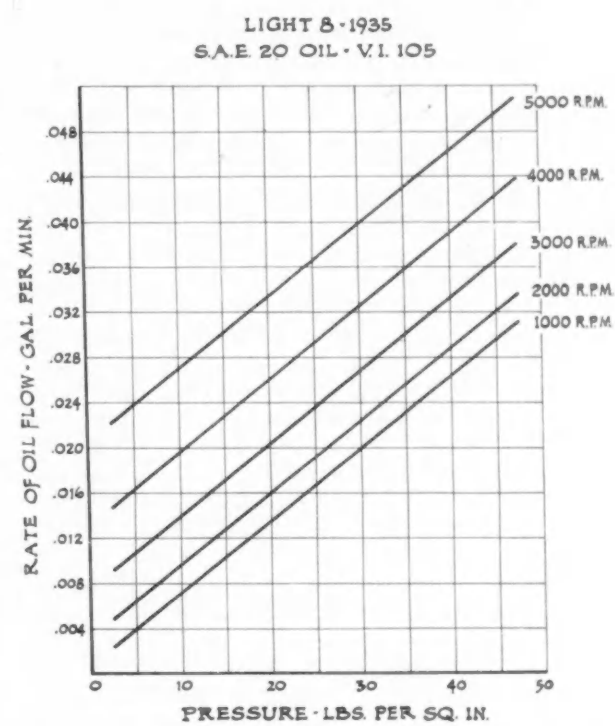
The oiling system from the pump out was then calculated at various speeds, viscosities, and pressures. Many think that such a calculation is so inaccurate as to be valueless, but this is certainly not true. The work is not even abstruse; it is merely tedious. As a proof of the accuracy, flow tests were run on the actual engine, checking the calculations within 10 or 12% in the low-speed range and, at the speeds of most interest, very much closer.

The method has been available for 40 years, but has apparently not been employed before for this purpose. Without going into too much detail, the process involves calculating the friction drop and impact losses for each member of the oil conduit system, and combining these graphically for equilibrium conditions. Most of the system is in the turbulent region of flow with the exception of the bearings which are in viscous flow. To the pump pressure is added the centrifugal pumping effect of the crank throw ducts.

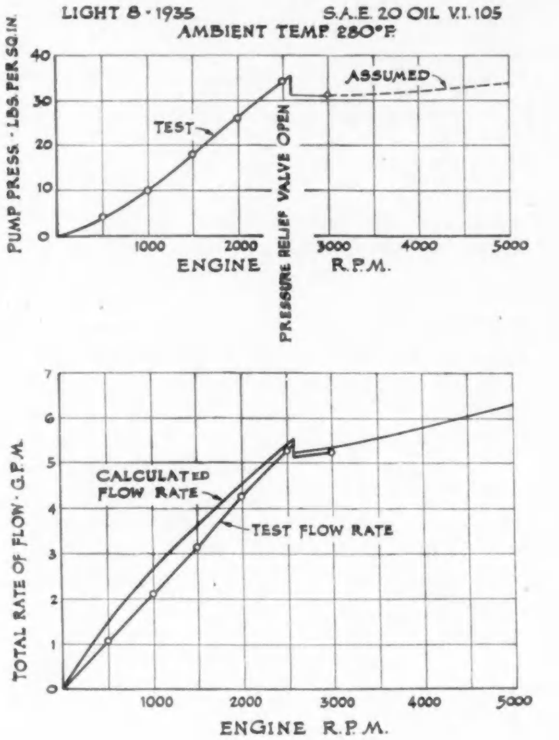
To fix the conditions for calculation, only two assumptions are necessary: (1) the eccentricity ratio of bearings; (2) the average temperature in the bearings (above crankcase temperature). A study of (1) showed that, for purposes of the analysis, a constant eccentricity ratio of 0.9 could be used without material error. The average temperature above crankcase was assumed for calculating flow and compared with capacity for heat removal; usually a second approximation gave the answer. For further work a unique solution can be obtained graphically with a little more slide-rule work.

Fig. 1 shows the schematic layout of the crankshaft oiling conduits.

Fig. 2 shows the connecting-rod bearing, having annular supply groove and two triangular grooves at the joint in



■ Fig. 3 – Rate of axial viscous oil flow at 320 F over crankpin bearing surface versus bearing inlet pump pressure



▪ Fig. 4 – Comparison between test and calculated data – Pump pressure versus engine rpm and total rate of flow through engine versus engine rpm

upper and lower halves, intended to remove dirt freely.

Fig. 3 shows the calculation of connecting-rod bearing flow for one oil with varying pump pressure, and shows the additional centrifugal pressure effect with engine speed.

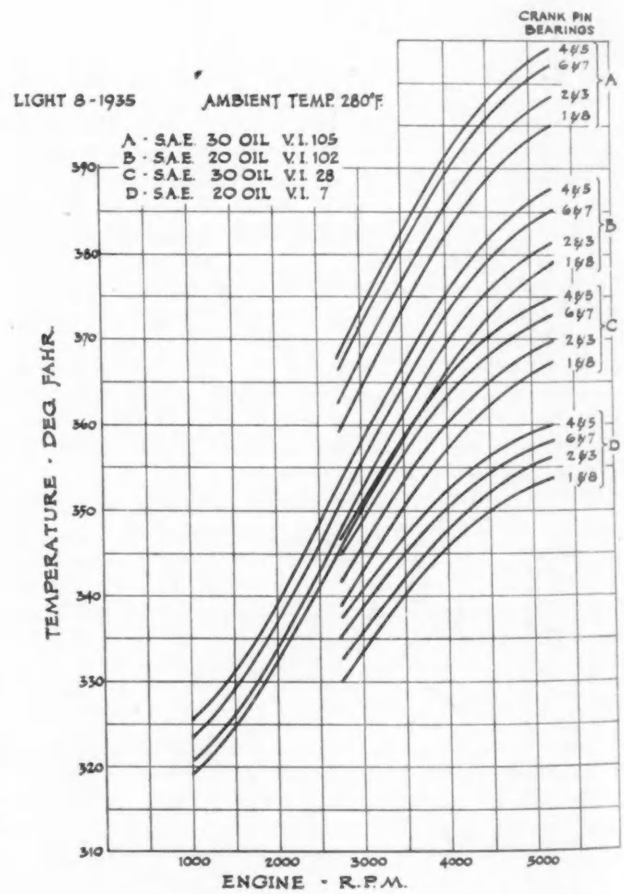
Fig. 4 shows total engine flow as calculated, against test results, establishing the accuracy of the calculation for all practical purposes.

Fig. 5 shows the maximum bearing temperatures for four oils; *A*, a high-viscosity-index oil, 30 SAE; *B*, a high-viscosity-index oil, 20 SAE; *C*, a moderate-viscosity-index oil, 30 SAE; and *D*, a low-viscosity-index oil, 20 SAE.

Table 2 locates the design trouble very clearly. Referring to Fig. 1, it is seen that connecting-rod bearings Nos. 4 and 5 are fed by a single main bearing, and the control orifices (0.152-in. diameter) are 6.5 times as long as the others. The first column in Table 2 corroborates this evident high resistance. The remaining inlet pressure after losses to the connecting-rod bearings is lowest for Nos. 4 and 5. But the striking fact is that the triangular dirt grooves have robbed the bearing surface of most of its oil supply.

The second column, Q_a, shows flow over the bearing surface; Q_r shows the amount bypassed through the dirt grooves. Only 3 to 7% of the oil supplied to the bearing is used for lubrication and cooling. The remainder is thrown out without assisting either lubrication or cooling. The difference in flow for the four oils is due to the difference in viscosity at the operating temperature. The flow for the SAE 20 low-viscosity-index oil exceeds that of the SAE 20 high-viscosity-index oil by 25% or so, and the viscosities at 320 F are in the inverse ratio. The 0.152-in. orifices to the crankpins were stated to control the distribution. The calculations show that these orifices are only a part of the total resistance, and that the bearing itself is the major control. Operating temperatures (Fig. 5) reached the very bad maximum of 402 F in the middle bearings with SAE 30 high-viscosity-index oil; and 359 F for the SAE 20 low-viscosity-index oil. The lower temperature results from the increase of flow due to lower operating viscosity. On actual test the Nos. 4 and 5 bearings failed first at 3500 rpm, thus proving the difference in flow.

In 1936, these bearing shells were changed by omitting the annular groove but retaining the dirt groove. Since the dirt groove then only registered with the oil hole twice per revolution, bypassing was cut from 93 to 97% to a small value; hence the total flow in the preceding part of the system was considerably reduced; the pump relief valve was reset from 35 to 50 lb per sq in.; so that initial pressure at the bearing jumped 80 to 120% above the 1935 values. Oil flow was consequently increased over 60%, and the bearings were out of trouble. Omission of the annular groove was probably a convenient move from the production standpoint. Our tests disclose that the annular-groove bearing has much more load-carrying capacity than the single-point feed, chiefly because the oil flow is much greater at the same supply pressure. If the dirt groove is required, it can be made harmless by changing from a



▪ Fig. 5 – Calculated maximum crankpin bearing temperature

Table 2 – Summary of the Operating Mean Oil Pressure and the Rate of Flow of Both the Viscous Flow Axially Across the Bearing Surface and the Turbulent Flow in the Annular Ring, out through the Dirt Slots
1935 Light 8 Engine

Crankpin Bearing	Oil *A*, SAE 30 V.I. = 102				Oil *B*, SAE 20 V.I. = 103				Oil *C*, SAE 30 V.I. = 28				Oil *D*, SAE 20 V.I. = 7			
	P	Q_a	Q_r	Q_t	P	Q_a	Q_r	Q_t	P	Q_a	Q_r	Q_t	P	Q_a	Q_r	Q_t
3000 RPM																
No. 1	20.0	0.025	0.595	0.620	20.2	0.021	0.589	0.610	20.4	0.027	0.584	0.611	20.5	0.032	0.572	0.604
Nos. 2 & 3	18.7	0.024	0.574	0.598	18.9	0.020	0.569	0.589	19.0	0.025	0.563	0.588	19.1	0.030	0.550	0.580
Nos. 4 & 5	16.5	0.021	0.534	0.555	16.8	0.017	0.533	0.550	16.8	0.023	0.525	0.548	16.8	0.026	0.511	0.537
Nos. 6 & 7	17.4	0.022	0.550	0.570	17.6	0.018	0.538	0.556	17.6	0.024	0.538	0.562	17.7	0.027	0.528	0.555
No. 8	20.1	0.025	0.590	0.615	20.2	0.021	0.589	0.610	20.3	0.027	0.583	0.610	20.5	0.032	0.572	0.604
4000 RPM																
No. 1	24.7	0.031	0.649	0.680	25.1	0.026	0.644	0.670	25.4	0.034	0.638	0.672	25.7	0.040	0.627	0.667
Nos. 2 & 3	23.3	0.030	0.627	0.657	23.6	0.025	0.622	0.647	23.8	0.032	0.613	0.645	24.1	0.037	0.604	0.641
Nos. 4 & 5	20.7	0.026	0.585	0.611	21.0	0.022	0.580	0.602	21.1	0.028	0.571	0.599	21.4	0.033	0.560	0.593
Nos. 6 & 7	21.7	0.028	0.601	0.629	22.0	0.023	0.616	0.639	22.1	0.030	0.587	0.617	22.4	0.035	0.576	0.611
No. 8	24.8	0.031	0.650	0.681	25.1	0.026	0.645	0.671	25.4	0.034	0.637	0.671	25.7	0.040	0.627	0.667
5000 RPM																
No. 1	31.1	0.040	0.714	0.754	31.5	0.033	0.707	0.740	31.9	0.043	0.701	0.744	32.6	0.051	0.689	0.740
Nos. 2 & 3	29.3	0.037	0.689	0.726	29.7	0.031	0.682	0.713	30.0	0.040	0.673	0.713	30.6	0.048	0.663	0.711
Nos. 4 & 5	26.2	0.034	0.640	0.674	26.7	0.028	0.637	0.665	26.9	0.036	0.626	0.662	27.4	0.042	0.611	0.653
Nos. 6 & 7	27.3	0.035	0.597	0.632	27.8	0.029	0.654	0.683	28.0	0.037	0.643	0.680	28.6	0.044	0.633	0.677
No. 8	31.1	0.040	0.714	0.754	31.5	0.033	0.707	0.740	31.9	0.043	0.699	0.742	32.6	0.051	0.690	0.741

P = pounds per square inch mean effective pressure at the bearing.
Q_a = gallons per minute flow rate of the viscous flow axially across the bearing surface.
Q_r = gallons per minute flow rate of the turbulent flow in the annular ring and out through the dirt slots.
$Q_t = Q_a + Q_r$ = gallons per minute total flow through the bearing.

triangular groove as in Fig. 2 to a land 0.03125 to 0.0625 in. wide, and 0.002 to 0.003 in. deep on one edge of each shell. This groove will only carry off 5 to 10% of the total oil supply instead of 95%, and dirt will go out through an opening twice the height of the clearance, nearly as readily as through a deeper groove.

The use of cadmium-silver bearings in this case aggravated the trouble, as cadmium silver is particularly catalytic to oil breakdown. If the oil supply across the bearing surface were increased, either by increasing pressure, using the annular groove as in the preceding, or slightly increasing clearance, babbitt would be satisfactory.

On a racing engine developing over 275 bhp at 7000 rpm, babbitt is perfectly satisfactory at loads of 3148 lb per sq in. peak and 2015 lb per sq in. But 14 gpm is circulated through the bearings; crankcase temperature is 160 to 180 F, using an external cooler; bearing temperature is not over 230 F; straight mineral oil is used.

Two years ago, a small 2-cyl aeronautic engine was in bearing trouble. Originally, a ball bearing was used on the connecting rod, replaced by a sleeve bearing in the same space. Straight mineral oil gave trouble, and doped oils, while better, were not wholly satisfactory. The crankpin oil hole was drilled on the *inside* of the throw, where the connecting-rod bearing had its smallest clearance most of the revolution, and no annular groove. Consequently, there was no appreciable outlet for oil, and the bearing was starved; result, the same as the preceding case – too low oil flow to carry off heat, too high oil temperature. The natural result is rapid oil breakdown, bearing trouble, and a dirty engine. This is not the only small aeronautic engine in bearing difficulties.

One popular, low-priced car has been in difficulties for two years with varnishing on undoped oils. As a quick indication of the trouble on break-in runs for block oil testing, water cooling on the crankcase must be employed to keep the temperature of the crankcase oil down to 280 F at 15-hp load, 3000 rpm. A similar, low-priced car engine of another make requires heaters to bring the crankcase oil temperature up to 260 F, load 55 hp, same rpm. The connecting-rod oiling in the first case is by jets in the crankcase, and a very small oil flow through the bearing is the result. The cooling, obviously, must be done chiefly by the splash of oil on the *outside* of the connecting-rod bearing end, and not nearly as effective as flowing it over the bearing in a thin, continuous film. The cylinder has been so dried up that there is trouble with scuffing on SAE 10 oil during break-in. Although the clearances have been kept nominally to the same values, we were informed that the manufacturer is working on the low side so that actually the working clearances have been reduced.

The overall effect is a general reduction of oil flow to bearings and piston. Everybody is aware that it takes very little oil to lubricate a bearing, but what generally is forgotten is the essential corollary: the heat generated must be removed.

For many years steam-turbine bearings, running at far higher surface speeds than does any internal combustion engine, employed water jacketing to remove heat from bearings. Lately by opening up bearing clearances and increasing supply pressure, enough oil is flowed across the bearing to carry off the heat, and jacketing is eliminated. Incidentally, the turbine operators use coolers to hold oil temperature down to a maximum of 150 F, and they expect an oil to stay good for six or seven years, not for 60 or 70 hr. The main reason why they succeed is – low oil temperature. In the internal-combustion engine, lubrication is really the small end of the job; the major one is cooling. Jacketing bearings is generally out of the question; only one has appeared – Harry Ricardo's design on which cooling liquid is circulated through the crankshaft. The only recourse then is to provide higher circulation of oil and, if necessary, to provide external coolers since the crankcase is an exceptionally poor oil cooler – thick layer of oil, slow flow, hot top to the crankcase.

An additional source of difficulty in the foregoing case

comes from the amount of copper exposed therein. In order to get equal distribution to each pin, it was apparently decided to use the same length from each take-off. So, the copper tubes are curled around like macaroni and expose a considerable surface of the objectionable copper. A much simpler and cheaper way to get equal distribution would be to use a single large header, straight tubes, and thin disc orifices, drilled holes of different sizes. If the header is large enough, the difference in friction drop is insignificant, and the holes can be the same size throughout. This crankcase is very difficult to clean properly, and considerable used oil is retained in draining. It is well known that, when fresh oil is added to oxidized oil, sludge and varnish may be precipitated, causing quick deposits thereafter. We have had this trouble for two or three years, especially when paraffinic-type oils are added to badly oxidized naphthenic oils. Few of the engines now produced drain properly; most of them retain from 1 to 2 qt which may be up to 25% of the full charge. One other make of engine uses the same jet lubricating system and, in our experience, the bearings in this design are short-lived.

One or two other designs that we have examined (not for passenger cars) show, in many cases, the same type of trouble. In most cases, it is connecting-rod bearings; in some cases, camshaft bearings. All indicate too low a flow of oil: plenty, no doubt, for lubrication; entirely insufficient to remove heat and keep the oil temperature and oil dwell low enough.

In a recent taxicab design, heavy varnish trouble has appeared in several fleet operations, largely confined to the hilly service where the pulls are heavy. This job was evidently designed to avoid winter sludge troubles.

In the case of these engines, the ring pressure was high and, on break-in with SAE 20 oil, moderate viscosity index, the top ring was found to be dry. An SAE 10 oil should be less stable than SAE 20 but, in this case, the substitution of the lower viscosity oil allowed enough increase of flow to lubricate the top ring, bringing down ring-belt temperature as well as flowing more oil through the bearings, cutting down deterioration, and the trouble was largely eliminated.

■ Diesel Adds Problems

The diesel engine adds some other problems, chiefly centering around the piston. Since much more of the blowby is air instead of exhaust gases, the oil is subjected to much more oxygen contact. Since more rings are used, and flame impingement occurs, piston temperatures usually run higher than in gasoline engines, in spite of little higher initial temperature, and lower terminal temperature. Moreover, contamination by fuel in the ring belt is much more pronounced, and the fuel is a relatively unstable material.

The large, slow-speed diesels have had relatively little trouble, operating during the last 30 or 40 years on straight mineral oil. Naphthenic types usually have been preferred, as they volatilized off rather than remaining to fry on the piston. The higher-speed automotive diesels have, however, been in some trouble from the start. But we are now faced with the problem of supercharging, which again raises piston temperature and, in two-cycle engines, the temperature is up in any case.

One recent design of two-cycle diesel adopted oil cooling for the piston to get temperatures down. As first tried, only a small flow was used, not enough to cool properly; it just stayed on the piston and cooked. By considerably increasing oil flow, the temperature was lowered, so that the oil temperature conditions were eased and, in addition, the oil did not remain on the hot-spots as long.

A similar condition may obtain in force-feed oiling of the wristpin for any engine. If the flow is kept low, the oil remains too long in the hot regions. If the flow is increased, not only are the temperatures lowered, but the oil is moved out of the hot region faster.

Since the general rise in temperatures, the metals involved in engine construction become of interest, from their effect on the oil. The chemists inform us that, in order of merit, the common metals run: tin, aluminum, iron, copper, lead, and cadmium. Much also depends on how they contact the oil, whether as permanent surfaces, or as small particles – even colloidal in size – resulting from wear. Tin seems to be anti-catalyst; no one heard of corroded bearings before cadmium-silver and copper-lead went into automotive engines. So far as iron is concerned, its effect in the engine changes a good deal with conditions. An experiment occurred to us that might shed light on the situation. Since we are continually breaking-in stock engines to standardize for certain varnishing tests, it was simple to take frequent samples and determine how deterioration changed as the engine broke in. Fig. 6 seems to us illuminating. Starting with fresh surfaces, such as honed cylinders, new pistons and rings, and so on, neutralization number shot up to 6.8 in the first 13.5 hr. Oil changed, and in 13.0 hr the neutralization number rose from 0.6 (admixture with undrained oil) to 3.6. Oil changed, it took 21 hr to get up to 3.65. Another change, 14.5 hr, rising slower, 1 qt of fresh oil added; still further flattening, 36 hr more time to reach 3.0; the total elapsed time was 83.5 hr. After this change, the rate dropped down enough to look like steady state.

It seems clear that the first hours were bringing down considerable colloidal iron, which was affecting the oil rapidly; as the conditioning proceeded, less and less iron came down and the fresh surfaces began to form some protective films. Blowby change is not, in this case, important, as the blowby settles down to a steady rate before the surfaces are fully conditioned.

Some conclusions may be drawn. It would seem sensible to change oil every 500 miles for the first 1000, and perhaps not over 1000 for the third change, in new engines. If the values shown in Fig. 6 can be attained with a first-class uninhibited oil, how much worse can they be with an inferior oil? The practice of many manufacturers, of putting the cheapest oil they can buy into the new engine, means that they are developing heavy deposits and attacks that may only show up as troubles after a good oil has been put into the car by the owner, and the good oil gets blamed.

Our experiments show that, the less copper in contact with oil, the better. Substitution of all-steel for copper tubing in the oil system of a test engine made a noticeable difference in the deterioration rate.

With regard to fresh metal surfaces, the period just after overhaul is important. There have been a flock of troubles in fleet operation that are almost inevitably laid to the oil. In addition to the fresh, rough surfaces of new rings and perhaps a honing job and new pistons, the engine is frequently cleaned with solvents that not only take off the deposits, but the sealing lacquer originally put on to seal core sand particles.

This stripping exposes much more clean metal than is present in a new engine, and the deterioration rate of oil is increased.

Another situation that comes up mainly after overhaul, and in most cases in the fleet shops or independent repair shops, is that there is undoubtedly some misalignment due to relaxation of casting strains, or cylinder distortion, occasionally some out-of-round in crankpins, and frequently the big end of the rod is out of shape. Most of this class of repairman fit bearings tight, as with babbitt, but copper-lead should be given considerably more clearance. When the makers of copper-lead bearings make a clearance recommendation for an industrial bearing, it is always more than for babbitt and sometimes twice as much.

There seems to be some neglect of this need in automotive practice.

Another practice which can give trouble after overhaul is that of line-reaming copper-lead bearings. As received from the manufacturers, the roughness, as measured with the profilometer, is 7 microns. After reaming, it is frequently up to 200 microns.

In addition to the mere roughness risk, this operation leaves the surface of the shell in fine shape for accelerated attack.

When copper-lead bearings are tightly fitted and idled to run-in, local high-temperature spots develop which undoubtedly start corrosion, due to high temperature and very low oil flow. It is our belief that, in many of these cases, actual lead-sweating may occur. The claim is made, especially for some proprietary mixtures, that lead cannot be sweated out of copper-lead alloys. We know that this claim is not true; there is no difficulty; it comes out at about the fusion temperature of lead in every case that we have tried. Fig. 7 proves that it can be done in a lubricated bearing. This bearing was run at 4000 rpm, 500, 1000, and 1250 lb per sq in. for 5 hr each, unidirectional load, oil inlet temperature 225 F. The load side shows perfectly clearly that the lead has sweated out and wiped; oil throw-off temperature, 356 F. Undoubtedly this action can occur in an engine, and may run some time before trouble shows. It will then look like a case of oil corrosion, as the conditions are excellent for rapid deterioration of oil, and for attack on the metal.

We have seen a large number of copper-lead bearings showing corrosion and, in many cases, the corrosion is localized very noticeably in two and occasionally three positions around the diameter. Comparison of these shells with the rods shows that the corrosion is localized opposite the weakest sections in the big end. We are all aware that corrosion is accelerated greatly by mechanical stress, particularly of reversing type. The conclusion is inescapable that, while the rod is no doubt strong enough, it is not stiff enough to provide a bearing substantially free from deflection. In racing engines, the rods are immensely stiffer at the bearings than in commercial designs, and the loading is more than twice as heavy as any commercial engine. The usual automotive big ends and wristpin ends wouldn't stay in a racing engine at all; the major reason is that they are too limber to support the bearings.

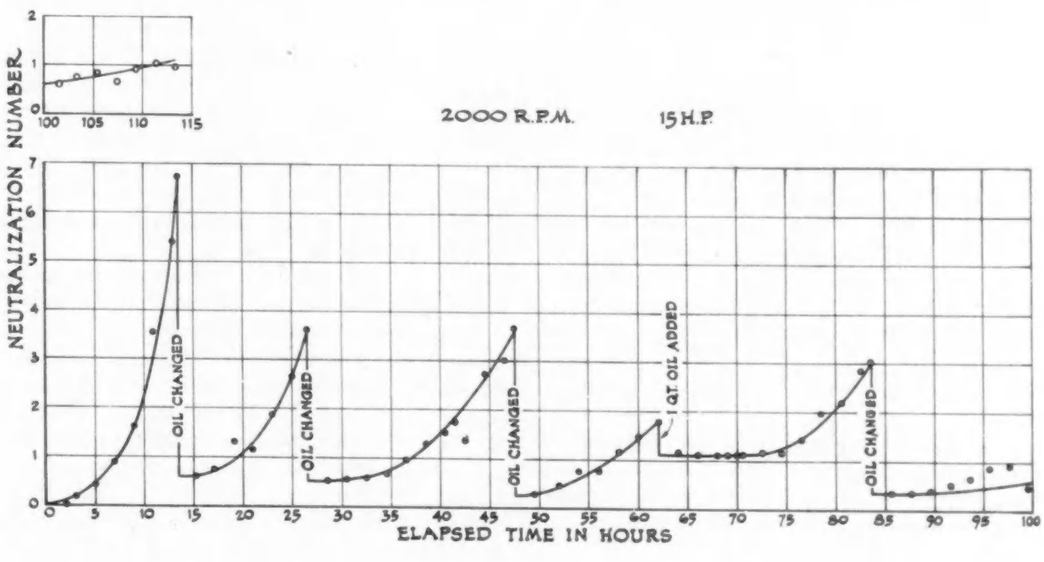
■ Deflection or Misalignment

Experimental evidence shows that a deflection or misalignment of 0.002 in. in 12 in. can cut the ultimate load-carrying capacity of a bearing 30 or 40%. There are plenty of cases in automotive practice, particularly connecting rods, that exceed this amount either in deflections or misalignments.

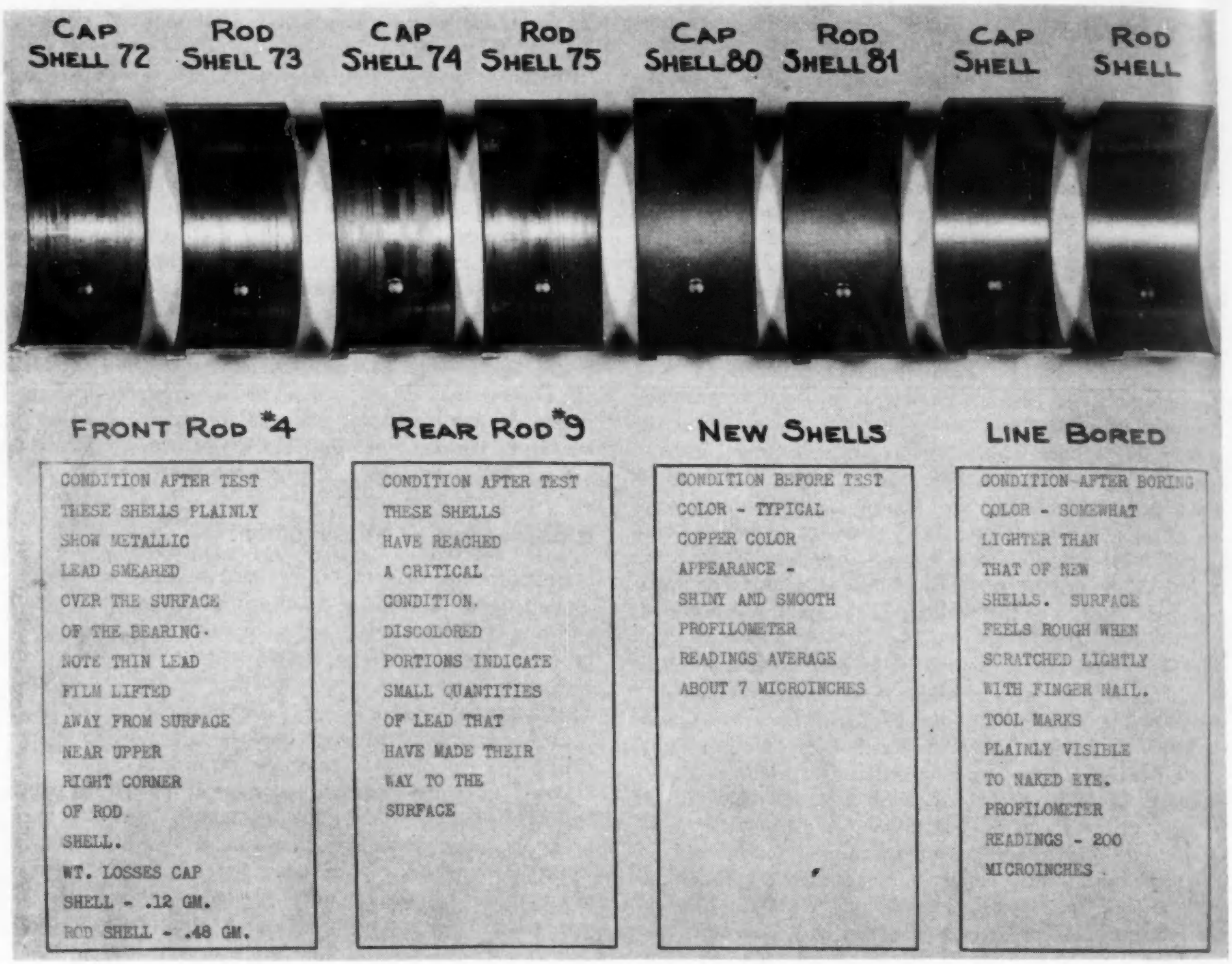
Heavy-duty bus and truck engines, as a rule, have been giving less trouble than passenger-car engines, for several reasons. They are limited to much lower top speeds, which helps considerably; the crankcase temperatures are more reasonable; the whole design is usually further removed from the danger line, and more rugged, since the buyer expects a much more extended economic life. But, even in this type of design, much could be done in detail engineering, at little or no cost, to eliminate or reduce present troubles with lubrication.

It is interesting to note the big wave for new alloys in bearings recede, and poor old discredited babbitt come back strong.

From an examination of the last five years' evidence, we



■ Fig. 6—Conditioning run—Stock test engine No. 4



■ Fig. 7 – Lead-sweating from copper-lead alloy and roughness from reaming

are convinced that very few new alloy bearings are needed in automotive engines and that, with proper cooling, oil system and bearing design, babbitt would be the most satisfactory material for 90% of the cases.

Summing up, the following conclusions appear reasonable.

(a) Since oil is organic material cracking around 650 F, oxidizing at a noticeable rate at 300 F, and inhibitors cannot change the situation, but would only slow down the rate of oxidation, it seems unwise to continue to disregard the effect of high crankcase temperature due to insufficient cooling, and still higher bearing temperature, due to too small oil flow. These temperatures can be and, for future increased horsepower per cubic inch, will have to be, kept down. Unnecessary high temperature is the major factor in present lubrication troubles, and the hurried development of an army of special oils.

(b) Oiling systems can be, and for the future must be, designed, not whittled. Drafting room or engineering department errors should not be corrected, as now, at the expense of the customer or of the oil company, after the design is in production. Most of the points covered in the foregoing examples can be caught before they leave the drafting board, and all of them should be caught in road tests or on the proving ground.

(c) The immense effect of mechanical conditions, overshadowing *all* oil improvements, must be more fully realized. Bearings must be made rigid enough to function to full advantage, and oil flows must be great enough to do the necessary cooling at reasonable temperatures for the oil.

(d) Increased cooling is now done by oil; in the future this will probably further increase, especially if piston cooling becomes general. The crankcase has barely enough cooling surface now and, if horsepower per cubic inch increases, as it will, external coolers should be used and the job can be done properly.

(e) The mania for very light oils and completely "drying-up" the engine should be reconsidered. The "drying-up" should stop before the top ring dries up and withers. The light oils make winter starting easier, but they are less stable than heavier grades, and the ultimate load-carrying capacity of the bearings is cut down until the danger zone is near operating loads.

(f) The compounded oils are here to stay, and the oil companies have provided them now for satisfactory performance, as even faulty designs, when they are on the market, must be kept running. The oil companies have done their part in improving oils to meet the emergency situations, but it must be remembered that suitable inhibitors are not picked out of the air and it frequently takes as long to settle on the best inhibitor for some particular purpose, as to develop a new model car. In one case, an

oil for one use was worked over and changed for over four years, to get the present satisfactory product. It is certainly bad judgment to keep on calling for more and more special products, when what is needed is adequate designing to avoid that call.

(g) The improvement in horsepower of engines during the last ten years has been 20% due to increase in compression ratio following improvement of octane rating in gasolines, and 80% to straight engineering design. The same large margin of mechanical over chemical improvement is available to better lubrication and bearings. Why not take this advantage? None of the moves are novel, they are all "old stuff"; perhaps that is why so little weight has been given them lately.

(h) Crankcases should be designed to drain clean; many of them retain up to 15 or 20% of the full charge. This condition is bad for a new charge. And the practice of cutting down crankcase capacity is likewise unfortunate. Rated capacity per quart of charge has gone up 25 to 30% in the last 10 years. One manufacturer recently cut crankcase capacity 16% to make room for the steering gear. That kind of economy is just hunting for trouble.

The unnecessarily rapid increase in demand for compounded oils has cost the oil companies very heavily, and in the earlier experience, many compounded oils were excellent in one or two respects, but far from satisfactory in others. It is only recently that the general tendency to increase wear by compounding has been corrected. In the diesels, particularly, the compounded oils have been beneficial, due to the greater trouble with ring-sticking. There is little doubt that we shall be putting additives in oil to a greater extent in the future, but there is also the necessity of getting compounded oils that have as wide a range of usefulness as some of the best straight oils had for years.

Now a word about gasoline. The opinion often has been expressed that the lead byproducts from tetraethyl lead might be causing some of the oil deterioration. So far, we have not been able to show experimentally that this lead has any appreciable effect.

But the talk is all of increasing compression ratio; everybody seems to feel that higher octane gasoline is responsible for the large increase in performance – and a lot of economy too. But we have indicated heretofore that only 20% of the increase was due to octane rating, the rest is design. If we go to higher compression ratio by way of higher octane, we choose a very expensive path. It costs millions to raise octane number a couple of points now, and the oil companies will be expected to swallow this without much chance of getting it back. It may therefore be well to consider a less expensive path. In the racing engine mentioned once or twice before, the overall compression ratio, including supercharge, is 12.5:1 using 82-octane pump fuel, no detonation. This same fuel would undoubtedly detonate at 8:1 in the present automobile engine. The reason for successful use at 12.5:1 is that full advantage is taken of the supercharging.

No sensible air-compressor designer would think of designing for more than 4.25:1 in a single stage if he were looking for efficiency; and when he makes a two-stage compressor, he invariably adds an intercooler. By using a really adequate intercooler on the racing car, temperature is dropped over 70 F in the intake, wet cylinder sleeves and an aluminum head are used. The cooling is therefore such that terminal temperature at the end of compression in the cylinder is over 100 F less than that corresponding to single-stage compression in the usual engine.

Superchargers have been used in racing cars with a small and inadequate cooler; none of the passenger car designers seems to have thought of using any. If supercharging is used at all, full intercooling should be used to cut down work for compression, lower terminal temperature to ward off detonation, and deliver a denser charge for high horsepower.

If high horsepower is desired, this is a good way to get it without going to synthetic chemicals for high-octane gasoline. The racing engine delivers 1.50 hp per cu in. at 6300 rpm (peak over) and 1.15 hp per cu in. at 4500 rpm, the usual top speed of passenger-car engines.

As to economy, the cycle gain has been 11% in 10 years. But this is at full throttle. At part throttle, where any car operates most of the time, the advantage of high compression ratio is largely nullified by the throttling, and the average economy of the car cannot be improved nearly as much as the cycle improvement indicates.

Since high horsepower affects lubrication, this digression into fuel will perhaps be forgiven.

None of the foregoing paper is intended for anything but constructive criticism since the possible corrections are indicated. But both the automotive designers and the oil refiners have missed some bets, and we cannot afford that in the next few years.

DISCUSSION

Takes Issue with Author's Statements on Bearing Design

– Arthur F. Underwood

Head, Mechanical Engineering Dept. No. 5, Research Laboratories Division, General Motors Corp.

THE paper by Mr. Pigott is interesting in showing that the automotive engineer is using the materials of construction to better advantage in obtaining an increase in output per pound of material. However, our tests do not agree with his mathematical calculations regarding bearing design.

The calculations are intended to show that sufficient oil leaks from the bearing grooves to cause the bearing to run hot. The temperature is supposed to be high enough to cause oxidation of the oil which forms acids to corrode the bearings. No test data are given to show that one of these engines can be made not to oxidize oil by the change suggested.

Our tests show that the rate of heat generation with a particular oil is largely controlled by the physical dimensions of the rubbing surfaces and the speed of operation. An engine which is revolved by a dynamometer (no gasoline supplied) will attain nearly the same oil sump temperature as when it is pulling a road torque. The rate of oxidation is largely dependent on the oil sump temperature. It has been shown[a] that, when an engine is revolved by a dynamometer (engine having no gasoline and no compression), the rate of oxidation is the same as when firing, and yet the horsepower per pound of material is a minus quantity. Hot-spots, such as excessively hot piston heads or ring belts, can deteriorate oil, but we would like to see experimental evidence that changes in bearing design can do so.

We have put continuous-reading thermocouples in connecting-rod bearings and have never approached the calculated values given in Fig. 5. Bearings operating near 400 F would fail because of low fatigue strength and increased plasticity in such a short time that the oil would probably never be run long enough to oxidize.

The statement is made that "it is interesting to note the big wave for new alloys in bearings recede, and poor old discredited babbitt come back strong." This "come back" has been caused by the fact

[a] See "Underwood Oxidation Test," by H. C. Mougey, presented at the SAE World Automotive Engineering Congress, New York, N. Y., May 23, 1939.

that oils are not commercially available to allow the engineer to utilize the stronger alloy bearings. It is the writer's opinion that, if improved oils were available, alloy bearings would be found in many more engines today.

We have been unable to find that adding an annular groove increases the load-carrying capacity of a bearing. If the load-carrying capacity is great enough with an annular in the bearing, then the fatigue life of an ungrooved bearing can sometimes be improved by the addition of the groove.

Also, with regard to bearing design, we quite agree that slight deflections can be shown to decrease greatly the load-carrying capacity of a bearing. However, in the special case of the connecting rod, which is the only one mentioned in the paper, misalignments due to deflections are at a minimum. The long flexible connecting-rod column allows the big end of the rod to follow the crankshaft deflections to a very reasonable degree. Fatigue of babbitt in connecting-rod bearings is only rarely concentrated at the edges. Edge fatigue is a definite indication of misalignment or deflection between the bearing and journal. Main bearings, with their more inflexible supports, are frequently found to be fatigued at the bearing edges.

In Defense of One Engine Design

– R. R. Hutchison
Pontiac Motor Division, General Motors Corp.

THE primary theme of this paper, which urges designers to avoid making engines which demand special or "prescription" oils, is highly commendable.

Since the Pontiac 1935 engine is referred to at some length as a flagrant example of poor lubrication design, it should be made clear that the author paints some unrelated items very black for the purpose of emphasis, but that nowhere does he admit the simple combination of facts which occurred. It so happened that the solvent refined oils introduced by several oil companies in 1935 produced a concentration of acid, at the temperatures then existing (and, for that matter, still existing) in engine crankcases, so high that the cadmium-silver engine bearings which Pontiac introduced the same year were doomed from the moment the combination met. No designer can have a clear understanding of the problems involved without the foregoing facts, and the controversy should not be reviewed without that information being made available at the outset.

To illustrate the designers' point of view, take a few of the items just referred to. In the instance mentioned of the relation between clearance and slip in a gear pump, it might be pointed out that tolerances of 0.001 in. on gear length, 0.002 in. on pump body depth, and 0.002 in. on gasket thickness, result in an even greater variation than the correction mentioned. Since closer tolerances are not consistent with our quality of product, our effort is to provide a pump of sufficient capacity even with maximum clearances, which Mr. Pigott later mentions is true.

Another criticism was of the method of drilling the leads to numbers 4 and 5 pin bearings in the 8-cyl crankshaft. Quoting from the paper: "Table 2 locates the design trouble very clearly." This sounds as if all bearing corrosion in all Pontiac engines was due to this condition. The design was not perfect in 1935 and it cannot be claimed that it is in 1941, as that item remains unchanged. It has the merit only of being simple and cheap to make; it does not entirely equalize the temperatures and pressures, the designers' efforts again being to make the supply adequate to the least favored bearing. In support of this statement, it can be emphasized that inspection of hundreds of bearings never revealed any trend toward more bearings corroding or fatiguing in Nos. 4 and 5 rods in the 8-cyl engine than in the other six locations. Further, it seems significant that, while the 6-cyl engine did not have this type of drilling at all, when solvent-refined oil was used with cadmium-silver or copper-lead bearings, the bearings corroded with discouraging regularity and uniformity.

The author states, a few paragraphs later, that in 1936 the Pontiac design was altered by the omission of the annular grooves in the rod liners and by an increase in oil pressure, and states triumphantly that then "the bearings were out of trouble." This statement is entirely misleading and is the conclusion most strenuously objected to. The bearings were out of trouble in 1936 because they were made of *tin babbitt* and not because of detail design changes as was repeatedly proved in engine test work and can be readily proved today.

The discontinuance of the annular groove in the rod bearings was primarily a step toward improving oil economy and one which automatically increased pressure. The paper, for some reason, does not mention the major change made by Pontiac to reduce oil temperature, which was carrying the water jackets the full length of the cylinders. Further changes would gladly have been made to retain the advantage of the increased fatigue capacity of the new bearings, but it was quickly realized that cadmium would not survive in the acid concentrations existing with solvent-refined oils at temperatures far below anything that could be hoped for and Pontiac, therefore, returned to the 1934 bearing material. This fact is not mentioned in Mr. Pigott's paper.

Since the previously mentioned service difficulties, it has been the effort at Pontiac to produce engines which will operate with reasonable satisfaction on the entire range of commercial engine oils of the recommended viscosity and we agree with the author of the paper that this is a characteristic which should be maintained.

Acid Formation in Highly Refined Oils

– C. F. Smart
Pontiac Motor Division, General Motors Corp.

MR. PIGOTT'S paper appears to be a denouncement of automotive design and a citation of the trials and tribulations of the lubrication scientists.

It is true that engine design has progressed in the direction of increased output per cubic inch of displacement and per pound of material as the author states, but it is difficult to believe that he would have us revert to 1929 design and performance in the interest of simplified lubrication requirements.

Mr. Pigott's statement that for 60 or 65 years we were able to lubricate with straight mineral oils apparently overlooks the well-known fact that back in the ox-cart days our forefathers added sulfur to wagon-wheel grease for better performance!

An important point that Mr. Pigott, in common with some other petroleum scientists, seems to overlook in discussing the effects of oil oxidation products on cadmium-alloy bearings is this – the cadmium bearings did not *cause* the highly refined oils to form acids, but merely *recorded* the fact that such acid formation was taking place, and brought this oil condition out in the open where it was painful to behold and cried out for something to be done about it. The undisputed fact that many highly refined oils were much more susceptible to acid formation than some of the oils of less highly refined character has been beclouded by many a verbal smoke screen. It is the usual thing to state that such acid formation only took place under hot climatic conditions and excessively hard driving, when the fact is that the first evidence of excessive acid formation in highly refined oil was produced in South Dakota in January of 1935 with atmospheric temperatures at around 20 F below zero.

The oil industry is to be commended for the unceasing research work exploring for acid-formation inhibitors, and undoubtedly excellent results have been obtained as is attested by the number of "heavy-duty" oils which have been put out for fleet operation. The effective use of such inhibitors in highly refined engine oils of regular service-station quality is, however, subject to some doubt.

Contrary to the author's statement, bearing alloys with improved physical properties, greater load-carrying ability and fatigue resistance are being used in increasing amounts in automotive, truck, diesel and aircraft engines. The writer has pointed out[b] that a simple treatment of cadmium-alloy bearings or copper-lead bearings with a very small amount of indium makes such bearings immune to oil acid attack. Bearings so treated are recently finding applications in truck, diesel and aircraft engines in production quantities.

Author Gives Further Evidence To Back Points

– R. J. S. Pigott
Gulf Research & Development Co.

REFERRING to Mr. Underwood's comments, this paper was a general exposition of cases and cures, and was not intended to furnish individual test results. This can be done in a separate paper, if needed. We do not understand Mr. Underwood's assumption that only calculations were made to show the temperatures attained. Fig. 4 shows the flow-test results compared to the calculations and proves the substantial accuracy of the calculation, as discussed in the paper in some detail. With regard to temperature, these tests could not be conveniently duplicated in the engine, but were run in a separate machine using two similar bearings lubricated the same way as in the engine. There is not the slightest difficulty in producing rise

[b] See Technical Publication No. 900, Institute of Metals Division, American Institute of Mining and Metallurgical Engineers: "Indium-Treated Bearing Metals," by C. F. Smart.

in oil temperature in the bearing of over 100 F, and this can be done with a number of other commercial types besides the one under discussion. We use continuous-reading thermocouples for this work.

The calculations are intended to show, not that the bearings ran hot (we knew that), but *why* they ran hot. The proof is in two fields: First, we predicted by this calculation that the middle pair of bearings should fail first; by laboratory test in the engine, these two did fail first. Second, the manufacturers changed the grooving to eliminate the serious oil bypass, thus increasing oil flow and lowering temperature, and the bearings stopped failing. During the period when the original bearing grooving was used, failures occurred in as little as 6 to 10 hr operation; or might run to as much as 25 hr. If, as Mr. Underwood appears to imply, corrosion could not take place in this time, and the temperatures did not reach 400 F, what caused the bearings to fail? Not one of the failed bearings examined by the writer showed fatigue, but all showed corrosion.

Mr. Underwood says: "No test data are given to show that one of these engines can be made not to oxidize oil by the change suggested." When one observes that, in the bearings under consideration, the flow over the six connecting-rod bearing surfaces totaled 0.15 to 0.20 gpm, and the total flow to all bearings was around 5 or 6 gpm, the effect in the whole body of oil could hardly be detected.

Other warm spots, such as the main bearings and the pistons, contribute to the oxidation but, in all cases, the time element is relatively short. Nevertheless, we all have seen that even these short dwells can accelerate oxidation of oil, not only in ring belt, but in bearings. There is no statement or implication anywhere in the paper that reduction of bearing temperature is going to eliminate oxidation, but there can be no question that reduction of hot-spots anywhere in an engine tends to reduce the oxidation rate.

There are thousands of tests, in addition to those of Mr. Underwood, supporting the hydrodynamic theory of bearings, showing that heat generation is dependent on actual viscosity of oil in a bearing, clearance, rubbing speed, and bearing area. It does not follow from that condition, however, that the crankcase temperature will rise to the same value in a motored engine as in a firing engine. This would imply that no heat came down the rods or from the cylinder walls. If a motoring test is made with cool cylinders and oil at the same viscosity as in the running engine and with the same cooling in the crankcase, the temperature will not rise nearly as high as in the operating engine.

The bearing losses, under identical conditions of oil viscosity in the bearings and speed, never account for all of the heat showing up in the crankcase. Our own tests confirm this. Of course, in a motoring test, crankcase temperature can be made anything low or high by changing the amount of air blast on the oil pan. Crankcase temperature results from equilibrium between heat input from the engine and heat removal from the oil pan.

We must also disagree with the statement that the annular groove does not raise load capacity. Our tests show that, in most cases, it does, and very materially. This result also agrees with calculation. When an annular groove is substituted for an oil hole, at the same inlet pressure and viscosity, the flow is at least doubled, and for many cases more than tripled. The consequent decrease in temperature rise increases average viscosity in the bearing sufficiently to more than make up for the reduction of area due to the groove, and the effective reduction of l/d ratio; load-carrying capacity therefore rises.

With regard to the return of babbitt, Mr. Underwood claims it is due to lack of suitable oils. If that is so, it proves pretty clearly that when forced to it, the engineer must have redesigned his bearings so that babbitt would work with available oils. Some of the best internal-combustion engines have never changed from babbitt, and have had less trouble than the newer alloys are giving.

As to misalignment effects, the writer was not referring to misalignment of the type discussed by Mr. Underwood, but to the failure of a too-limber big-end of the rod to stay in shape under load, thus concentrating bearing loads and stresses locally. The type of corrosion thus encouraged is parallel to the pin and is due to flexing.

From the reception of this paper, the writer is encouraged to hope that his desire to offer only constructive criticism was understood; the main purpose is to show that there are available little-used tools and methods for checking bearing and lubricating conditions generally, to the end that neither the bearing material nor the oil should be unnecessarily overloaded.

REFERRING to R. R. Hutchison's discussion of the paper, I believe that nowhere in the paper is the statement made that the 1935 Pontiac engine was a flagrant example of poor lubrication design. As a matter of fact, it was the one with which we were chiefly concerned, and on which the major calculations and tests were based. We also calculated one other engine, used in much larger quantities than the Pontiac, and found that, while this was also short of oil at some of the bearing points, it was not sufficiently bad to be in trouble, and cadmium-nickel bearings were used in this engine without serious difficulty. The crankcase temperature was much lower. It was not the writer's purpose to avoid acknowledgement of any of the facts involved, and it is perfectly well known to everybody that many of the highly refined oils may show the tendency to more corrosion on cadmium-silver and copper-lead bearings than some of the less highly refined oils. I think that the chemists have cleared up this matter for us at the present time, but the fact still remains that other engines on the market with cooler crankcases and with better supply to the bearings than those in trouble, had no trouble with the highly refined oils.

My purpose was not to belabor any particular engine design, but to point out how the lubrication system of many engines has received insufficient real analytical attention. The term "whittled" with regard to such designing did not originate with the writer; it was the term used by one of the engine designers in talking to him. With regard to the statement "Table 2 located the design trouble very clearly," the writer does not understand how it is possible to avoid this conclusion. The fact remains that the great bulk of the Pontiac cars in this year gave no trouble whatever with corrosion on any oil, as is stated in the paper. The fact also remains that the calculations show, as they ought to, that Nos. 4 and 5 bearings get less oil than the others and would be likely to fail first. While we did not examine hundreds of Pontiac bearings, we did examine quite a few and, in most of the cases, all bearings were sufficiently corroded so that no evidences as to which started to go first could be discovered.

The point I would like to make is that, in a racing engine with bearing loads more than 1½ times as high as Pontiac or any other commercial automobile engine and speeds two or more times the rate, babbitt bearings and straight mineral oil have been shown to perform perfectly satisfactorily, and the reason is that the bearings are relatively cool, and the bearings get enough oil through them to keep a local temperature rise down to a reasonable amount, in this case 50 F.

With regard to the statement that by the omission of the annular grooves in the rod liners and by an increase in oil pressure the bearings were out of trouble, the only information we received on the subject was that the oil pressure was raised, the annular groove was omitted and that, in addition, the bearings were changed to copper-lead. If they were later changed to tin-babbitt, we were unfortunate in not receiving this information, but do not feel that it in any way vitiates the statements made in the paper. With regard to the length of the jackets, the engines we used for 1935 testing had full-length jackets. Perhaps, for some reason, we received the '36 job but, at any rate, the writer would fully agree that the lengthening of the jackets to the full length of the cylinders would very definitely assist in this problem. In fact if the paper is read, some remarks were made on this subject. The writer would like to explain again that the purpose is not to attack engine design as such, but to attack weak points in the designs which are evidenced by analysis. The rest of the engine just referred to is undoubtedly a very satisfactory piece of designing and I neither have, nor had, any criticism to offer.

The criticisms on oiling systems which showed up most distinctly on the Pontiac can be found in quite a few other engines, but not necessarily to the same degree, and it is certainly a fact that, since 1935, Pontiac has had no appreciable bearing troubles of which we have any knowledge. It is interesting to note that all the changes made corroborate what the writer has had to say in his paper.

REFERRING to C. F. Smart's discussion, he apparently has missed the point of what the writer has tried to do. The paper is not a denouncement of automotive design and a citation of the trials and tribulations of the lubrication scientists. Such attack as it was, was directed toward something most of us are pretty well aware of and that is, the lubrication systems have not received the intelligent and analytical approach that all the other parts of the engine have had. The writer has best proved this by example and has offered a method of correction. Of course, it is difficult to believe that the writer would have the automotive design reverted to the 1929 design and performance. There is nothing in the paper which indicates any such inference.

With regard to lubrication with straight mineral oils, the writer did not overlook the fact that sulfur has been added to wagon-wheel grease for a long time, and that we still use a great deal of tallow and other animal and vegetable oils with short oxidation life up to the present time, nor did he forget that the Model T Ford used compounded oil in the transmission for years. The fact still remains that many of the new designs with high outputs were run very satisfactorily on uncompounded, straight mineral oils, and the chief reasons are that the crankcases are relatively cooler and the bearings get enough oil to keep the local temperature rise down to a reasonable value. The writer fully agrees with Mr. Smart that the cadmium-silver bearings did not cause the highly refined oils to form acid; it was the high crankcase and high bearing temperatures that raised this trouble, and that is certainly a feature of design, since at the same time on the market we have engines with equally

good output with lower crankcase and bearing temperatures that do not give trouble.

So far as climatic conditions are concerned, it would not make any difference to the engine what produces the high temperature. If it is under-cooled in the winter, as Mr. Smart points out, the same results can be obtained. Referring to his last paragraph, I think it is still true that, so far as the automotive industry is concerned, there has been a return to babbitt, and it must be stated also that the babbitt now used in bearing alloys has greater load-carrying ability and fatigue resistance. I believe that the only statement made in the paper was that it was not an objection to the copper-lead and cadmium-silver bearing (although there are objections to them) but that in many cases having been employed before, it was necessary to use them.

I will restate that we believe that the compounded oils are here to stay and will be used to a greater extent and, in all probability, more variations of alloys will be used by bearings, but there seems still to be no question that babbitt at least has a surface which can be employed in a great many cases where there has been too quick a change to other materials, and the correction of crankcase temperature and bearing temperature by proper oil flow will correct what were thought to be necessary failures in babbitt. We are now in the middle of plenty of trouble on copper-lead corrosion, not only of the type discussed in the paper, but also some cold corrosion which does not come from the lubrication oil so far as we now see. It is certainly evident enough from the conditions at the present time that the copper-lead is not entirely satisfactory. The indium treatment to which Mr. Smart refers no doubt does protect a new bearing which is installed untouched, but we have been into plenty of cases where an indium coating is completely taken off by line reaming. We find that quite a lot of this was done in repair shops, in which case not only is the applied protection destroyed, but the bearing surface is very much roughened as a rule. The indium coating, therefore, cannot be, for the present time at least, a full answer to this situation.

I would like to say again as I have said several times before, that, particularly as the writer is an engineer and not an oil scientist, this paper was not in any sense an attack on engine design but simply an attempt to prove that there is a weak spot in the lubrication systems that has not yet received enough attention. While oils may be the cause of considerable trouble in engines, it is impossible to admit that all of the troubles with engines are due to oils. There is no question that some of it is due to details of design, and there is also no question that these details can be improved. The answer to this is, of course, that the oil manufacturers and the engine designers must meet on common ground in order to solve these problems to best advantage. The oil companies have had a good deal dished out to them in the last few years, and it is no surprise, or should it be either, that some time or other they will be in a position to come back with a little in their own defense.

Quality in American Warplanes

QUALITY in military aircraft must be maintained at all costs. It is not a question of quality versus quantity, as is so often thoughtlessly believed. Rather there must be both quality and quantity. Where one adversary has quantity without quality, the other may alleviate somewhat his numerical disadvantage through the quality of his smaller force, as Britain has done in some cases. But where, as in the case of Britain and Germany, the latter has both large numbers and high quality, England would be playing into the enemy's hands to compromise with excellence of performance.

Lest there be a misunderstanding as to my meaning in using the term "quality," let me explain that I do not use it to indicate mere excellence or refinement of design and material from the point of view of artistic or esthetic value or luxury. Rather, I use the word to include those attributes and characteristics which make for maximum utility. The military airplane is a weapon and its utility is, therefore, measured by its efficiency in war. The attributes which make for combat efficiency in this airplane-weapon are its aerodynamic performance, its offensive power, and its defensive strength.

This is a short and simple list, but let us examine it a little more closely to see what are its contributory factors and to appraise their importance in affecting the efficiency of the complete airplane as a tool. First, aerodynamic performance includes, chiefly, speed, rate of climb, ceiling, maneuverability, range, and take-off and landing characteristics. These characteristics depend on airplane and propeller design, engine horsepower at varying altitudes and temperatures, and on strength of materials. Each one of these factors must be exploited to the *n*th degree if the maximum performance is to be obtained.

The next item on our list of qualities is offensive power, which is exerted by means of machine guns, cannons, bombs, torpedoes, chemicals, and their related equipment. Here too quality is vital. A weapon is, in the last analysis, an instrument for bringing force to bear upon a selected point. In the case of guns the point is usually another airplane in rapid and divergent flight, and moreover it is necessary to hit a specific vulnerable spot in that airplane. The problem of hitting with heavy impact a target traveling at a relative rate as fast as 500 to 700 mph on a changing course should appeal to the duck-hunters among my readers. A large number of projectiles of sufficient weight and accurately aimed will do the trick. But how to obtain this volume of fire and carefully direct it under the nerve-wracking conditions of air combat is a problem which has engrossed the attention of the best experts of many nations for years past. Some advocate more guns of 0.30 caliber to give the greatest rate of fire per second, as firing of this nature seldom lasts more than a few shots. This shot-gun effect should compensate for aiming difficulties. But, with the advent of self-sealing fuel tanks and protective armor, small-caliber guns have lost much of their sting, and their limited range makes close-in fighting necessary, and increases the need for maneuverability in the airplane. Bigger projectiles, up to 37-mm diameter, with irresistible hitting power, say others. But here the omnipresent bugaboo of weight rears its exasperating head. Comparatively few rounds of 20 or 37-mm ammunition can be carried if the plane is to meet the ceiling, maneuverability, speed, and climb requirements. And, if accuracy of fire is to be obtained at the longer ranges of which the larger guns are capable, a satisfactory sight must be supplied.

The temptation to make concessions in the matter of excellence of materials and design in order to speed production in these trying days is very great. But two impelling reasons, one generous and one selfish, forbid that we accept anything less than the best. The first of these reasons is the obligation we owe to those intrepid young flyers who daily in peace and hourly in war risk their limbs and lives that this nation may live and its democratic ideals be preserved for posterity. I have not dwelt upon, but you can easily surmise, the thorough training of mind and body and superior skill that are required on the part of the airmen who operate and maintain the military airplanes of today. But the least that a grateful nation can do is to supply them with the best machines which it is possible for American genius to produce.

The other reason for demanding the highest quality in our combat planes is that our very safety as individuals and as a nation depends on this high standard. Inferior instruments can produce but mediocre results, and in this fatal contest we cannot afford to come out second best.

Excerpts from the paper: "The Air Corps' Program in National Defense," by Major-Gen. George H. Brett, Chief of the Air Corps, presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 8, 1941.

A Method for Measuring DIESEL FUEL IGNITION LAG

by A. E. TRAVER and W. S. MOUNT
Socony-Vacuum Oil Co., Inc.

A STUDY of the behavior of diesel engines involves, among many other things, a measurement of the time between injection of fuel and the beginning of combustion. Measurement of this delay period, or ignition lag as it is called in this paper, may be accomplished by a variety of methods. Indicator cards are often used for the purpose and, for engine design work, they are probably most satisfactory. However, it has been found that the combustion process varies appreciably from one cycle to the next, and a rather large number of cards must be obtained before an accurate indication of ignition lag is obtained. Where fuels are to be studied for their effect on diesel-engine combustion, it is desirable to have an instrument which measures this ignition-lag period with the greatest possible accuracy, and it is best to use a meter incorporating a means for averaging the ignition lag for a large number of cycles.

Other Instruments in Use

Many instruments have been built to simplify the problem of ignition-lag measurements. The Standard Oil Co. of Calif., Standard Oil Co. of N. J., Sinclair Refining Co., Gulf Oil Co., Shell Oil Co., The Pennsylvania State College, Waukesha Motor Co., and others have offered equipment original in design and helpful in many respects. The device described in this paper incorporates many features suggested originally by these organizations.

The problem for fuel-quality measurements is a difficult one if the apparatus is to provide accuracy and ease in handling. Most engines show relatively small changes in ignition lag when fuels differing by 5 cetane numbers are compared. In fact, cycle-to-cycle variations in ignition lag may exceed the total change due to fuel quality. Such cycle-to-cycle variation may be handled nicely by averaging individual cycles over a fairly long period by electrical means. Another variable, introduced by the injection system, causes the actual point of injection to vary somewhat with each revolution of the pump shaft. Compensation is provided by starting the measuring period by means of a pickup attached to the injection valve itself.

Electrical circuits of the type used for time measurement offer no problem in getting the electric current flow started at the beginning of the time interval. Only a very small voltage impulse is required to trip the grid and initiate the flow of plate current in the vacuum tube. For this purpose either a mechanical contactor or an electric pickup is used on the injection valve. Stopping the current flow at the end of the time interval is more difficult because of the higher current values to be arrested. Early devices were equipped with a combustion pickup having heavy contact breakers of capacity sufficient to open the plate circuit without undue arcing. Breakers of this type often have high inertia and are too slow to follow the combustion process at high speeds. They have the further disadvantages that frequent adjustment is required, and secondary contacts may be introduced by bouncing of the moving parts.

A common method for stopping the current flow at the end of the time interval is to trip the grid of a secondary tube and, in effect, bypass the current through another part of the circuit. By this means, the voltage impulse required of the combustion pickup is also kept to a low value. For grid operation the injection and combustion contacts may be of either the mechanical contact or breaker type, or electrical pickups may be used.

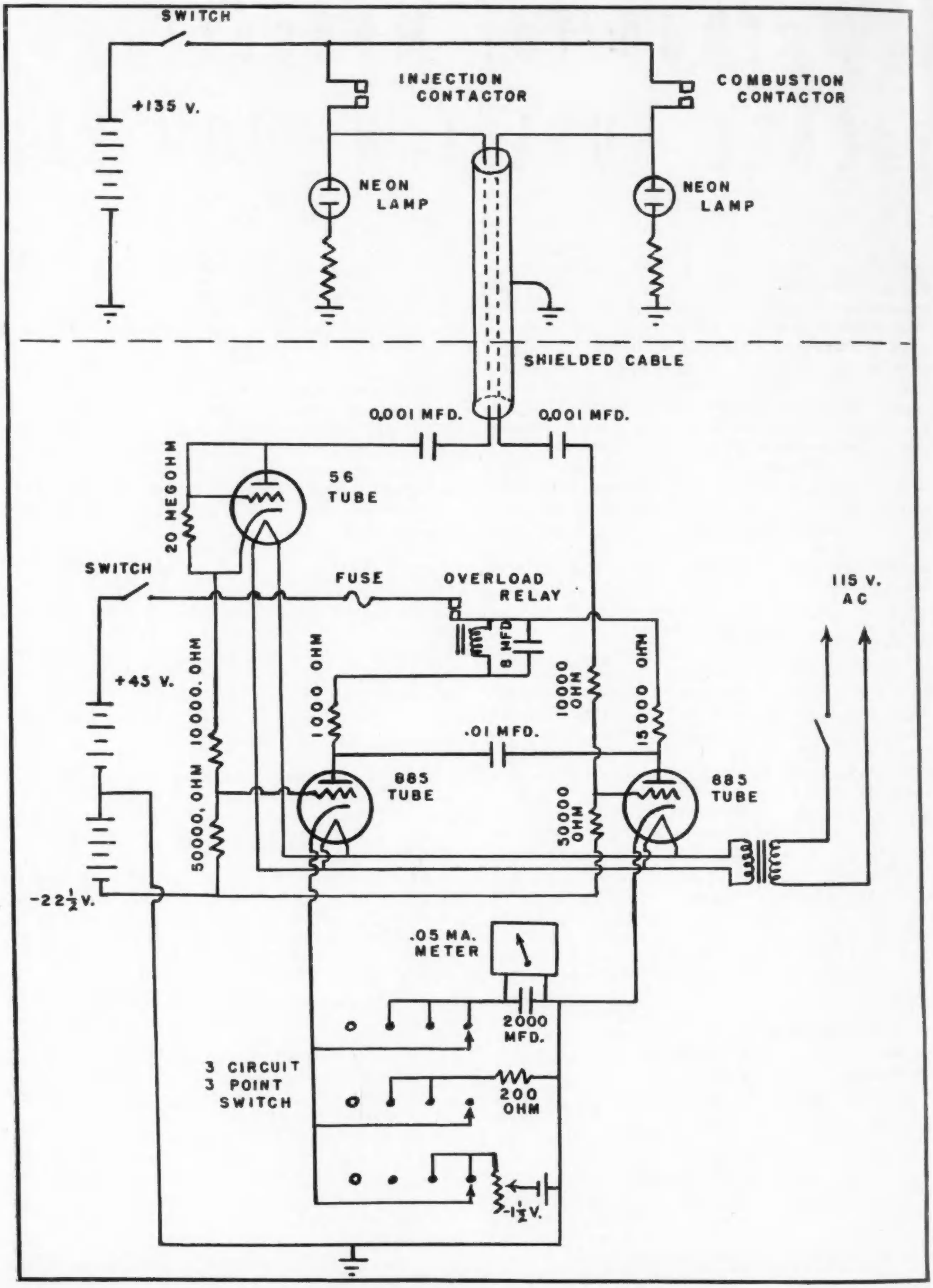
Description of Equipment

The device described in this paper is one currently used on a CFR engine for measuring cetane number of diesel fuels. It is built in such a way, however, that it can be applied to any diesel engine where the injector can be fitted with some sort of an electrical pickup or mechanical contactor, and where an outlet is provided in the combustion chamber for pressure measurements. Minor changes in the wiring diagram are required to switch from the mechanical to the electric type of contactor, but the type described in this paper will refer principally to the mechanical type.

The meter is essentially an electronic switch which is turned "on" by the voltage impulse from the first closure of the injection contact points and is turned "off" by the first closure of the combustion contact points. The current flow through the circuit during the "on" period of approximately 13 deg of crankshaft rotation is accumulated by a large condenser. The condenser is continuously discharged through a micro-ammeter throughout the whole cycle of 720 deg. Changes in the ignition lag due to fuel quality are indicated as changes in the micro-ammeter reading but most cycle-to-cycle variations are damped out.

In order to increase the meter sensitivity, the zero on the meter scale is depressed electrically. The amount of zero depression can be controlled by means of a rheostat and is set to approximately 11 deg of crankshaft rotation. Half scale deflection of the meter is obtained by a current

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.]



■ Fig. 1 – Wiring diagram – Combination coincident flash and meter instrumentation

flow for a time interval corresponding to 2 crankshaft deg. Therefore, with a zero depression of 11 deg, the meter reads at the mid-scale position with a current flow for 13 deg of crankshaft rotation. Changes in meter reading with changes in ignition lag are almost linear and the variations can be read to 1/25th of a degree on a 100-division scale.

A time delay circuit has been introduced in the injection impulse circuit so that the injector can trip its relay not oftener than once each 360 deg of crankshaft rotation. Without this delay feature the injection relay tube will often be started again shortly after top-center by the chattering of the injection contacts and will then pass current for 720 deg, causing an overload on the meter.

Another safety feature is a sensitive mechanical relay which is used as an overload circuit breaker. In case the engine runs out of fuel and the combustion relay fails to trip, the overload relay will open the plate circuits of the tubes temporarily, preventing current flow until the next cycle in which fuel is injected, and both injection and combustion contactors regain control. The circuit breaker is set to open at a point after top-center so that it does not interfere with the normal operation of the meter. A mechanical engine driven contactor may be used in place of the overload relay. When it is used it may be set to close at 10 deg before injection and open 10 deg after TDC.

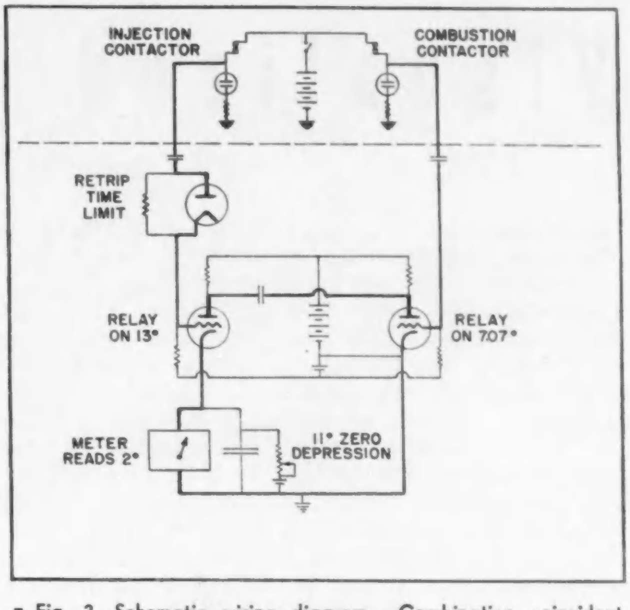
■ Current Flow Traced

A wiring diagram of the instrument is shown in Fig. 1. The same wiring diagram has been redrawn in simplified form in Fig. 2. The circuit may be best described by tracing the current flow along with the combustion process in the engine. The injection relay tube is tripped by a voltage impulse from the injection contactor, and passes current for the ignition-lag period of approximately 13 deg. The meter, which indicates the ignition-lag interval, is connected in this circuit only. The combustion relay tube is tripped by a voltage impulse from the combustion contactor. Characteristics of these circuits are such that only one tube may pass current at any given time; therefore, the current through the injection relay tube is stopped at the start of combustion.

The current flow is transferred from the injection relay tube to the combustion relay tube in the following manner: Plate circuits of the two relay tubes are connected by a small condenser. The initial current surge through the combustion relay tube is drawn from this condenser, having the effect of bypassing the injection relay tube current. Current flow through the injection relay tube is thus arrested and the grid resumes control until the next voltage impulse is received from the injector. The ignition-delay period of 13 deg has been accounted for in the description up to this point. The combustion relay tube passes current during the remaining 707 deg of the complete cycle. The end of this cycle marks the time at which the injection valve opens, tripping the grid of the injector relay tube. The combustion relay tube current is then stopped and the current flow bypassed to the injection relay circuit in the same manner as just described.

The zero depression of the meter is provided by a separate battery circuit in which a reverse current is applied to the meter, the value of which can be controlled by a rheostat.

The instrument is battery-operated except for the tube filament heaters which are heated by a 2½-v transformer from the 110-v a-c line. One 45-v "B" battery, one 22½-v



■ Fig. 2 – Schematic wiring diagram – Combination coincident flash and meter instrumentation

"C" battery, and one 1½-v dry cell are used, and all of the parts including the milli-ammeter and batteries are assembled in a small portable case.

■ Procedure for Cetane Ratings

The methods generally used for laboratory engine cetane ratings are based upon the amount of adjustment required in compression ratio to give some prescribed ignition-lag period. Such a procedure can be employed with the ignition-lag meter but it seems more accurate, and quicker, to base the ratings on the changes in ignition lag, leaving compression ratio fixed until a wide range in fuel quality must be covered. In making any given rating the comparison of sample and reference fuels is made at one compression ratio.

In general, the type of instrument described herein has the following advantages:

First – Cycle-to-cycle variations in ignition lag are averaged electrically, giving a much more accurate result than visual methods.

Second – It is unnecessary to make adjustments in compression ratio with each sample or reference fuel used. This is of particular help where the instrument is to be used on a multicylinder engine having fixed compression.

Third – Pickups of several types may be used for obtaining the start of injection and beginning of combustion.

Fourth – The procedure for adjustment of combustion pickups is somewhat simplified.

■ Summary

Where the measurement of ignition lag is principally one of engine design, there is no doubt that the engine indicator drawing a complete curve of combustion cycle is the most direct method for measurement of ignition lag. If, on the other hand, the problem is one of measuring small differences between diesel fuels, an ignition lag meter of some kind is indispensable. This problem is more apparent to petroleum refiners because of the necessity for measuring small changes in ignition quality of fuels.

Visualized AERODYNAMIC R

AERODYNAMIC development obviously would be facilitated greatly if air were a visible fluid, for we would then have the very real advantage of directly analyzing primary causes (velocity phenomena), rather than secondary pressure effects from which we theorize as to causes by indirection. On the other hand, there would be little real need of visualizing the flow if air were an ideal fluid, for mathematical flow analysis would then exactly define the actual physical phenomena. Unfortunately, such is not the case since air is both compressible and viscous, and thus practical flow problems involve certain detrimental functional faults which are non-existent in potential flow theory. It is unlikely that these unorthodox aerodynamic phenomena, some of which are very elusive and naturally complex, will be fully understood unless they can be brought from the abstract realm of mathematical hypothesis into that of concrete physical reality. Obviously, force measurements alone, by their very nature, are inherently insufficient in this respect.

Those concerned with "diligent inquiry in seeking facts or principles," to quote a dictionary definition of research, naturally want the benefit of every facility to help clarify fundamental concepts. And, since the three classic limitations of the conventional airfoil are still with us, that is, the phenomena of separation, transition, and the compressibility burble, it may be fair to pose the question whether we are putting forth sufficient research effort in the right places in the field of fundamental aerodynamics. This doubt would seem to be borne out by the fact that the major contributions to increased performance of the airplane, so far, have resulted from improvements in engines, propellers and structures, while its basic element, the airfoil, still seriously limits the all-around utility of the airplane in these important fundamental respects.

■ Opportunities for Improvement

The opportunities for aerodynamic improvement are perhaps nowhere more apparent than in the matter of high-lift development, a field in which, up to the present time, there has been only one basic flow-control contribution to the art. In this particular category, it would seem to be a self-evident fact that the stall terminates the lift increase of any airfoil and that, to attain higher actual values of lift based on the area used to provide that lift, calls for improved control over the upper surface separation phenomenon as an elemental requirement. Many attempts to increase lift flout this fundamental consideration, and often resort is had to area extensible arrangements which do not, of themselves, provide increased energy conversion. The wing slot, in one form or another, is the only functionally new means which has been evolved to effect lift improvement aerodynamically. Further indication that the fundamentals of this problem are not too well understood is evidenced by a disposition to accept the wing stall as a necessary or proper attribute of flight, or at least some have failed to admit the serious limitations thereby

AIRFLOW pictures comprise the major part of this paper. Before presenting them, the Griswold Smoke Tunnel used for making them is described.

The photographic figures show some typical flow pictures obtained in this tunnel, mostly of conventional devices, some of which give an idea of the sort of investigations that may be profitably conducted therein, visually.

The tunnel is of the non-return-flow type which design is dictated by the need for continuous supply of fresh air when smoke lines are injected into the flow. The test area, which is spanned by the model so as to give two-dimensional flow, has a high narrow rectangular cross-section. A heavy plate glass window is mounted flush with the internal surface in the front face of the tunnel to expose the model to view, together with a field of 24 streamlines.

imposed (not overlooking associated stability and control difficulties) and, at the same time, have sought higher values of lift – an obvious contradiction.

Improper concepts of essential flow requirements, whether depending upon a lack of fundamental knowledge or merely from a failure to correlate primary causes and secondary effects, are not likely to persist long even in a prejudiced mind, when high-lift devices, for instance, are developed with the assistance of exploratory visual investigations. Even those of profound aerodynamic knowledge can profit from such qualitative analysis of flow separation problems which so seriously impair the efficiency of many devices other than airfoils, such as diffusors, engine cowls, wing ducts, and so on. Separation occurring when the relative velocity energy of the flow is high, involves excessive dissipation of energy. Certainly, this one characteristic type of flow disorganization, which forms a common root of a large family tree of related aerodynamic problems, offers a fertile territory where at least initial research can be pursued most effectively by means of visual flow studies, which so helpfully facilitate comprehension of the flow process. It has been said that a problem understood is already half solved. Other characteristic flow disorders, such as the transition phenomenon, can be explored with proper qualitative test equipment, which can be so designed as to provide also for momentum loss wake surveys, pressure distribution, control surface balance and hinge moment tests and, finally, force measurement balances can be added if desired.

But first of all, a practical and convenient method of visualizing the airflow is quite essential and a brief de-

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 6, 1941.]

C RESEARCH

by ROGER W. GRISWOLD II

Aerodynamic Consultant

The powerplant consists of a centrifugal blower belt-driven from an internal-combustion engine, with which combination air speeds up to 80 mph can be attained. Exhaust heat from this engine is utilized to activate the smoke-supply apparatus. Smoke is generated by smoldering rotten wood in a special-type stove.

With this equipment a variety of model changes and modifications can be provided, the principal flow characteristics of which can be explored in a fraction of the time (and thus expense) and, in certain elemental respects, more thoroughly than is normally the case for the less flexible three-dimensional quantitative tests. This tunnel is by no means a substitute for the latter equipment, the author qualifies, but it already has proved to be a useful and convenient piece of supplemental equipment for preliminary and exploratory flow research concerned with aerodynamic problems susceptible to two-dimensional investigation.

scription of the Griswold Smoke Tunnel developed at Old Lyme, Conn., for this purpose may be of some interest. This is believed to be the largest piece of equipment of this type now in use and is capable of operating Reynolds' Numbers quite considerably in excess of those usually obtaining in such tunnels. There would seem to be little need to review what has been done elsewhere with visual flow tunnels as the story is generally familiar. However, it is perhaps significant, in view of the importance of the method for research investigations within certain limits and also because of its inherent educational value, that there has not been more activity in this field. Which suggests that there is room for improvement in the methods so far devised to overcome Nature's unfortunate oversight, from the aerodynamicists' standpoint, with respect to the optical properties of air.

■ Special Type Test Equipment

As shown by the diagrammatic sketches of Fig. 1, the tunnel is of the non-return flow type which is dictated by the need for continuous supply of fresh air when smoke lines are injected into the flow. The test area, which is spanned by the model so as to give two-dimensional flow, has a high narrow rectangular cross-section (6.5 ft x 4 in.). A large heavy plate-glass window 2½ x 4½ ft, is mounted flush with the internal surface in the front face of the tunnel to expose the model to view together with a field of twenty-four streamlines. Photoflood lamps sunk in the top and bottom tunnel walls provide an effective method of indirect lighting without glare which brilliantly illuminates the flow for direct visual observation and is satisfactory for ordinary photographic results, although the present lighting is insufficient for high-speed photography. The window is hinged for convenience in opening to change models or make minor modifications or adjustments with the model in place. The depression in the test section resulting from a downstream-energized pressure differential inducing the tunnel airflow is sufficient to hold the window securely closed during operation.

The powerplant consists of a large centrifugal blower belt-driven from a model A Ford engine, with which combination air speeds up to 80 mph can be attained, although the usual operating speed for visual flow investigation is 40 mph. All flow pictures are made at this latter velocity unless special effects are noted at other speed relationships (as with a cylinder rotating at constant speed). Below 20 mph, the streamlines become quite unsteady and considerable dispersion takes place.

The internal-combustion engine makes a most convenient source of power for this type of tunnel inasmuch as the exhaust heat (but not the exhaust gases) activates the smoke-supply apparatus. This latter consists of a closed stove (for slow combustion) containing a heater unit through which the engine exhaust gases pass prior to a common discharge with the smoke-filled blower exhaust outside the building. Smoke is generated by smoldering rotten wood in this special-type stove after which it passes through a radiator to condense the tar distillates which latter collect and are led off through a water-sealed trap. The cooled and dried smoke then enters a small high-speed centrifugal blower which feeds it under throttle-valve controlled pressure into a large streamlined tube extending vertically across the tunnel throat. From the trailing edge of this tube 24 special-type nozzles, spaced on 1⅜-in. centers, discharge the smoke midway between the tunnel side walls somewhat upstream of the minimum pressure point of the main flow stream entering the working section. It will be observed that the momentum energy of the smoke jets can be equalized with that of the main stream at their point of confluence throughout the range of operating speeds. Also, that only the central part of the stream, as viewed through the window, is rendered visible, thus leaving the tunnel-wall boundary layers out of the picture, so to speak.

Upstream of the tunnel entrance cone a series of turbulence-filtering wire-mesh and cheese-cloth screens are interposed to smooth out the flow, although these are not effective to continue the flow in the purely laminar form across the observable part of the test section. A pitot-static tube is located in the stream at a point not appreciably affected by model reactions and communicates with an inclined U-tube manometer calibrated in mph mounted on the front face of the tunnel.

Several different types of model-mount supports extending through a 4-in. diameter opening in the rear wall of the tunnel are available. The one normally used has a 1-in. shaft with a faceplate flange flush with the inner wall and is free to rotate in its double-bearing support by means of

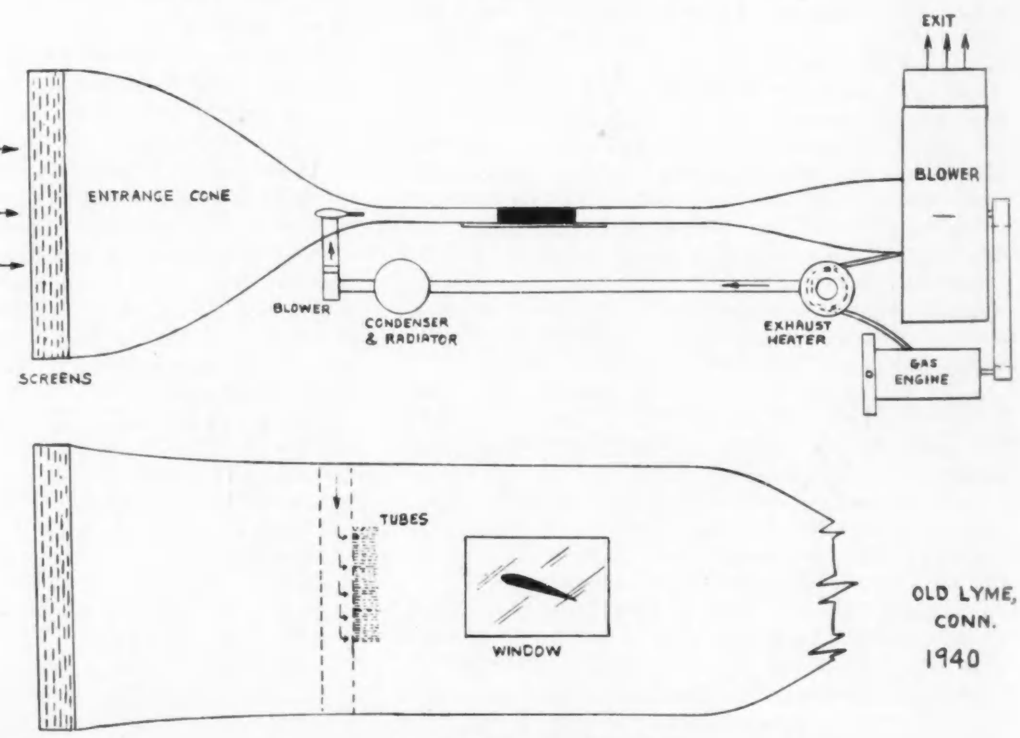
an angle-of-attack control operable from the face side of the tunnel. Provision is made to lead out auxiliary model controls, such as flap mechanism, as well as any desired pressure-measurement connecting tubes, and so on. If force measurements are desired, one of the bearings can be disengaged which leaves the model pivotally supported along the longitudinal axis of the mount by a universal joint, with consequent free-floating freedom of movement substantially in the vertical plane of the tunnel (there being a slight tolerance gap between the model and tunnel walls, in this case). The normal-to-the-flow lift forces and the down-stream drag forces are suitably led by cable and pulley from the far end of the model shaft to balance scales mounted on the face side of the tunnel above the window. Other hook-ups readily permit large-capacity powered slot blower connections from the rear side of the tunnel to the model. None of the foregoing arrangements interfere with independent angle-of-attack control.

One other highly useful piece of force measurement equipment is a 27-tube comb pitot which can be inserted in the tunnel through a small sealed opening down-stream of the working section and then suitably positioned aft the trailing edge of a model for momentum loss wake surveys in the minimum drag range. The developed pressures are communicated to a 30-tube multiple manometer which is also available for pressure-distribution tests or other pressure observations. Photographic records of manometer bank readings can be made if desired.

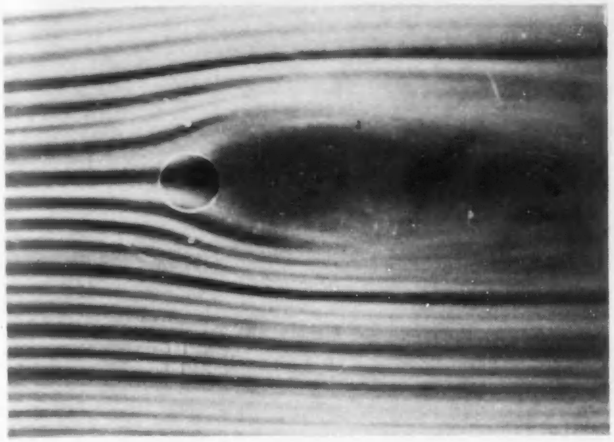
Models intended for study of high-lift flow phenomena are usually of 24-in. chord; those used only for minimum drag comb pitot tests run up to 48-in. *C*, although a possible maximum of 6-ft *C* for this latter case might be used which could thus provide a test Reynolds' Number of about four and one-half million (4.5×10^6). Making of ordinary conventional models is simplicity itself in comparison with the usual three-dimensional type of wind-tunnel model. All that is required is cutting out two solid ribs on a jig-saw and wrapping a sheet metal cover about them. Other models, involving internal flow, for instance, where such phenomena must also be visualized with internal lighting, are relatively more complicated. However, with a little care and thought in laying out the model, it is usually possible to provide for a variety of changes and modifications, the principal flow characteristics of which can be explored in a fraction of the time (and thus expense) and in certain elemental respects more thoroughly than is normally the case for the less-flexible three-dimensional quantitative tests. A tunnel of this type is by no means a substitute for the latter, but it has already proved to be a useful and convenient piece of supplemental equipment for preliminary and exploratory flow research concerned with aerodynamic problems susceptible of two-dimensional investigation.

Just as the vastly more elaborate high-speed force-measurement wind tunnels, in general, are more particularly designed for accurate scale-model test work on which guaranteed performance specifications may confidently be based and which tunnels necessarily preclude much desirable research, due to the inflexible time-expense factor inherent in such investigations, so is the relatively simple, inexpensive, visual flow tunnel primarily suited for development work in certain aerodynamic realms.

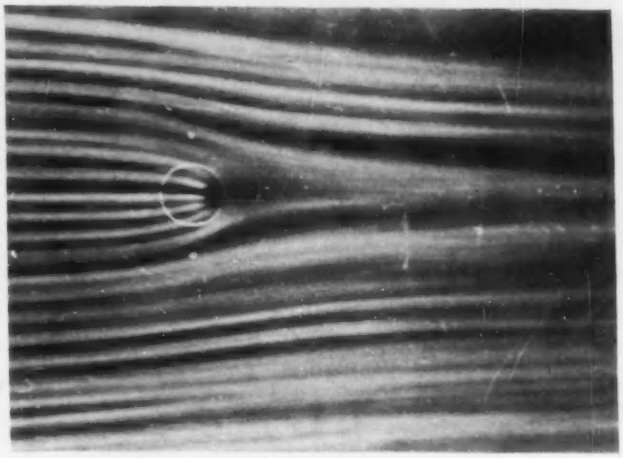
Initial qualitative flow analysis of new or improved aerodynamic devices is, in many cases, a logical first step, since it can often be the means of sorting out the aerodynamic sheep from the goats, as it were. This not only saves time and money through elimination of unnecessary later quantitative testing but, from a more positive standpoint and in the most convincing way (visually) by showing up primary causes of unsatisfactory effects, it thus helps point the way to the most favorable solutions. Within its field of usefulness, visualized aerodynamic research is of essential importance to the aerodynamicist, and perhaps if there were more of it, we might see greater progress toward releasing the conventional airfoil, and thus the so-called modern airplane, from some of its present fundamental limitations. Certainly, the post-crisis airplane will have to be largely free of these detrimental factors if it is to measure up to the herculean economic task expected of it.



■ Fig. 1 – Griswold Smoke Tunnel



© C. T. L. Fig. 2



© C. T. L. Fig. 3

■ Figs. 2 and 3 – Fig. 2 shows a source where flow is created and Fig. 3, a sink where flow is absorbed

A purely illustrative attempt to bridge the gap between a very useful mathematical concept, non-existent in the aerodynamic system, and actual physical phenomena. These flow patterns do not correspond to the classical source and sink, which is perhaps due to the fact that the smoke lines are displaced towards the observer from the energizing pressure or suction tube leading to the blower in the rear of the tunnel

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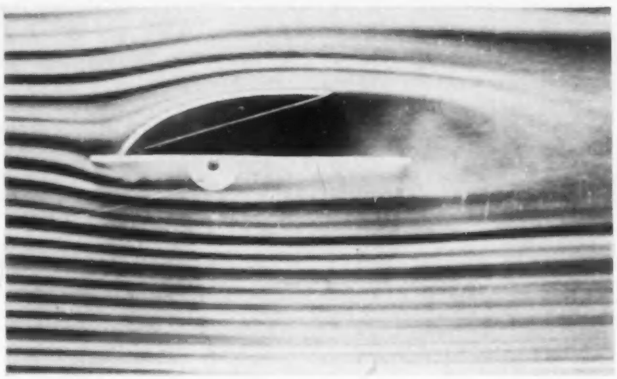

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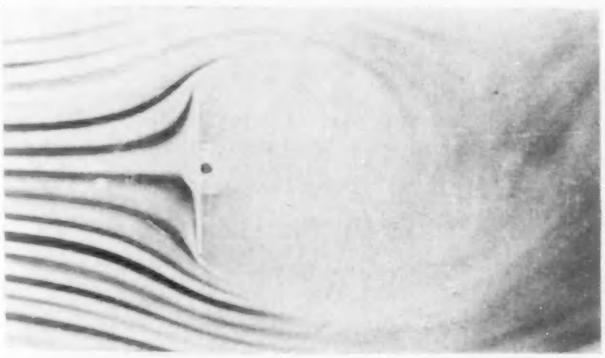
■ Fig. 4 – A demonstration of Bernoulli's theorem (that is the total energy of a fluid particle is constant at all points on its path in a steady flow or, in other words, the dynamic pressure plus the static pressure equals a constant)

The curved surface (simiiar to the upper surface of an airfoil), which is hinged at its leading edge to a fixed ground plane, "crowds" the streamlines, with consequent acceleration of the local flow. The augmented velocity increases the dynamic pressure, thus correspondingly reducing the static pressure sufficient to lift the surface as shown. Such dynamic energy conversion can be productive of relatively large negative pressures



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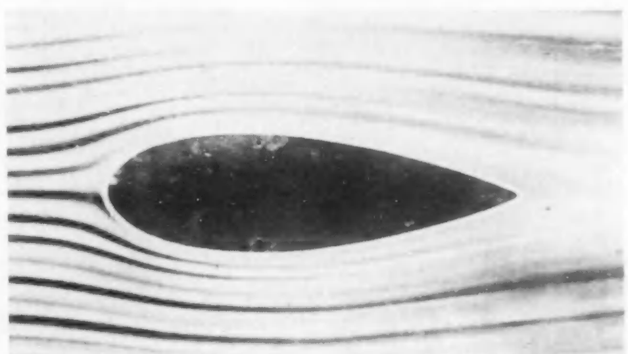
■ Fig. 5 – Flat plate normal to the flow, partially showing one unsymmetrical phase of a Karman Vortex Street forming the highly turbulent wake

Upstream of the model, the deceleration region of the stream is apparent from the widening or divergence of the flow, which is effectively brought to rest at the center of the plate, thus developing full stream dynamic or "ram" pressure at this stagnation point. The maximum attainable positive pressure increase is thus equal to the dynamic pressure of the flow ($1q$) while, as pointed out in Fig. 4, negative pressures several times this value can be realized



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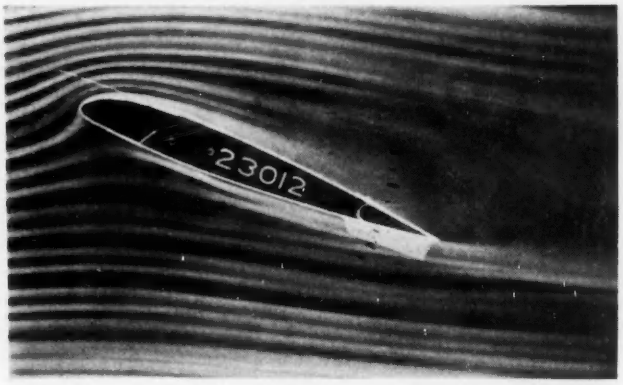
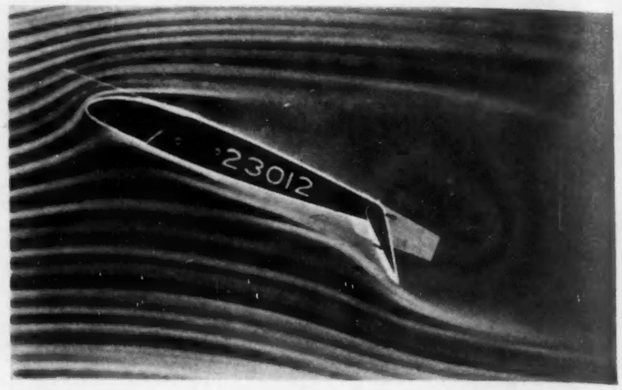
■ Fig. 6 – A typical streamline shape, offering a striking contrast in "form" drag with that of Fig. 5

The evolution of the former has effected a reduction in drag to about one-twentieth that of the latter on the basis of projected frontal areas. Though the smoke lines of Fig. 6 are not well adjusted, the thickening of the boundary layers towards the trailing edge of the model can be detected, confirmation that the turbulent wake of the so-called perfect streamline still leaves great room for improvement with respect to the ultimate least-drag laminar-flow airfoil



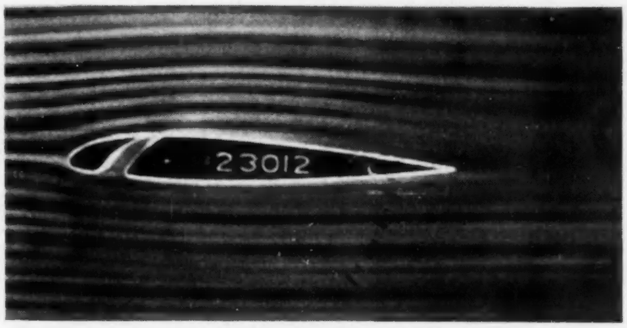
© C. T. L. Fig. 7



© C. T. L. Fig. 8

▪ Figs. 7 and 8 – Two views of the same airfoil (NACA-23012) at the same absolute angle of attack

Depression of the trailing edge flap (or aileron) (Fig. 8) induces such relative increase in the effective angle of attack as to precipitate the stall. In the case of the model with flap neutral (Fig. 7), deceleration of the flow over the upper surface towards the trailing edge diffuses the smoke so that the streamlines merge and thus lose their visual identity which makes this area of the flow appear less favorable to the photographic eye than is actually the case



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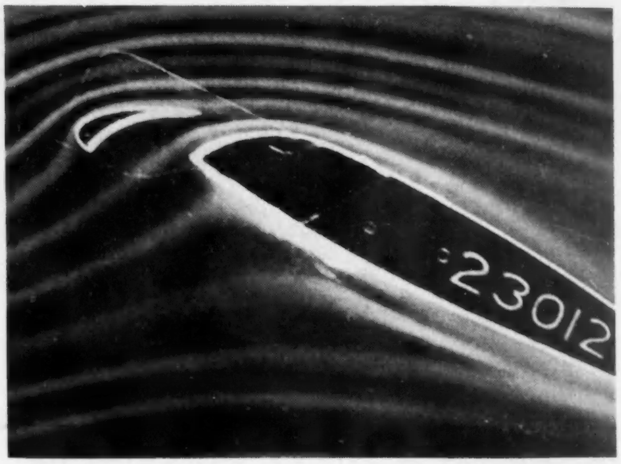
▪ Fig. 9 – The leading-edge fixed or cut slot

This design is currently coming into popular favor to delay separation of the upper surface flow at high angles of attack, to thus mitigate tip stalling and lateral control difficulties. This flow picture clearly reveals the adverse effect on high-speed drag of a permanently open slot of this type. It will be seen that the upper surface boundary layer attains excessive depth progressing downstream towards the trailing edge, an inevitable result of the relatively small pressure differential energizing the slot flow at low angles of attack, with consequent low jet velocity discharging into, and impairing the momentum energy of, the local flow over the upper surface



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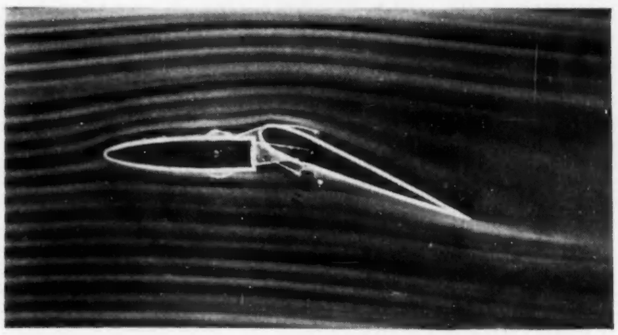
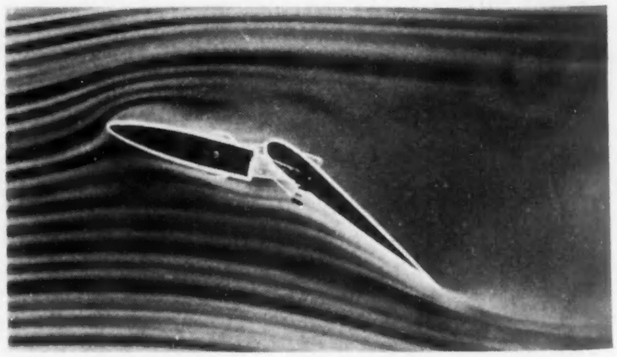
▪ Fig. 10 – Close-up view of a Handley-Page type of movable leading-edge slot at a high angle of attack

Note the high energy conversion, giving a large lifting statical pressure differential, as indicated by the diverging and converging streamlines (the former being low velocity-high pressure energy flow and the latter high velocity-low pressure flow). Observe also, the "quilting" effect on the external local stream due to the low pressure of the slot jet discharge



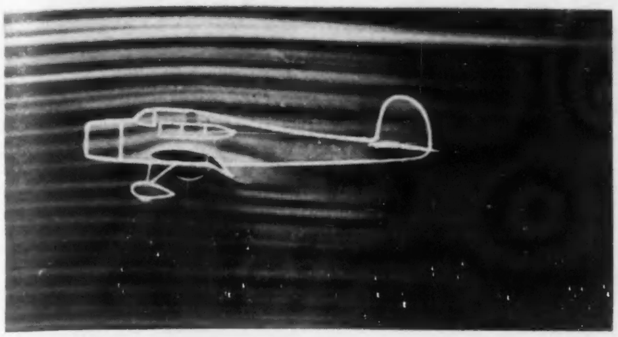
© C. T. L. Fig. 11



© C. T. L. Fig. 12

▪ Figs. 11 and 12 – The Handley-Page type of balanced control surface (such as a fin-and-rudder combination)

The illustrations show favorable flow phenomena at zero angle of yaw, but early separation at a moderately higher incidence caused by the disruptive action of the jet issuing approximately normal to the upper surface flow through the tolerance gap between the fixed and movable parts of the control surface – These figures are illustrative of the functional distinction between the wing slot and a wing gap



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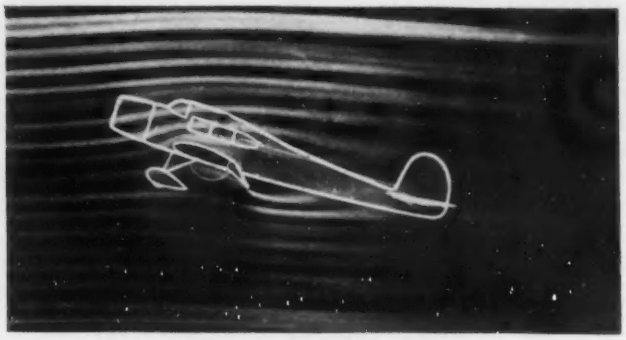

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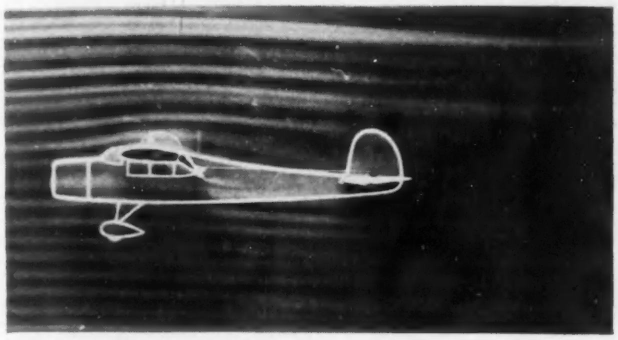

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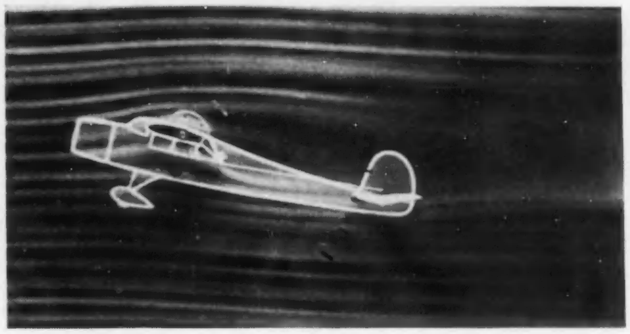
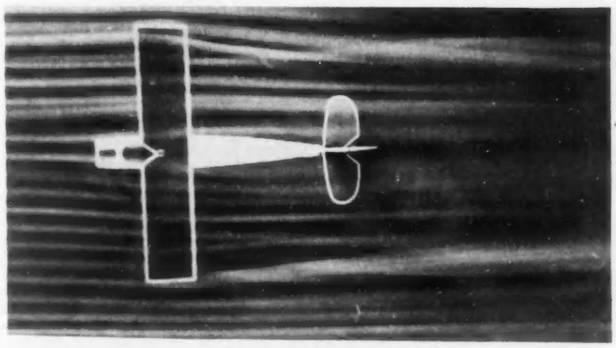


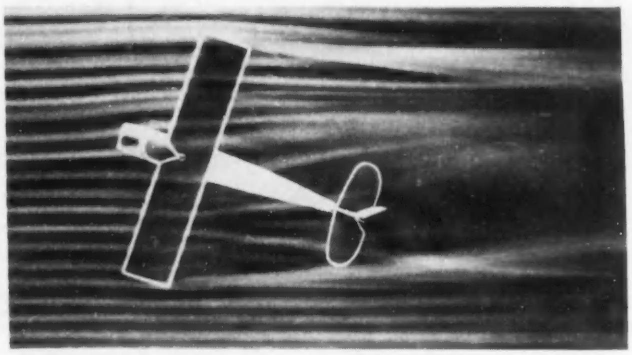
© C. T. L. Fig. 16

■ **Figs. 13-16 – Downwash effects of a flapped-wing and horizontal-tail, model combination**

The airplane is a purely decorative silhouette adjacent to the rear wall of the tunnel and on which the two model surfaces are mounted. Angle of attack is controlled by movement of the elevator from outside the tunnel. Note that the wake from the low wing position changes from below to above the horizontal tail, indicating possible stability or tail-buffeting difficulties at some intermediate angle of attack. By holding a straight edge along the smoke lines, one will see the relatively great vertical displacement of the flow (the wing is 6-in. chord) and also, the noticeable bending of the streamlines ahead of the wing



© C. T. L. Fig. 17



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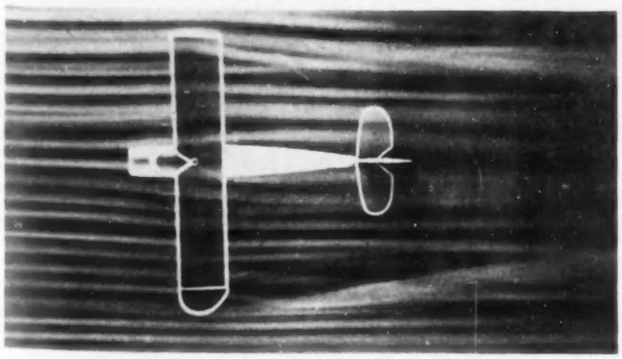
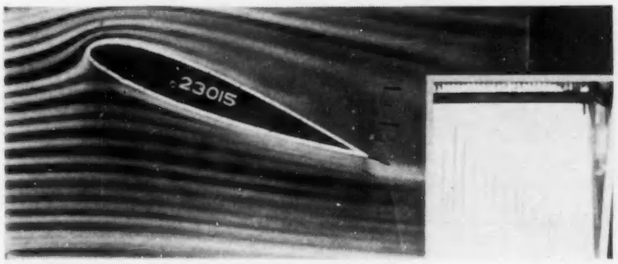


© C. T. L. Fig. 19

■ **Figs. 17-19 – Three-dimensional flow about a wing mounted vertically in mid-stream, thus intersecting the smoke lines**

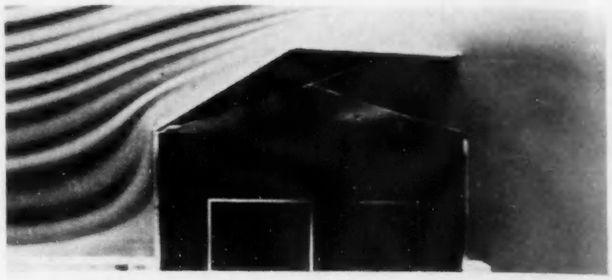
Flow lines passing under the wing showing increasing divergence away from the airplanes longitudinal axis towards the tips, while those over the upper surface similarly converge inwardly. Thus we have the clockwise-rotating vortex developing off the left-hand wing tip (facing upstream) and counter-clockwise rotating right-hand tip vortexes. The forwardly yawed wing (as controlled by the rudder of the model, Fig. 18) suggests a stronger lifting vortex field, thus offering a physical explanation of roll due to yaw. The rounded tip of Fig. 19 appears to contribute little, if any, improvement for this particular incidence, lift, and low aspect ratio (5)



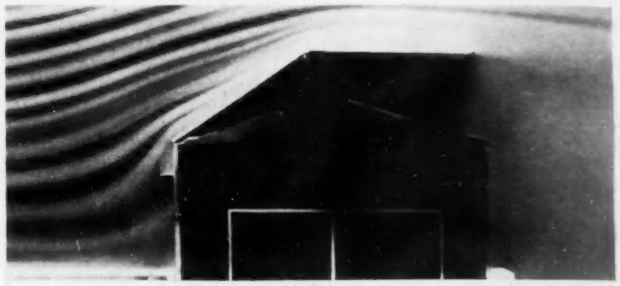
© C. T. L. Fig. 20

■ **Fig. 20 – Pressure distribution over the surfaces of the NACA-23015 airfoil prior to the stall**

The manometer bank is mounted below the window but is viewed through a mirror for simultaneous observation with the flow lines about the model. The extreme right- and left-hand tubes record the tunnel static pressure, the total-head tube having unfortunately been left out of the picture. The intermediate tubes communicate with static pressure orifices in the upper and lower surfaces of the model (alternately, except for the two obvious exceptions of adjacent pairs of upper surface tubes), the orifices being located throughout the mid-span section of the model at the chordwise stations indicated by the marks on the profile of the model. Full stream dynamic pressure is developed at the stagnation pressure point shown by the flow lines and corresponding No. 3 lower surface pressure tube near the leading edge



© C. T. L. Fig. 21



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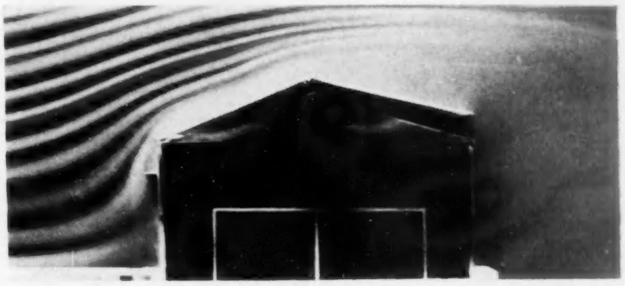


© C. T. L. Fig. 23

■ **Figs. 21-23 – "It's an ill wind," and so on – an idea suggested to C. T. Ludington by experiences in the New England hurricane of 1938 – spoilers, to prevent de-roofing of cheap factory buildings in particular**

The model roof is hinged at the upstream eave. In Fig. 21 (no spoilers) the roof lifted at 42 mph; with the wall spoiler of Fig. 22, a velocity of 63 mph was required for de-roofing; while the double spoiler combination of Fig. 23 pushed this critical velocity up to 73 mph which is equivalent to a reduction in roof-lift coefficient of over 60%. These pictures are shown through the courtesy of the Franklin Institute

Figs. 24-32 have been made available by courtesy of United Aircraft Corp.

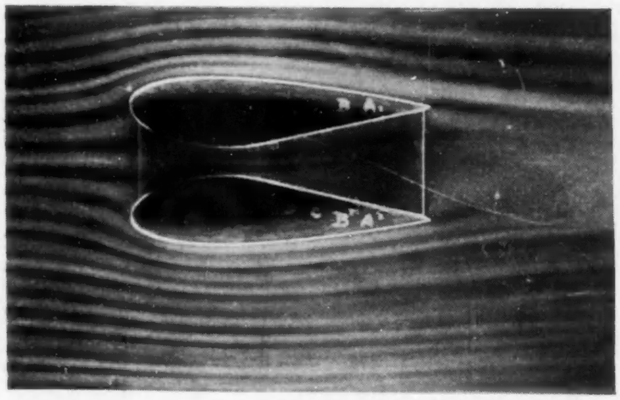


Fig. 24

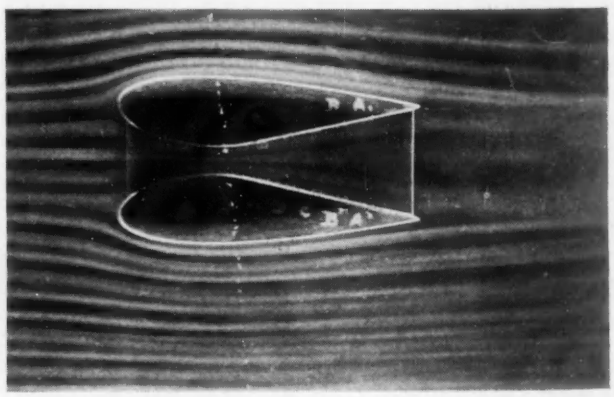


Fig. 25

■ **Figs. 24 and 25 – Flow through a venturi having a 25-deg. diffusor angle**

In Fig. 24 separation occurs over most of the lower surface of the diffusor which, in turn, has been effectively controlled in Fig. 25 by the action of suction slots a short distance downstream of the venturi throat and which are energized by a blower in the rear of the tunnel. The respective flow data at a tunnel speed of 15 mph are:

	Fig. 24	Fig. 25
Volume Flow – Q	2.34 cfs	3.95 cfs
Efficiency – n_d	80%	97%
Net Efficiency – n_t	....	65%
Relative Suction	....	8.95%

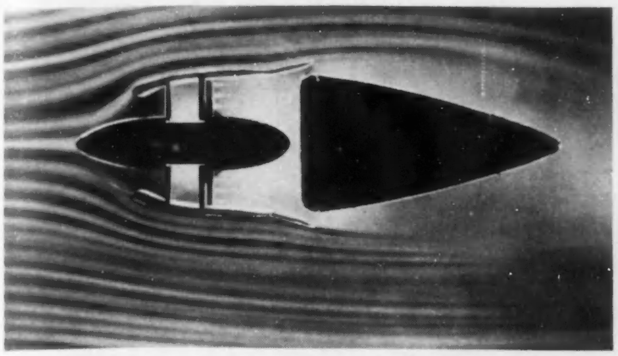


Fig. 26

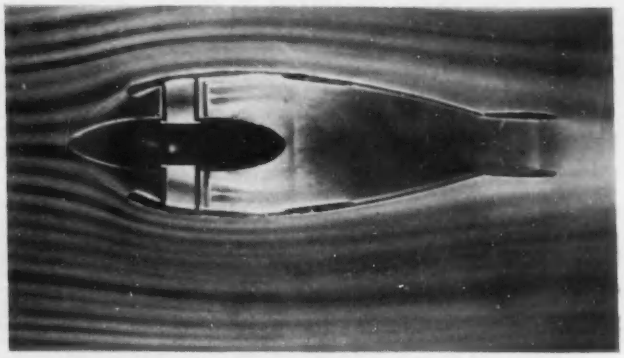


Fig. 27

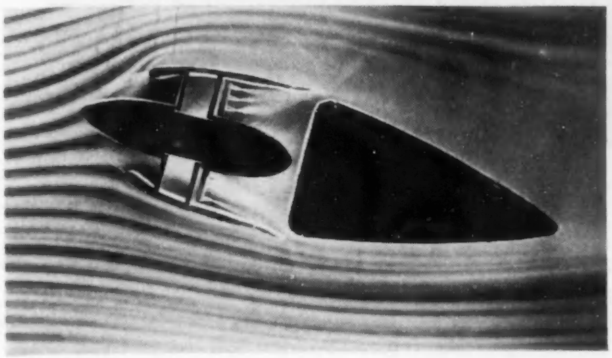


Fig. 28

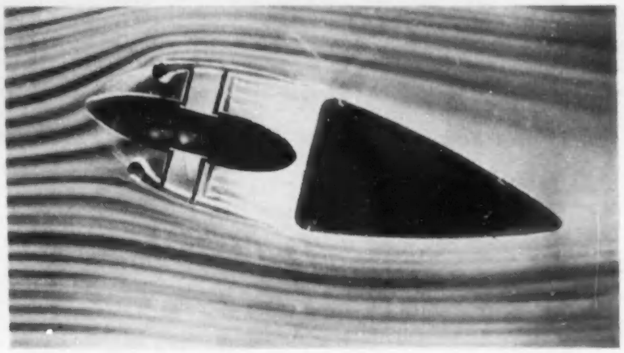


Fig. 29

■ Figs. 26-30 – Various arrangements of engine cowl combinations

The projectile-shaped object (which contacts the tunnel window) is intended to simulate the engine crankcase, while screens extending across the internal passage provide engine flow resistance factors. Auxiliary smoke jets and internal lighting to illuminate the flow within the cowl are also used. It will be seen that the deflected cowl flaps of Fig. 26 (at zero angle of attack) result in an excessively turbulent wake which could produce tail-buffeting difficulties, while the nacelle trailing edge discharge jet of Fig. 27 is considerably more favorable in this respect. The inner leading edge face of the model is bulbously faired to reduce the rectangular inlet area equivalent to that of the corresponding three-dimensional circular nacelle. The crankcase section is similarly faired

Figs. 28 and 29 have the same flap settings at 15-deg angle of attack of the nacelle in each case, but Fig. 28 shows separation occurring over the upper surface of the cowl while such is not the case in Fig. 29. Inspection will reveal that this difficulty was overcome by increasing the leading-edge radius of the cowl. A phenomenon of this kind is more readily isolated (and a satisfactory remedy evolved) by visual studies, than is the case with quantitative tests alone

Fig. 30 shows an unsymmetrical type of cowl at 10-deg angle of attack. The nature of the engine cooling-flow cowl problem is such as to suggest that other, somewhat different, unsymmetrical arrangements may be developed in the future

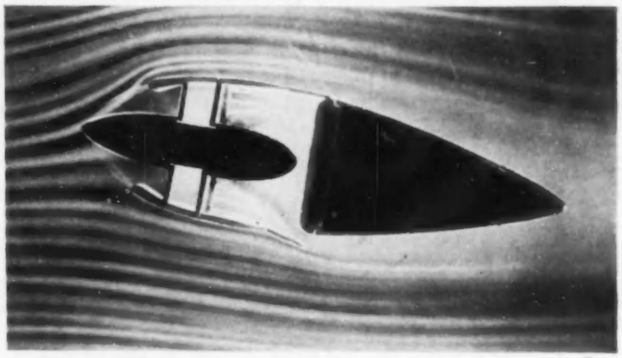
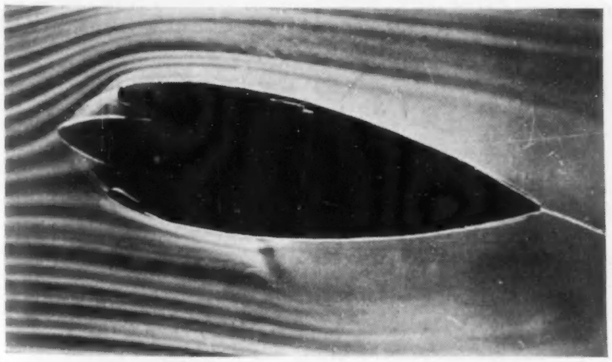
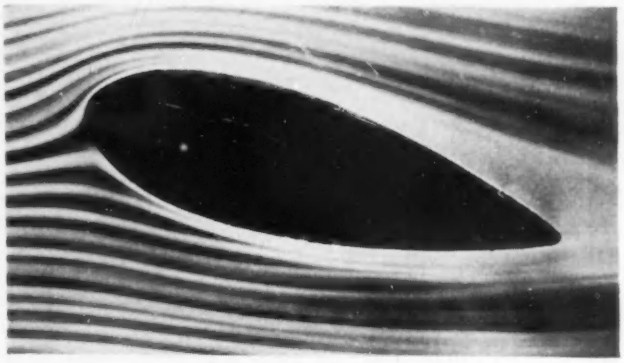


Fig. 30



■ Fig. 31 – The highly disruptive action of a pressure jet issuing normal to the flow (indicated by the shadow at about mid-chord, extending below the lower surface)

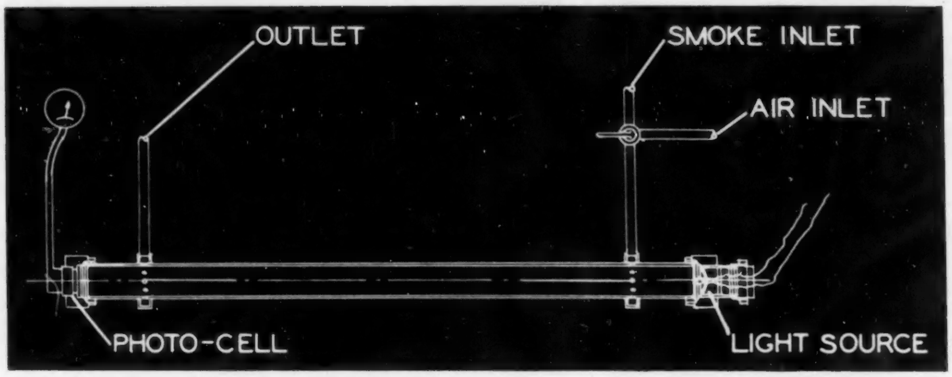
Thus, exhaust jets, not directed tangentially downstream into the flow, will precipitate a turbulent eddying backwash productive of relatively large interference drag



■ Fig. 32 – Nacelle of the previous figures at 15-deg angle of attack, faired in to the basic airfoil shape (developed by John G. Lee of United Aircraft Corp.)

Note the smooth flow lines in contrast with the disruptive effect of some of the cowl arrangements

Requirements of



■ Fig. 1 – Schematic diagram of smokemeter

THERE is an increasing realization of the importance of being able to evaluate the density of the smoke in the exhaust of a diesel engine. In commercial units in field use, smoke is objectionable. In the development of a diesel engine, smoke is an important criterion of the performance. It is an indication of the completeness of burning, which has an effect on the efficiency and power of an engine. Furthermore, the cleanliness of burning is a factor in determining the useful life of an engine and the length of periods between overhauls. Consequently we desire to indicate small changes in the smoking tendency of an engine so that we may know in which direction to go to improve this quality.

Since we wish to evaluate smoke density, the question is how are we going to determine it. The most obvious and simple method is by visual observation. We do use this method in our laboratory, and rate smoke by observation as "clean," "trace," "light," "moderate," and "heavy." The disadvantages of this method are that two different observers may rate the same smoke differently; that one observer from day to day cannot be always consistent; that surrounding conditions, such as the nature of the background against which the smoke is observed, or the amount of light at the time and place of observation, affect the appearance of the smoke.

■ Human Element Reduced

To reduce the human element in the visual observation of smoke, smoke charts have been made, consisting of cards on which patterns of varying density are drawn. By comparing the smoke with the appropriate card pattern, a rating of the smoke may be obtained. However, some human element still remains in these observations and, with an experienced observer, they offer little advantage.

Another method of rating smoke is to expose a piece of cloth or paper to direct impingement by the smoke for a definite length of time, and then to observe or measure with the aid of a photo-electric cell the amount of darkening of the cloth or paper. But this method is not as direct, or as easy to use, as the light absorption photo-electric cell method.

If smoke is passed between a source of light and a photo-electric cell, the smoke will absorb a certain amount of the light, and the resultant reading of the photo-electric cell output may be used as a measure of the smoke. Obviously, the length of the column of smoke will affect the magnitude of the reading. Such an arrangement has been mounted on exhaust stacks, with a light source located in a suitable enclosure so that the light is directed across the stack on to a photo-cell. This arrangement has several disadvantages, however. Since the distance across the stack is usually not very great, the sensitivity will be low unless means are taken to increase the length of the smoke column sighted through. There is the problem of how to standardize the readings to allow for variation of light-source intensity or for fouling of the windows. Protecting the photo-cell from the high temperatures of the exhaust gas presents another problem. Furthermore, there is the problem of keeping the windows clean.

All these difficulties have led to the development of the sampling-type, photo-electric cell smokemeter, in which a continuous sample of the exhaust gas is brought into a measuring tube for a short interval (Fig. 1). At one end of this measuring tube is a light source and, at the other end, the photo-cell. The length of the measuring tube may be any desired value to secure the sensitivity required. A two-way valve may be located at the entrance to the measuring tube so that, at one setting of the valve, clean air may be drawn into the tube. This provides a means of adjusting the light intensity to some predetermined value by observing the photo-cell output. By providing dead air spaces in front of the windows at the ends of the measuring tube, the windows may be isolated from the smoke so that they will stay clean for a greater length of time. Finally, this type of smokemeter provides a portable instrument, which may be used wherever desired.

The fundamental requirement for a smokemeter of this type is that it give a true indication of the smoke density. The difficulty arises here as to how we are to know whether a smokemeter gives a true indication of smoke density. The smokemeter itself is our only means of measuring

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.]

a SMOKEMETER

by KENNETH M. BROWN
Caterpillar Tractor Co.

THE necessity of being able to evaluate the degree of smoke in the diesel engine exhaust is becoming universally recognized. Various methods of measuring smoke are discussed briefly.

Of these methods, the light-absorption, photoelectric cell, sampling-type smokemeter appears to be the most satisfactory. The fundamental requirement for this type of smokemeter is that it gives a true indication of smoke values.

Other design requirements are listed, and a smokemeter described which appears to meet them with reasonable satisfaction. The principal departures in the design of this smokemeter from that of others of the same type are in the greater length of the measuring tube, the use of a flashlight-type bulb with reflector for a light source, and the use of a vacuum pump to draw the sample through the smokemeter.

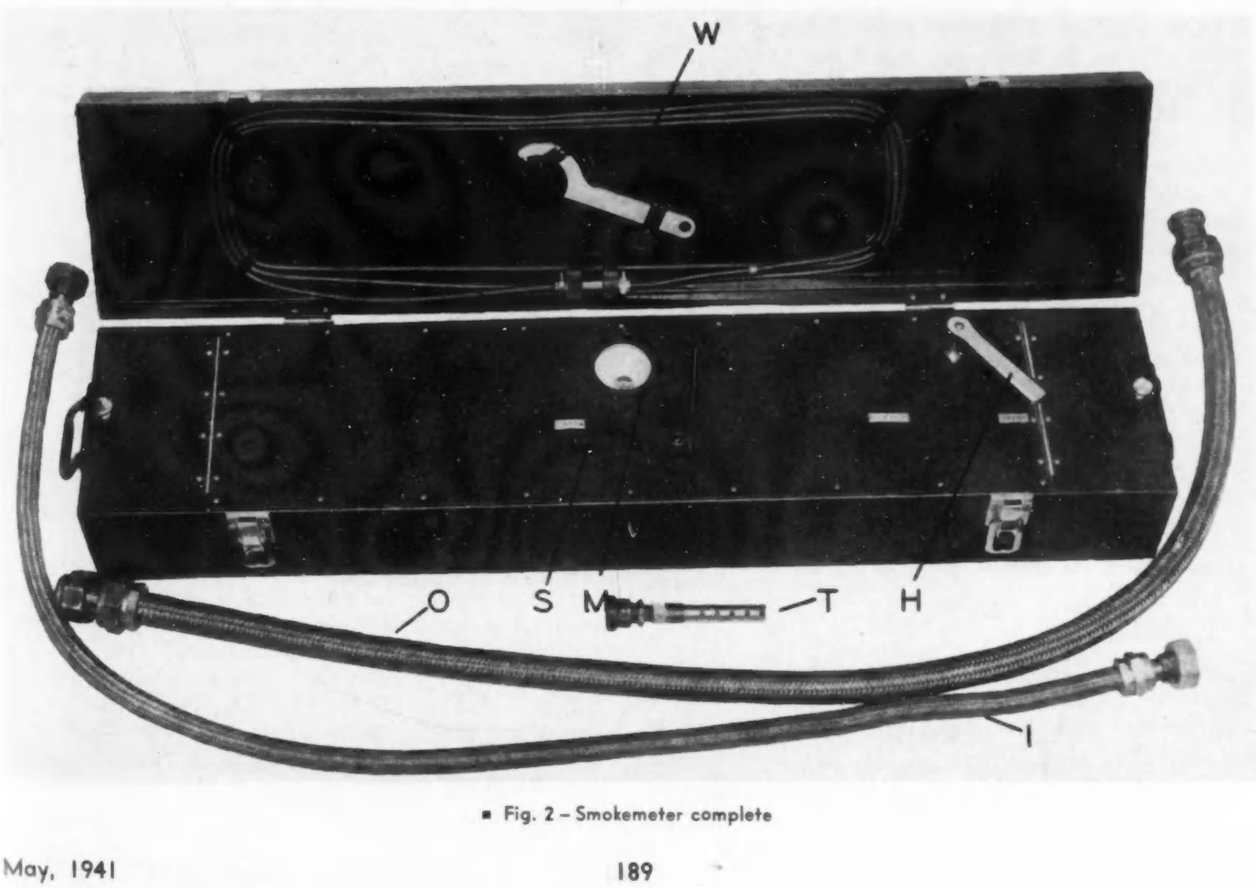
A scale for measuring smoke, "smoke density per unit length of smoke column," is proposed, which should allow all laboratories to interpret smoke readings the same.

smoke density precisely. All that we can do is to require that our smokemeter give reproducible results; that two different smokemeters give the same results; that the results be consistent with what we would expect from the engine being tested; and that they agree with visual observations.

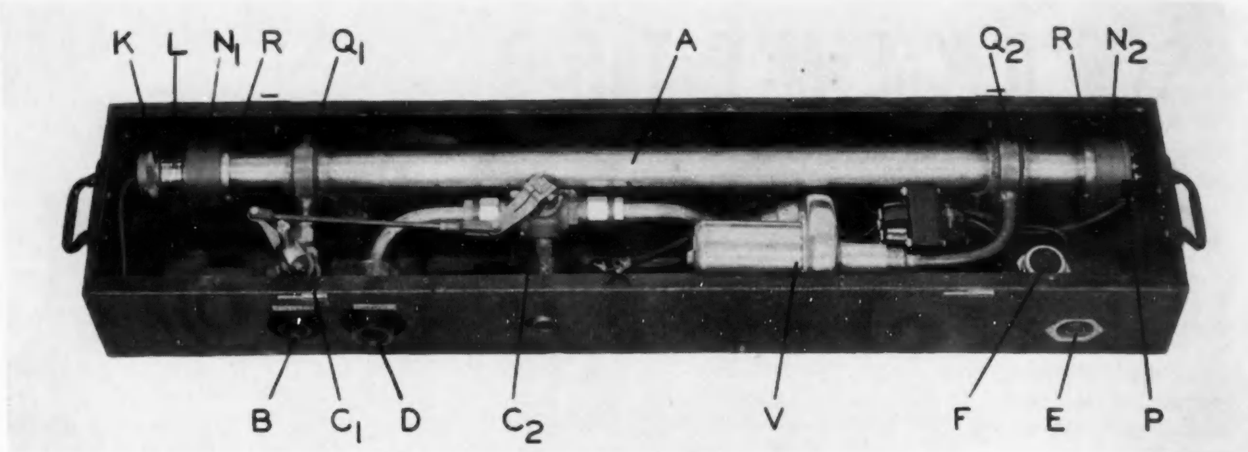
Among important design requirements for this type of smokemeter are the following:

1. It should be sufficiently sensitive to the range of smoke values that it is desired to measure.
2. The light intensity should be capable of easy and smooth adjustment.
3. The light intensity should remain constant for any setting.
4. The windows should not become dirty too quickly.
5. The windows should be easily accessible for cleaning when they do become dirty.
6. The smokemeter should be easy and simple to operate.
7. The smokemeter should be quite portable.

The smokemeter shown in Figs. 2 and 3 seems to fulfill these requirements fairly satisfactorily. The development



■ Fig. 2 – Smokemeter complete



■ **Fig. 3 – Smokemeter with panel removed – Back side**

of this smokemeter was started in our laboratories in 1937. Since then, other smokemeters of this type have appeared, but they differ in several important details.

The total length of the main measuring tube (*A*, Fig. 3) is 36 in., but the distance between inlet and outlet is 28½ in., leaving 3¾ in. of dead air space at either end. This amount of dead air space seems to be quite effective in prolonging the interval between window cleanings.

The length of the measuring tube in this smokemeter is somewhat greater than in most smokemeters of this type. This greater length increases the sensitivity of the smokemeter in the region of low smoke density and decreases the sensitivity in the region of high smoke density. This effect is shown on Fig. 4, where "% Smoke" by a 14-in. meter is plotted against "% Smoke" by a 28½-in. meter.[1] It will be seen that a sample of smoke which gives a reading of 10% on the long meter would only give a reading of 5% on the short meter. On the other hand, a sample of smoke that read 95% on the long meter would read 76% on the short meter. But this latter sample of smoke would be very dense, in fact, quite objectionably so. Consequently, we are really not so interested in this end of the smoke scale since it represents a region that we desire to stay away from in engine operation. On the other hand, the increased sensitivity in the low-smoke-density region we find quite advantageous in our laboratory, since we are striving to produce as clean an exhaust as possible, and this increased sensitivity allows us to detect smaller changes in the region in which we are most interested.

The sample of smoke is drawn from the exhaust pipe through a sampling tube of conventional design (*T*, Fig. 2). This sampling tube extends across the exhaust pipe, and has equally spaced holes pointing into the exhaust stream to obtain a fair sample.

---

[1] This curve is a theoretical relationship, but it has been checked by experiment at a number of points. The relationship is derived by assuming that unit lengths of smoke column absorb a certain fraction of the light incident upon them. This assumption leads to the equation of the curve:

$$S_L = 100\left[1 - \left(1 - \frac{S_S}{100}\right)X\right]$$

Where S_S = % smoke, by short smokemeter.
S_L = % smoke, by long smokemeter.
X = Ratio of effective length of long smokemeter to that of short smokemeter.

The problem of obtaining a good sample is one of the most difficult with this type of smokemeter. We have found that great care must be exercised in choosing a location for the sampling tube. Pressure waves and surge conditions seem to have a great effect on proper sampling. No hard-and-fast rules can be set for the proper location, but each installation must be considered on its own merits.

The sample of smoke is conducted through a 5-ft length of ½-in. flexible tubing to the smokemeter, where it connects at (*B*, Fig. 3). This tubing is of sufficient size to keep the restriction to a minimum; yet, being flexible, it is convenient to handle.

The smoke sample enters through the two-way valve C_1 (Fig. 3), which is controlled by handle *H* (Fig. 2) so that either clean air or the smoke sample may be drawn into the measuring tube.

The sample enters the measuring tube through the piezometer ring Q_1 (Fig. 3), and leaves through piezometer ring Q_2. These piezometer rings insure an even distribution of the smoke in the measuring tube, which contributes to the reproducibility of the meter.

■ Vacuum Pump Used

The sample of smoke or air is drawn through the meter by means of the vacuum pump *V* (Fig. 3). The use of the vacuum pump means that the pressure in the measuring tube is slightly below atmospheric. This condition has the advantage that, if there is any leakage at the ends of the measuring tube, it will be of clean air into the tube rather than of smoke out of the tube. This fact tends to keep the dead air spaces at the ends of the tube consisting of clean air rather than smoke, which contributes to the continued cleanliness of the windows.

It is obvious that the absolute pressure in the measuring tube will have some effect on the smoke density. However, within the range of pressures that we have tried in our smokemeters, from a few inches of water below atmospheric to a few inches above atmospheric, there was no noticeable effect on smokemeter reading.

The smoke leaves the smokemeter through the two-way valve C_2, and connection *D* (Fig. 3), where another flexible tubing may be connected to return the smoke to the main exhaust line. In our laboratory, the exhaust line operates under a slight vacuum, so this return provision was made. Ordinarily, it would not be necessary.

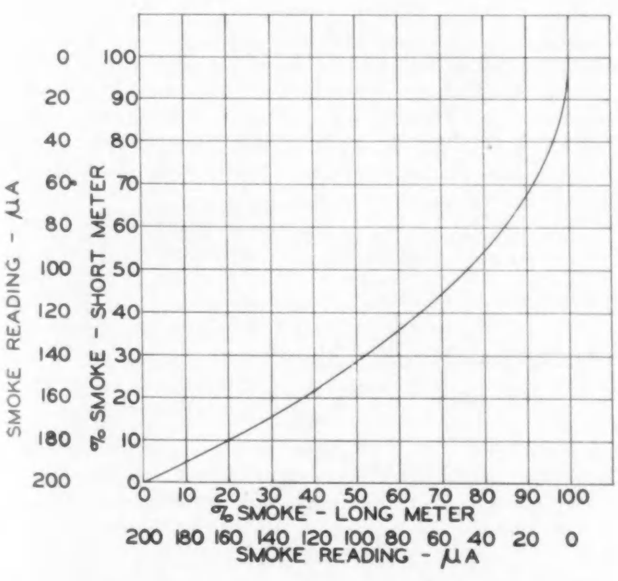
The light source consists of a 6-v flashlight-type bulb in an ordinary flashlight-type reflector, with a plane-glass window. Provision is made for focusing the bulb in the reflector (*K*, Fig. 3). The power is supplied from the regular 110-v, 60-cycle a-c line through a regulated-voltage type step-down transformer. The intensity of the light is controlled by a rheostat in series with the lamp. This light source provides ample light and is highly efficient, as is evidenced by the fact that the consumption of the lamp is less than 2 w as compared with the consumption of the usual light source for this service of around 60 w. While we are not primarily trying to save on our electric light bill, this low power consumption does have an advantage in that it allows us to use a smaller rheostat with smaller wire, which makes for finer control. Also, since we require only small units, we can make them of ample size for the power consumed, so that resistance change due to heating is reduced to a minimum. The net result is that we have an easily controlled and stable light source.

The photo-cell is a General Electric light-sensitive cell. The output is read on a 200 micro-amp ammeter. This combination has proved quite satisfactory.

The light source and photo-cell are connected to the measuring tube *A* by means of fiber adapters N_1 and N_2, (Fig. 3). The fiber tends to insulate the photo-cell from the measuring tube in case it accidentally should become overheated. (Normally the measuring tube only becomes warm.) The windows at each end are located in the adapter, held in place by screwed-in rings. The adapters are held on the measuring tube as shown in Fig. 5 by means of the rings *R*. To gain access to the windows, it is only necessary to loosen ring *R* with the special spanner wrench provided (*W*, Fig. 2) and lift off the adapter.

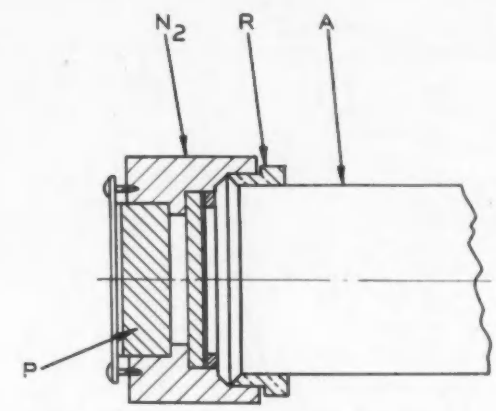
■ Simple Controls Employed

The controls of the smokemeter are very simple. There are only two – a combination switch and light-intensity control (*S*, Fig. 2), and the valve control handle (*H*, Fig. 2). The normal position of the valve control handle is at "Air." When it is desired to take a smoke reading (after the meter has been connected to the engine and the power line plugged in), the switch *S* is turned until a reading of 200 is obtained on the micro-ammeter, and the handle *H* turned to "Smoke." After waiting a few seconds for the reading to become stabilized, the micro-ammeter reading is taken, and the control handle *H* returned to "Air."



■ Fig. 4 – Relation between smoke readings on long and short smokemeter

Now that we have taken a smoke reading, the question is: What does it mean? Our reading is in micro-amperes. We may simply use this reading as an indication of our smoke; a high reading would indicate a relatively clean exhaust, a low reading would indicate relatively heavy smoke. In our own laboratory we have used these micro-ampere readings as a basis for plotting smoke against some other function.



■ Fig. 5 – Detail of smokemeter end connectors

Some laboratories use "% Smoke" as a measure of smoke. "% Smoke" is defined as:

$$\% \text{ Smoke} = \frac{A_0 - A}{A_0} \times 100$$

where A_0 = meter reading with clean air.
A = meter reading with smoke.

It will be observed that "% Smoke" bears a linear relationship to the micro-ampere reading.

Both of these methods of rating smoke have the disadvantage that their value for a given degree of smoke depends upon the length of the smokemeter used. This fact has already been pointed out, and illustrated by Fig. 4. Their value also depends to some extent upon the photo-cell micro-ammeter combination used, that is, what the relationship between light intensity and meter reading is for the combination.

We would like to suggest a measure of smoke that would be independent of the smokemeter used. The unit of smoke that we propose is "smoke density per unit length of smoke column."

To illustrate the properties of this unit, we shall begin with the standard definition of density. Density is defined as the common logarithm of the reciprocal of transparency (or opacity):

$$D = \log_{10} \frac{1}{T}$$

Transparency is the ratio of the intensity of the transmitted light to that of the incident light:

$$T = \frac{I}{I_0}$$

So $$D = \log_{10} \frac{I_0}{I}$$

where D = Density
T = Transparency
I_0 = Intensity of incident light
I = Intensity of transmitted light

To obtain the density of the column of smoke in our smokemeter, then, we must obtain the intensity of the light incident upon the smoke column, and the intensity of the light transmitted by the smoke column. If the photo-cell micro-ammeter combination used in our smokemeter gave a linear relationship between light intensity and meter reading, then we could use the initial clean air reading A_0 for I_0, and the smoke reading A for I in the preceding formula. However, if this linear relationship does not hold, then we would need a calibration curve of meter readings against light intensities to use the formula.

Now the density of a column of smoke may be assumed to be proportional to the length of the smoke column. This conclusion follows from the assumption that equal lengths of the same smoke column absorb the same fraction of the light incident upon them. Experimental evidence also seems to bear out this conclusion. Therefore, we may convert our value of smoke density in our smokemeter to the basis of the smoke density of unit length of smoke column by dividing this meter value of smoke density by the ratio of the effective length of the smokemeter to some standard length of smoke column. Then an absolute rating of that sample of smoke should be obtained which would be independent of the smokemeter used. If some other smokemeter of different length were used on the same smoke a different meter reading would be observed but, when converted to this standard unit, the same value should result.

To obtain a numerical value for this unit of smoke density that would be convenient to use, it is proposed that the standard length of smoke column be taken as 100 m (or a hectometer). This would give values of smoke density of 10, 20, 30, and so on, rather than 0.1, 0.2, 0.3, or 0.01, 0.02, 0.03, and so on, as would be the case if 10 m or 1 m were chosen as the standard length.

Fig. 6 shows smokemeter readings for two different lengths of smokemeters converted to smoke density. These readings were converted by means of the formula:

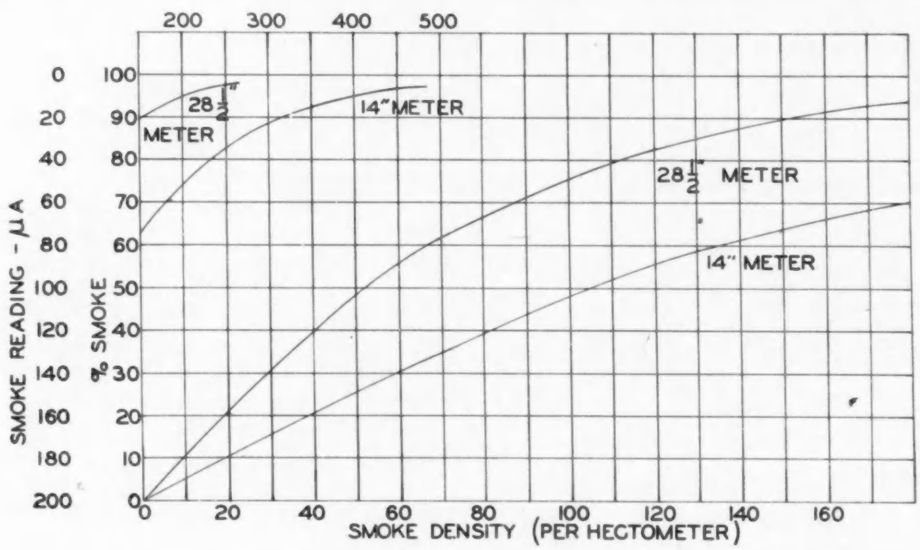
$$\text{Smoke density} = \frac{3937}{L} \log_{10} \frac{I_0}{I}$$

where L = Effective length of smokemeter in inches.

The light intensities were read from a calibration curve of the photo-cell.

To illustrate the use of this unit of smoke density, take, for example, a sample of smoke that gives a reading of 160 micro-amp (clean air equals 200 micro-amp) on a smokemeter of effective length equal to 28½ in. This reading would be equivalent to 20% smoke. From the curve, it is seen that the smoke density would be 19. Now, if this same sample of smoke were measured in a smokemeter of an effective length of 14 in., the reading would be 180 micro-amp, or 10% smoke. But, from its appropriate curve, the same smoke density of 19 would be obtained for the sample of smoke.

The universal use of this smoke-density scale should allow smoke readings to be interpreted the same by everyone.



■ Fig. 6 – Relation of smoke reading to smoke density

Thermal Conductivities Quoted Incorrectly in Spark-Plug Paper

In the latter part of Table 4 of the paper: "Ceramic Insulators for Spark Plugs," published on p. 241 of the June, 1940, SAE Transactions, the thermal conductivities in cgs units at 38 C of high-tension porcelain and mica (phlogopite) were given erroneously as 0.0003 and 0.0001, respectively. The correct values, as quoted in the "Electrical Engineer's Handbook," Vol. 5, by Pendlar and McIlwain, are 0.0025 for high-tension porcelain and 0.0012 for mica.

1940 ROAD DETONATION TESTS

(Compiled from Report[1] of The Cooperative Fuel Research Committee)

presented by

J. M. CAMPBELL,[2] R. J. GREENSHIELDS,[3] and W. M. HOLADAY[4]

THE 1940 CFR Road Tests have developed new information that can be used for the development of fuels and engines. Application of the principles worked out in these tests is expected to result in a more efficient utilization of fuel antiknock properties and more effective engine design and adjustment to meet the requisites of current motor fuels.

These tests indicate that the ASTM octane number alone, or even a road octane number as determined by methods heretofore widely used, does not give sufficient information for present needs relative to fuel behavior in service. Neither do test methods previously used provide sufficient information concerning the fuel requirements and knocking characteristics of engines. The new methods of approach which have been developed furnish needed information relative to the fuel and engine relationship that heretofore has been obscure, and indicate paths for future developments.

The entire program has been supported generously by the cooperating organizations and their respective representatives who actively participated in the tests. Various phases of the work have extended over most of the past year and, in the concluding and most important stage – the centralized road tests at San Bernardino, Calif. – 32 organizations were represented. In these concluding tests, a study was made of the characteristics of 23 fuels representing a wide variety of fuel types and the complimentary behavior of 24 different automobiles. During this latter period alone, the 24 test cars were driven over 100,000 miles.

OVER the past ten years the Cooperative Fuel Research Committee, which is sponsored by the Automobile Manufacturers Association, the American Petroleum Institute, the Society of Automotive Engineers, and the National Bureau of Standards, has given considerable attention to the problem of evaluating the knocking characteristics of motor fuels, both by laboratory tests and by actual road tests. The results of these investigations have been made public from time to time through the publications of the Society of Automotive Engineers and those of the American Petroleum Institute[5].

Basically what has been sought is some procedure or procedures for the evaluation of the knocking characteristics of different motor fuels in service and the corresponding behavior of engines. This program was divided into two parts: Part I, in which a number of cooperating laboratories each rated a selected group of fuels by whichever procedure they were then using; and Part II, consisting of centralized tests made after completion of Part I, and in which an effort was made to find an improved approach to the entire problem.

PART I – COOPERATIVE EXCHANGE ROAD TESTS

The purpose of this part of the program was to obtain some facts about the type of information that was being supplied by the road-test procedures then in use. If the road ratings obtained independently by different laboratories did not agree within reasonable limits, then there could be no question as to the desirability of further intensive work on the problem. On the other hand, if the results obtained in this way showed substantial agreement among different laboratories, then there might be considerable question as to the necessity of conducting extensive centralized tests.

Invitations to participate in this part of the program were issued on Feb. 7, 1940, to members of the Cooperative Fuels Research Committee, the CFR Motor Fuels Division, the CFR Motor Fuels Exchange Group, the Pacific Coast Technical Group, and to all participants in previous CFR road tests. This invitation carried with it a questionnaire requesting each company to list the type of gasolines that they would be in a position to supply for the project with

[This paper will be presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 1-6, 1941; it contains a portion of the discussion: "1940 Cooperative Road Tests – A Report from the Cooperative Fuel Research Committee," presented by J. M. Campbell at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1941.]

[1] Copy of the Report of the Cooperative Fuel Research Committee on the 1940 Detonation Road Tests may be obtained from C. B. Veal, secretary, Cooperative Fuel Research Committee, Society of Automotive Engineers, 29 West 39th St., New York, N. Y.

[2] Chairman, Motor Fuels Division, Cooperative Fuel Research Committee.

[3] Leader, Detonation Road Test Analysis Group, Cooperative Fuel Research Committee.

[4] Leader, Detonation Road Test Project, Cooperative Fuel Research Committee.

[5] See SAE Journal, Vol. 24, February, 1929, pp. 212-123: "Standard

(*Footnote continued on following page*)

Table 1 – Average of Individual Car Ratings Reported By Various Companies in Exchange Road Tests. Part I

General Description of Road-Test Method	Laboratory	No. of Cars	Road Rating – Octane Number 23P	24P	25P	26P	27P	28P	31R	32R	33R	34R	35R	36R	37R	38R	39R	Ave. Dev. from CFR Average
CFR	A	6	78.9	82.4	73.9	84.9	87.2	82.8	70.7	75.8	71.8	74.8	70.0	77.3	73.5	74.1	75.6	1.02
CFR	B	4	....	....	71.9	....	....	....	70.4	68.7	68.1	72.1	72.2	75.5	71.7	69.8	74.8	2.21
CFR	C	6	78.3	80.3	73.8	....	....	80.3	70.7	77.3	71.1	73.4	70.2	74.8	....	74.2	74.3	1.00
CFR	D	4	80.3	82.3	75.0	79.3	80.6	79.3	71.1	70.2	69.2	72.0	72.0	77.0	73.1	72.1	74.6	1.59
CFR	E	3	81.2	83.3	76.0	83.3	80.5	80.7	70.7	70.2	71.3	73.7	70.2	77.3	74.0	69.2	76.8	1.54
CFR	F	18	79.8	82.0	75.9	82.4	82.3	80.1	71.5	73.3	71.8	73.9	71.5	76.0	72.9	72.4	75.4	0.65
CFR	G	3	82.0	83.5	73.0	80.5	81.5	76.5	70.5	74.5	70.0	72.8	76.5	74.8	71.8	72.3	73.3	2.04
CFR	H	6	79.9	81.6	75.7	83.6	84.8	81.4	71.0	75.5	72.6	75.1	72.0	77.5	74.6	75.8	76.5	0.85
CFR	I	4	78.3	79.7	72.6	82.6	83.7	79.6	70.0	74.8	72.0	73.6	69.3	76.0	71.9	74.7	74.6	0.98
CFR	J	4	78.6	80.1	73.1	85.0	88.9	81.7	69.7	77.0	72.3	73.8	71.6	76.0	74.2	75.2	76.2	1.27
CFR	K	3	80.7	82.3	77.3	85.0	90.7	83.0	73.3	81.0	73.7	76.7	73.3	80.0	74.7	73.3	80.0	2.72
CFR	L	6	79.5	82.0	73.5	84.0	85.0	82.0	71.0	73.0	71.5	73.0	70.0	77.0	73.0	75.0	77.0	0.89
CFR	M	1	81.0	82.0	76.0	83.5	89.5	81.5	71.5	78.0	74.0	75.0	72.5	77.0	73.5	75.0	75.5	1.33
CFR at Trace Spark	N	8	77.8	78.8	74.8	84.0	84.1	83.3	69.5	77.5	74.8	75.2	69.5	76.5	74.0	76.4	76.6	1.62
CFR at Trace Spark	O	2	78.5	78.6	74.2	81.3	80.8	75.3	73.0	74.5	72.0	73.7	72.0	73.8	73.7	72.1	73.9	1.59
Aggregate	P	2	80.0	80.9	68.7	77.7	81.4	77.7	72.3	....	66.9	68.7	73.1	75.2	70.3	69.9	72.1	2.76
Aggregate, 10-45 mph	O′	4	80.1	81.4	73.7	81.8	83.5	77.8	72.7	73.6	70.6	71.8	72.6	75.1	71.4	71.9	73.8	1.33
Aggregate, 10-70 mph	O′	4	80.4	81.8	72.9	79.1	80.7	77.9	74.2	71.0	68.7	69.8	72.6	74.1	71.7	69.2	72.4	2.63
Aggregate	K′	3	81.5	83.5	77.2	83.3	89.8	83.2	74.0	78.5	71.8	76.0	73.2	80.0	75.0	73.8	79.3	2.35
Aggregate	R	5	80.9	79.9	74.1	80.3	77.9	77.8	72.6	72.8	70.5	73.8	72.8	76.1	74.7	72.4	74.9	1.63
Integ. Knock Intensity	T	3	80.8	82.7	74.4	79.6	83.4	78.2	72.6	75.7	72.1	73.5	72.6	77.0	74.4	73.0	75.4	1.07
Knock Intensity 20 mph	Q	3	83.0	85.0	....	85.0	89.0	79.0	73.0	77.0	73.0	73.0	73.0	75.0	73.0	75.0	75.0	
Knock Intensity 40 mph	Q	3	81.0	83.0	....	79.0	83.0	79.0	75.0	73.0	69.0	75.0	75.0	81.0	73.0	73.0	75.0	
Knock Intensity 60 mph	Q	3	79.0	....	....	75.0	73.0	75.0	79.0	71.0	69.0	73.0	79.0	81.0	75.0	69.0	77.0	
Aggregate Grand Average	O″	4, 2	79.3	80.5	73.1	80.6	81.9	77.4	72.0	72.5	69.6	71.4	71.3	74.2	71.5	70.9	72.8	1.86
Borderline	K″	3	80.0	82.0	72.3	78.0	78.7	77.3	68.7	72.7	70.3	72.3	68.0	75.7	72.0	72.7	73.0	2.20
CFR Method	Average		79.7	81.2	74.4	83.0	84.5	80.4	71.0	74.8	71.7	73.9	71.5	76.4	73.3	73.4	75.7	
CFR Method	High Rating		82.0	83.5	77.3	85.0	90.7	83.3	73.3	81.0	74.8	76.7	73.3	80.0	74.7	75.8	80.0	
CFR Method	Low Rating		77.8	78.6	71.9	79.3	80.5	75.3	69.5	68.7	68.1	72.0	69.3	73.8	71.7	69.8	73.3	
CFR Method	Spread		4.2	4.9	5.4	5.7	10.2	8.0	3.8	12.3	6.7	4.7	4.0	6.2	3.0	6.0	6.7	
All Methods	High Rating		83.0	85.0	77.3	85.0	90.7	83.2	79.0	81.0	74.8	76.7	79.0	81.0	75.0	76.4	80.0	
All Methods	Low Rating		77.8	78.6	68.7	75.0	73.0	75.0	68.7	68.7	66.9	68.7	68.0	73.8	70.3	69.2	72.1	
All Methods	Spread		5.2	6.4	8.6	10.0	17.7	8.2	10.3	12.3	7.9	8.0	11.0	7.2	4.7	7.2	7.9	
ASTM Octane No.			81.1	82.6	73.5	77.8	82.5	75.9	76.5	71.2	69.5	72.7	75.4	76.7	74.0	71.1	74.3	
Research 1939 Octane No.			82.2	84.2	79.2	86.3	94.4	83.2	76.9	82.9	76.4	78.8	75.3	82.1	79.2	79.2	80.2	

the understanding that each laboratory might be called upon to furnish one or two gasolines. On the basis of the response to these questionnaires a group of 15 fuels ranging from 70 to 83 ASTM octane number was selected. For the most part, each cooperating company supplied one fuel to all the cooperating laboratories and received in return one drum of all the other experimental fuels from the other suppliers. For this reason these tests have been referred to as the cooperative exchange road tests.

Results of Part I, Cooperative Exchange Tests – The findings of the group appointed to analyze the data are summarized in Table 1.

Typical of the variations reported in different cars, even among different units of the same make, are those given by one company in Table 2. It is obvious from data of this kind that the road ratings as reported in this part of the program by any one company on a limited number of cars are largely affected by chance. It is not surprising, therefore, that the average road ratings reported by each laboratory and listed in Table 1 show considerable variation among different laboratories. For example, the 15 laboratories employing the original CFR road-test method with minor changes show differences in the road ratings of individual fuels varying from 3.0 to 12.3 octane numbers.

The magnitude of these variations is dependent to some extent on the characteristics of the fuel itself and to some extent on the chance selection of cars. Laboratory F, which used 18 cars altogether, had the smallest average deviation from the average and seldom differed from the average by more than 1 octane number. On the other hand, laboratories using only from one to four cars sometimes deviated from the average in their interpretation of fuel behavior by as much as 4 or 5 octane numbers. All this follows, of course, from the statistical nature of this method and

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(*Footnote continued from previous page*)
Engine for Fuel Tests"; Vol. 29, August, 1931, pp. 164-165: "Tentative Recommended Practice for Conducting Antiknock Tests"; See also SAE Transactions, March, 1933, pp. 105-120: "Antiknock Research Coordinates Laboratory and Road Tests," presented by C. B. Veal, H. W. Best, J. M. Campbell, and W. M. Holaday; May, 1935, pp. 165-179; "CFR Committee Report on 1934 Detonation Road Tests," presented by C. B. Veal; February, 1938, pp. 63-72: "Effect of Test Conditions on Fuel Rating," presented by A. E. Becker; June, 1938, pp. 244-252: "1937 Road Knock Tests," presented by T. A. Boyd; October, 1938, pp. 416-420: "Correlation of Road and Laboratory Octane Numbers," presented by J. R. Sabina; June, 1939, pp. 277-280: "CFR Research Method of Tests for Knock Characteristics of Motor Fuels."

See also Proceedings of the American Petroleum Institute, Section III on Refining, January, 1930, pp. 32-37: "Standardizing of Antiknock Testing," by H. L. Horning; December, 1930, pp. 38-45: "Progress Toward a Uniform Method of Measuring Detonation"; December, 1931, pp. 46-59: "The CFR Apparatus and Method for Knock Testing"; December, 1932, pp. 139-153: "Correlation of CFR Laboratory Knock Ratings with Behavior of Motor Fuels in Service," presented by C. B. Veal, H. W. Best, J. M. Campbell, and W. M. Holaday; November, 1934, pp. 33-54: "The 1934 Detonation Road Tests," presented by C. B. Veal; November, 1937, pp. 98-112: "1937 Road Knock Tests," presented by T. A. Boyd; November, 1937, pp. 112-125: "Effect of Test Conditions on Fuel Rating," presented by A. E. Becker.

Table 2 – Comparison of Road Knock Ratings Reported by Laboratory F, on One Fuel (32R) in 18 Different Units of Low-Price Cars

Make	Road Rating (Equivalent ASTM Octane No.)
1939 Car Make A	77
1939 Car Make A	70
1939 Car Make A	73
1940 Car Make A	75
1940 Car Make A	77
1940 Car Make A	75
1939 Car Make B	75
1939 Car Make B	73
1939 Car Make B	77
1940 Car Make B	71
1940 Car Make B	74
1940 Car Make B	72
1939 Car Make C	71
1939 Car Make C	71
1939 Car Make C	69
1940 Car Make C	73
1940 Car Make C	75
1940 Car Make C	71
Average	73.3

indicates how easily a misinterpretation of fuel behavior can be made from limited data.

On the basis of this information, it is evident that antiknock quality as expressed by road ratings by any method then current varied from laboratory to laboratory by a significant amount. Because of this situation, a plan to conduct a series of centralized road knock tests at San Bernardino, Calif., was approved by the Cooperative Fuel Research Committee on Sept. 6, 1940.

PART II – CENTRALIZED ROAD TESTS

The purpose of these tests was to study cooperatively the problem of evaluating the knock characteristics of fuels and engines. Invitations to participate in these tests were again sent out to the membership of the Cooperative Fuel Research Committee, the CFR Motor Fuels Division, the CFR Exchange Group, and the Pacific Coast Technical Group. Invitations were also extended to the Ordnance Department and to the Quartermaster Corps of the U. S. Army, and to the principal manufacturers of ignition equipment and carburetors for passenger-car engines.

The following gasolines, representing a number of fuel types, were used in the tests:

Straight-Run + Catalytically Cracked
California Straight-Run
Mid-Continent Straight-Run + Cracked
Mid-Continent Straight-Run + Cracked
Catalytically Cracked
Highly Olefinic Polymer
Mixed Crude, Highly Aromatic
Mid-Continent Straight-Run + Cracked Reformed
Mid-Continent Cracked + Straight-Run
Benzol Blend, 40% Benzol
Two Special Blends of Olefinic + Paraffinic Fuels
Natural Gasoline
Mid-Continent Cracked
California Cracked
California Straight-Run + Cracked
Paraffinic Straight-Run
Mid-Continent Straight-Run + Cracked
Mid-Continent Straight-Run + Cracked
Mid-Continent Straight-Run
71 Octane No. Reference Fuel (A6 + C12)
80 Octane No. Reference Fuel (C12 + F3)
Mid-Continent Straight-Run + Alkylate

Table 3 – Summary of Inspection Data on Test Fuels Used in the 1940 CFR Centralized Road Tests

Fuel No.	Octane Number ASTM*	Octane Number Research 1939*	Research Minus ASTM	Lead Content, cc TEL/gal	Reid Vapor Pressure, lb/in²	Gravity, deg API	ASTM Distillation IBP	10%	50%	90%	EP
1-B	65.0	65.1	0.1	0.8	8.3	62.8	95	146	233	350	395
2-B	64.9	68.7	3.8	0.0	9.6	62.5	91	136	228	341	412
3-B	71.0	73.4	2.4	0.0	6.1	64.0	119	163	208	290	365
4-B	74.9	73.9	−1.0	1.4	7.3	61.3	104	157	235	323	366
5-B	76.3	76.5	0.2	0.8	7.4	72.8	114	140	167	205	255
6-B	69.2	77.1	7.9	0.0	7.6	57.1	95	151	255	343	387
7-B	71.1	79.5	8.4	0.0	9.2	58.9	90	131	234	361	414
8-B	72.1	77.3	5.2	1.5	8.6	60.5	95	143	246	349	397
9-B	71.0	77.6	6.6	0.6	6.8	58.7	105	151	244	348	424
10-B	70.6	83.0	12.4	0.0	9.0	58.8	93	129	227	369	419
11-B	71.6	78.3	6.7	1.3	8.4	60.2	95	138	231	347	397
12-B	70.4	75.5	5.1	0.0	4.5	45.8	135	173	209	373	414
13-B	73.4	79.9	6.5	0.3	8.7	58.4	89	137	255	363	397
14-B	75.6	81.7	6.1	1.7	9.4	62.8	90	133	224	340	387
15-B	80.1	81.2	1.1	2.0	8.0	59.8	95	148	239	325	377
16-B	80.8	81.5	0.7	0.0	6.7	68.6	107	156	206	265	383
17-B	80.0	83.2	3.2	0.0	5.8	65.8	117	160	199	230	288
18-B	75.2	83.2	8.0	0.0	8.7	53.7	94	147	244	355	298
19-B	76.5	86.3	9.8	0.0	10.7	55.9	90	127	267	384	412
20-B	81.7	95.0	3.3	0.0	9.0	56.1	97	143	232	338	404
21-B	82.0	83.8	1.8	2.9	9.6	65.0	95	136	211	286	337
22-B	81.7	93.8	12.1	0.0	10.7	69.1	88	117	197	351	374
23-B	85.0	97.0	12.0	0.8	10.7	69.1	88	117	197	351	374

* Average of three engines
Note: All data are as of Nov. 14, 1940.

Table 4 – Cars Used in Centralized Road Tests

Car	Model	Year
Buick	60	1941
Buick	40	1941
Buick	40	1940
Cadillac	60 Special	1939
Cadillac	60 Special	1941
Chevrolet	Master	1940
Chevrolet	Special Deluxe	1941
Chevrolet	Special	1940
Chevrolet	Deluxe	1941
*Chrysler 8	C-26	1941
*Chrysler 8	C-26	1940
*Chrysler 6	C-28	1941
*Chrysler 8	C-30	1941
Dodge	D-14	1940
Ford	85	1940
Ford	85	1940
Ford	85	1941
**Oldsmobile	L-40	1940
Packard 6	110	1940
Plymouth	P-9	1940
Plymouth	P-10	1940
Plymouth	P-12	1941
Pontiac 6	40-25	1940
Studebaker 6	Champion	1941

* Equipped with Fluid Drive.
** Equipped with Hydra-Matic Transmission.

A summary of the inspection data for these fuels is given in Table 3.

The test cars are listed in Table 4.

In selecting these cars an attempt was made to obtain a wide representation among 1940 and 1941 models, including from three to four different units of each make in the low-price group.

■ Approach to the Problem

The tests conducted in Part I indicated the need for new techniques by means of which more complete and more reliable information relative to fuel knock characteristics in service can be obtained in a practical way by one laboratory without the necessity of using a large number of test cars in order to obtain sufficient data to be significant. One of the most important factors affecting knock characteristics, which was not taken into account by the earlier procedure, is engine speed[6]. Since the engine speed at which a given fuel will knock in a given engine is largely determined by ignition timing, the problem appeared to resolve itself into a study of the relationship between ignition timing, engine speed, and knocking tendency.

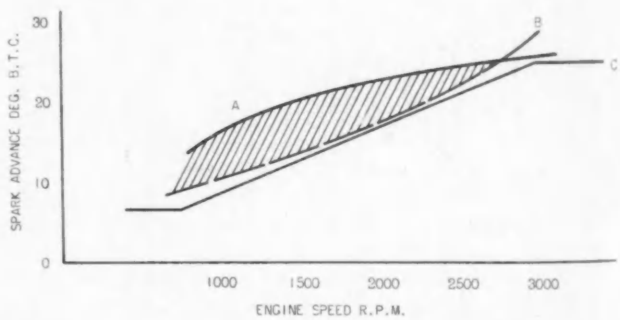
One of the most practical means of examining this relationship appeared to be that first suggested by Greenshields and Hebl in 1937[7], and later further developed and proposed as a road test technique by Drinkard and Macauley in 1938[8]. Accordingly, the group participating in the centralized tests began by thoroughly testing this general method of approach in a wide variety of cars and, except for minor variations, ultimately adopted this as one road-test procedure for obtaining basic information from which a number of important relationships between any given fuel and engine can be determined.

In order that the concepts involved in this new method of approach to the detonation problem may be understood readily, a series of four illustrative diagrams have been prepared. These diagrams represent typical relationships between ignition timing, engine speed, power output, and fuel knocking tendency, but do not represent any particular engine. They have been somewhat simplified and drawn for illustrative purposes only.

The relationship between ignition timing for maximum power and engine speed is a fundamental one and is illustrated by curve *A* in Fig. 1. The position of this curve varies from one engine to another and depends upon such factors as combustion-chamber shape, spark-plug location, and turbulence. In any given engine, however, tests made on a chassis dynamometer at San Bernardino indicated no perceptible difference in ignition timing for maximum power between the most highly olefinic fuel and technical iso-octane under non-knocking conditions at speeds between 10 and 50 mph. These same tests indicated that pure benzene required from 2 to 5 deg less spark advance for maximum power than the aforesaid fuels.

Another fundamental type of curve is that illustrated by curve *B*, Fig. 1. This curve represents the maximum spark advance which is permissible with a particular fuel for knock-free operation. If the engine is operated at the spark advance for maximum power with this fuel, detonation of various intensities will occur within the shaded area. On the other hand, if the ignition timing is controlled by means of an automatic device to conform with curve *C*, there would be no detonation at any speed. All this, of course, is elementary and is well known.

What is not so well known and widely appreciated is the type of information that is illustrated in Fig. 2. In this figure are represented two fuels that have different characteristic responses to changes in engine speed and ignition timing. For purposes of illustration, these gasolines may be considered to have identical ASTM octane numbers. The curves representing each fuel again indicate the spark advance at which knocking begins and the outstanding fact is that, at low engine speeds, Fuel 1 will not permit the spark to be advanced as far as Fuel 2. On the other hand, at higher engine speeds, the relationship is reversed and Fuel 2 will not permit nearly so much spark advance as Fuel 1.



■ Fig. 1 – Typical relationship between ignition timing for maximum power, distributor advance, and borderline knock curve for one fuel

[6] See SAE Transactions, May, 1935, pp. 165-179: "CFR Committee Report on 1934 Detonation Road Tests," by C. B. Veal; see also April, 1937, pp. 144-150: "Relative Knocking Characteristics of Motor Fuels in Service," by J. M. Campbell, W. G. Lovell, and T. A. Boyd.

[7] See SAE Transactions, April, 1937, pp. 148-150: "Additional Service Data on Effect of Ignition Timing," discussion by R. J. Greenshields and L. E. Hebl; see also May, 1939, pp. 210-220: "Spark Timing – Its Relation to Road Octane Numbers and Performance," by L. E. Hebl and T. B. Rendel.

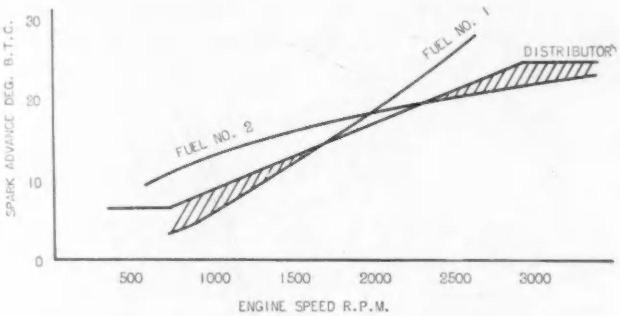
[8] See SAE Transactions, October, 1938, pp. 436-440: "Spark Advance and Octane Number – A Road-Test Technique," by W. E. Drinkard and J. B. Macauley, Jr.

This is an important concept because it illustrates a significant characteristic of motor fuels which is not indicated, satisfactorily at least, by any of the tests which have been applied customarily in the past. When these characteristics are considered in relationship to the automatic spark advance given by a distributor as shown in Fig. 2, the result is that Fuel 1 will knock only at low engine speeds, and Fuel 2 will knock only at high engine speeds.

Considered in relation to the spark timing for maximum power, these characteristics indicate that some types of fuels will permit the spark to be advanced closer to that required for maximum power than will others.

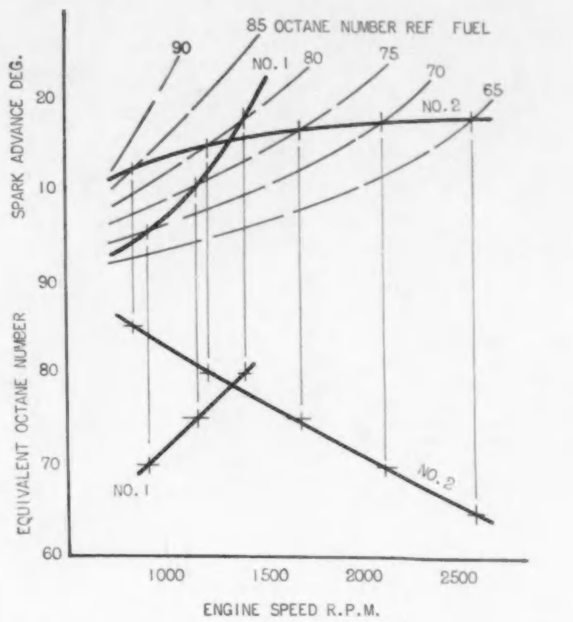
The technique that has been developed for the actual determination of these characteristics of gasoline in a given engine involves first, the construction of a reference-fuel framework representing the permissible ignition timings for fuels of different octane number such as the one illustrated in the upper section of Fig. 3. Reference fuels provide a common basis for comparison which can be used either to maintain a continuity over a period of time in one car or else to provide a common basis for comparisons made between different cars or between different laboratories. The reference fuels which were used in this investigation were the current standard reference fuels A-6, C-12, and F-3.

The broken lines in Fig. 3 represent the ignition timings at which knocking begins with a series of reference fuels of graduated octane number as indicated by the numbers attached to each curve. Superimposed on this framework are curves representing the characteristics shown by Fuels 1 and 2 in Fig. 2. In order to express the divergent characteristics of these two fuels in some terms which are independent of engine variables, the equivalent octane number may be determined either by interpolation with respect to the standard reference fuel framework at several successive intervals in speed or by a graphical construction based on the intercepts of the reference fuel framework and the fuels under examination. (By equivalent octane number is meant the ASTM octane number of the reference fuel which is matched under the particular engine conditions in question.)



■ **Fig. 2 – Borderline knock curves showing distinguishing characteristics of different fuels**

The graphical construction from which the equivalent octane number at various speeds can be determined is shown in Fig. 3. The speed at which each fuel intercepts a reference fuel line is determined and is then plotted in the lower half of the figure against the corresponding octane number shown on the vertical axis.



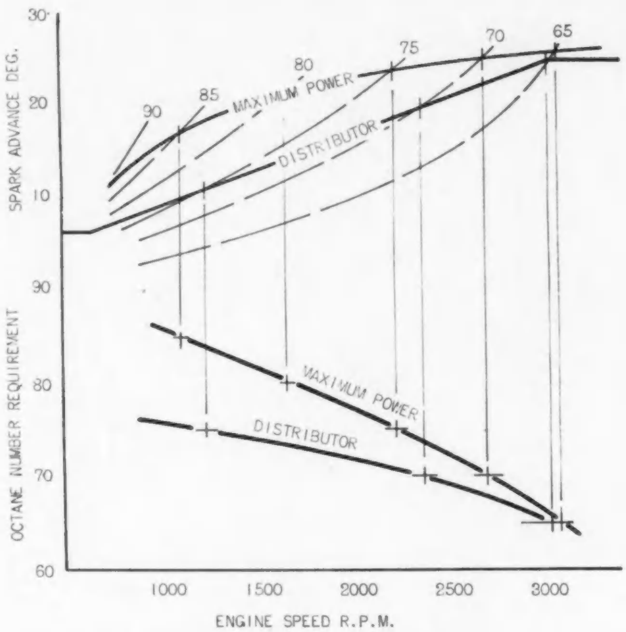
■ **Fig. 3 – Determination of octane number versus speed relationship**

Similarly by interpolation at different speeds, at 1000 rpm Fuel 1 is equivalent to a reference fuel having an octane number of 71. At 1250 rpm the equivalent octane number is 76 and, at 1500 rpm, it is 81.

The curves representing Fuels 1 and 2 in the lower section of Fig. 3 clearly distinguish the different characteristics of these two fuels. This difference is expressed in terms of equivalent octane numbers which have a more tangible meaning to the fuel technologist than an expression given in terms of spark advance. The latter is of more direct significance in connection with the utilization of any particular fuel in some particular engine.

The octane number versus speed relationship shown in Fig. 3 can be used in several ways. It can be used to distinguish between different fuel characteristics such as are shown by Fuels 1 and 2 in the example shown. It may be used to distinguish a difference between different engines, or it may be used as an intermediate step in the determination of average fuel trends in a number of different engines. That is to say, by combining the curves representing the octane number versus speed relationship for a single fuel in a number of different engines, it is possible to obtain a mean curve which is indicative of fuel characteristics based on results obtained in more than one engine.

The reference-fuel framework may be used further to express the fuel requirements of the engine, again in terms of the ASTM octane number at different speeds. Fig. 4 illustrates the method for doing this. The distributor curve and the curve representing spark advance for maximum power are superimposed upon the reference-fuel framework. The intercepts made by the reference fuels and the distributor curve or the curve representing spark advance for maximum power then determine the octane requirement of the engine at the corresponding engine speeds.



■ **Fig. 4 – Determination of octane-number requirement**

The octane numbers at the intercepts may then be plotted as shown in the lower section of Fig. 4, to produce a figure representing the octane requirement of the engine when operating either in accordance with a given distributor curve or a given spark timing for maximum power.

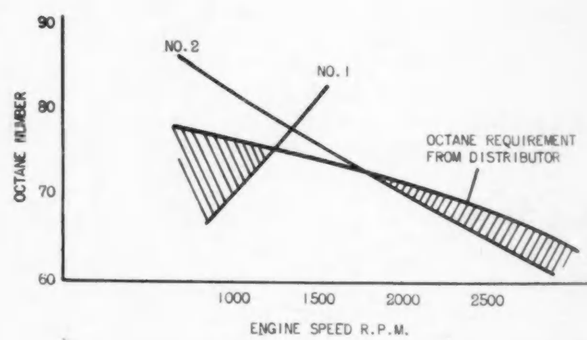
By superimposing the curves represented in the lower sections of Figs. 3 and 4, it is possible to construct a diagram which is independent of ignition timing, and which will indicate whether a given fuel will knock and in what speed range in a given engine. Such a diagram is shown in Fig. 5, and is useful in certain types of analysis.

■ Road Test Procedures[9]

The San Bernardino data given in this report were obtained during wide-open throttle acceleration on a slight upgrade (2-3%) and at atmospheric temperatures between 60 and 75 F. Each car was fitted with a spark indicator either of the electronic indicating type, or of the stroboscopic type with which the spark timing was read directly from the driving seat by observation of an illuminated scale attached to the flywheel. The connection for vacuum spark advance was disconnected from the distributor and the automatic advance mechanism was locked. Provision was then made for manual advance of the spark from the driver's seat.

Spark-advance curves representing incipient or borderline detonation were then determined on reference fuels and on the 23 test fuels either in accordance with the "Borderline Procedure" described in the previous publications which have been referred to or by a modification of this procedure, called "Modified Borderline Procedure," in which the spark was advanced manually during each acceleration in such a way as to keep the knock approximately constant at some light intensity. The spark advance at frequent intervals of speed was then observed and recorded during the accelerating period. To facilitate the rapid measurement of ignition timing during acceleration, readings of spark advance were taken by means of a calibrated dial attached to the shaft which was used to control the spark advance. Although the numerical values for spark advance that were recorded by the latter procedure differed from those obtained by the former or "borderline" procedure at corresponding speeds due to the difference in knock intensity that was used, the relative positions of different fuels were unaffected by the differences in the two methods. As a result, within the limits of knock intensity used in this work, knock intensity itself had no appreciable effect on the relative standing of different fuels.

A certain amount of data was obtained using a "Knock-Intensity Procedure" which was developed as a method of obtaining data on cars where extensive instrumentation was not feasible. Using this procedure, the antiknock value of fuel throughout the speed range can be determined in terms of equivalent octane numbers by comparing the knock intensity of the fuel with that of reference fuels at several speeds. Also an "Octane Requirement Procedure" has been developed for the purpose of determining octane



■ **Fig. 5 – Comparison of equivalent octane-number curves with curve representing the octane requirement with a given distributor – The shaded area indicates where each fuel will knock**

requirement as related to speed in a large number of cars as found in service.

Present indications are that different procedures each adapted to the analysis of some particular phase of the problem are required, although further information and experience are needed to evaluate definitely the relative merits of each.

■ Results of San Bernardino Tests[10]

Typical data representing results obtained in the three volume-production cars, will serve to illustrate the type of information obtained. Data on 1941 models of these three

[9] Complete information regarding the road-test procedures may be obtained from C. B. Veal, secretary, Cooperative Fuel Research Committee, Society of Automotive Engineers, 29 West 39th St., New York, N. Y.

[10] Final analysis of the data obtained in Parts I and II of the 1940 CFR Road Tests was carried out by the Detonation Road Test Analysis Group: R. J. Greenshields, Leader, Shell Oil Co.; H. W. Best, Yale University; K. Boldt, Pure Oil Co.; F. C. Burk, Atlantic Refining Co.; J. M. Campbell, General Motors Corp.; W. E. Drinkard, Chrysler Corp.; H. J. Gibson, Ethyl Gasoline Corp.; C. B. Kass, Standard Oil Development Co.; W. S. Mount, Socony-Vacuum Oil Co.; J. G. Moxey, Sun Oil Co.; B. R. Siegel, Sinclair Refining Co.; C. B. Veal, Society of Automotive Engineers.

different makes are shown in Figs. 6, 7, and 8. Since the data shown in these figures were obtained in only one unit of each make, they are not necessarily representative of all units of that make. However, study of these figures indicates that these three engines have several characteristics in common where fuel requirements are concerned.

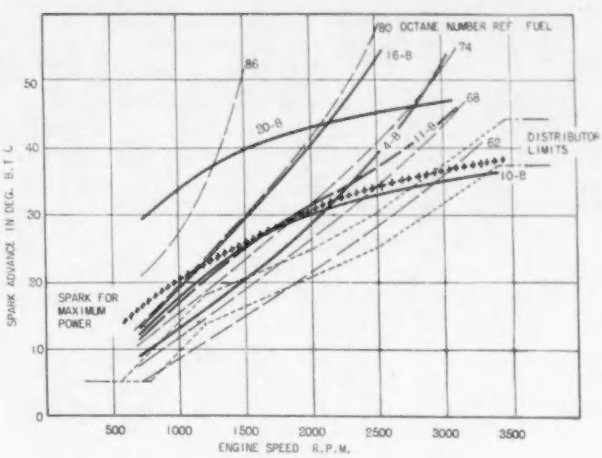
One outstanding characteristic, of course, is the slope of the reference-fuel framework upward as the speed is increased. Hardly less significant is the fan-like shape of the reference-fuel curves, tending to spread out at higher speeds. If the distributor happens to have been set so the engine knocks only at very low speed, a change of 1 or 2 deg in initial spark timing will produce a much greater change in octane requirement as expressed by reference fuels than the same change in initial spark timing in a car in which the distributor happens to have been adjusted to produce knock at higher speeds. This same general pattern of behavior was shown by all 24 test cars.

Superimposed on Figs. 6, 7, and 8 is the curve representing ignition timing for maximum power. In each case the road octane requirement for maximum power, determined at the intercepts between the curves representing ignition timing for maximum power and reference fuel borderline knock curves respectively, was between 75 and 85 at speeds below 1000 rpm whereas, at speeds above 2500 rpm, the road octane requirement was from 60 to 68. This characteristic, namely a decreasing octane requirement with increasing engine speed, was also shown by all engines, although there were wide differences between different engines in octane requirement at any particular speed.

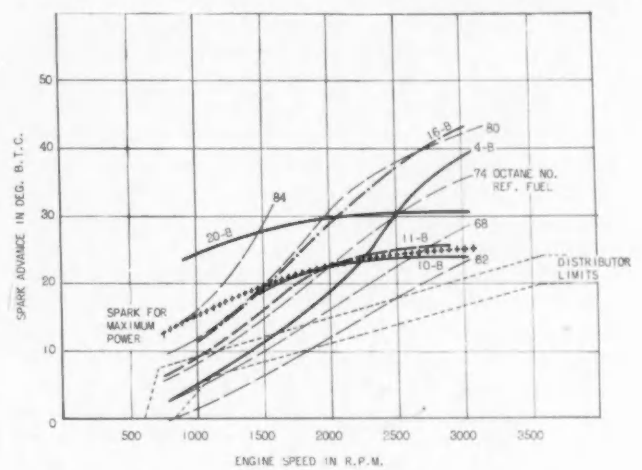
Different types of gasoline show common patterns of behavior in all three engines that are distinctly different from those of the reference fuels. Fuels 4-B, 10-B, and 11-B are three different types of gasoline of approximately 70 octane number and Fuels 16-B and 20-B are gasolines of approximately 81 octane number. Of the first three, Fuels 4-B and 10-B represent somewhat extreme types of behavior. Fuel 11-B, which is a mixture of straight-run and cracked gasoline containing lead tetraethyl, has characteristics intermediate between those of 4-B and 10-B.

The curve for Fuel 10-B is of particular interest because it so closely conforms to the curve representing ignition timing for maximum power in all three engines. This is a characteristic which may be of considerable significance from the standpoint of fuel utilization in engines and in the determination of future trends in fuel development. The reference-fuel data all indicate that, once the compression ratio has been fixed, straight-run gasolines knock more at low speeds than at high speeds when the spark is advanced for maximum power. This same characteristic is also shown by most commercial fuels of which 11-B is one example, and it is partly for this reason that most distributors are designed to produce a retarded spark – relative to that for maximum power – at lower engine speeds and full throttle. On the other hand, the characteristics shown by Fuel 10-B indicate that a more effective utilization of fuels may be possible through control of fuel characteristics along these lines. Of course, there are economic factors as well as the influence of combustion-chamber deposits and tendencies to preignite, that have not been taken into consideration in this discussion that would have to be taken into account in any such development.

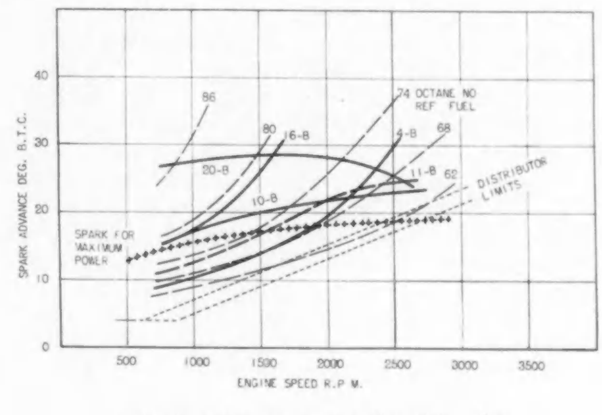
Fuel 4-B, which is representative of certain types of fuels sold commercially, is of interest because, even though the ASTM octane number of this fuel is from 4 to 5 units higher than that of 10-B, the latter fuel is better than 4-B



■ Fig. 6 – 1941 car No. 7 after 8000 miles



■ Fig. 7 – 1941 car No. 11 after 10,000 miles



■ Fig. 8 – 1941 car No. 24 after 3000 miles

Table 5 – Road Octane Number Summary – Centralized Tests

Road Octane Number of Fuel No. 4-B at Various Speeds in All Cars

	Car No.	Year	No. Groups	Engine rpm 750	1000	1500	2000	2500	3000
	6	1941	1	71.0	71.0	72.0	72.0	71.0	....
	7	1941	2	71.0	69.5	70.5	71.5	73.5	74.5
	16	1940	1	69.0	71.0	71.0	71.0	72.0	....
Low-	21	1940	1	69.0	69.0	....	69.0	73.0	....
Price	24	1941	2	66.0	67.5	68.0	68.5	70.5	....
Group	17	1940	1	70.0	68.0	70.0	....	....	....
	11	1941	2	68.0	68.0	70.5	70.5	74.5	77.0
	2	1940	2	70.5	70.5	70.5	70.5	71.0	71.0
	22	1940	1	65.0	68.0	71.0	74.0	76.0	77.0
	5	1941	2	71.0	69.0	70.5	70.0	70.5	....
	8	1940	.	....	....	....	....	....	....
	19	1940	2	69.0	68.0	72.0	70.0	70.5	....
	13	1940	1	66.0	69.0	71.0	72.0	75.0	....
	10	1940	1	....	71.0	73.0	72.0	....	....
	9	1941	3	65.0	66.0	68.5	70.5	73.0	73.5
	3	1940	.	....	....	....	....	....	....
Other	14	1940	1	72.0	72.0	74.0	74.0	74.0	77.0
Makes	15	1940	1	68.0	68.0	73.0	....	....	....
	20	1941	2	67.0	68.0	72.0	72.5	70.0	75.0
	23	1939	2	65.0	68.0	68.0	69.0	71.0	....
	4	1941	2	69.0	69.5	69.0	70.0	70.0	....
	18 H.C.*	1941	1	67.0	71.0	70.0	72.0	72.0	77.0
	11 H.C.*	1941	2	69.5	69.0	68.5	68.5	68.5	68.5
Average All Std. C.R. Cars				68.4	69.0	70.8	71.0	72.2	75.0
Average 3 Low-Price Group				68.8	69.2	70.4	70.9	72.7	74.9
Max. Spread, All Std. C.R. Cars				7.0	6.0	6.0	5.5	6.0	6.0
Max. Spread, Low-Price Group				6.0	3.5	4.0	5.5	5.5	6.0

* H.C. indicates experimental high-compression cylinder head.

by the equivalent of from 8 to 10 units in all three cars at speeds below 1000 rpm. Considered in relation to the distributor curves, 4-B would knock in Cars 7 and 11, but not in Car 24 with the timing shown. In the latter car, of course, the spark could be advanced for a gain in performance and economy.

Fuels 16-B and 20-B illustrate the same type of behavior among fuels having an ASTM octane number of about 81. In all three cars the antiknock value of Fuel 20-B at engine speeds between 2500 and 3000 rpm was lower than that of 4-B and was even comparable with that of 11-B. In a car in which the distributor advance produced knock at these speeds, Fuel 20-B would therefore be rated lower than 4-B and perhaps equal to 11-B by the CFR road-test procedure used heretofore. Such a car would be Car 24 with the initial spark advanced about 6 deg.

All of these fuels indicate the desirability of tailoring the distributor advance curve to fit the fuel characteristics as far as possible. It is obvious, however, that no one distributor curve can fit all types of fuels. Some compromise will have to be made and, in making this compromise, it will be desirable to have available information pertaining to representative current motor gasolines.

Fig. 3 showed how octane ratings for a given fuel at various speeds could be determined from the reference fuel-pattern. Following this procedure the octane ratings of Fuels 4-B and 10-B in different cars have been tabulated in Tables 5 and 6, respectively. These tables show that the distinctive characteristics of each fuel were shown consistently in all cars. The average values for all cars at each speed are tabulated at the foot of the tables and may be compared with the average values for the low-price cars which are shown also. Average octane number versus speed data for all fuels in all cars are summarized in Table 7.

Use of a Limited Number of Cars for Evaluation of Fuels – The information represented in Table 7 is of particular interest because it indicates that the average ratings obtained in the low-price cars agreed well with the average ratings in all cars and suggests the possibility of using a limited number of cars for evaluation of experimental or commercial fuels. It must be emphasized that three such cars will not necessarily show all the variations that all cars will show, although they may reliably indicate the trend.

■ Spark Advance and Compression Ratio

In order to evaluate fuels of high antiknock quality, it has often been the practice to advance the ignition timing beyond that for maximum power to produce knock. However, the validity of knock ratings obtained under such conditions has always been open to question. The high-compression cylinder heads which were available for use in these road tests made it possible to obtain a direct comparison, in some cars, between ratings obtained in engines having standard compression and over-advanced ignition timing, with ratings obtained in the same engines with higher compression ratio and correspondingly less advanced ignition timings.

The results of this comparison are shown in Fig. 9. At low engine speeds a change in compression ratio, accompanied by a corresponding change in ignition timing, had no appreciable effect upon road ratings. At high engine speeds, however, the fuel ratings were materially altered by changes in compression ratio, "sensitive" fuels being affected most.

It is concluded, therefore, that, although the performance of a fuel of high antiknock value can be evaluated by over-advancing the ignition timing, the rating obtained in this way is not necessarily indicative of the performance of such a fuel in an engine designed to utilize the fuel effectively.

Table 6 – Road Octane Number Summary – Centralized Tests

Road Octane Number of Fuel No. 10-B at Various Speeds in All Cars

	Car No.	Year	No. Groups	Engine rpm 750	1000	1500	2000	2500	3000
	6	1941	1	78.0	77.0	75.0	71.0	65.0	63.0
	7	1941	2	77.5	77.5	76.5	73.0	67.0	63.0
	16	1940	1	76.0	72.0	75.0	74.0	68.0	65.0
Low-	21	1940	1	81.0	79.0	75.0	74.0	71.0	64.0
Price	24	1941	2	78.5	78.0	75.5	70.5	66.0	....
Group	17	1940	1	80.0	79.0	76.0	70.0	....	62.0
	11	1941	2	....	....	79.0	64.0	67.5	63.5
	2	1940	2	79.5	76.5	74.0	71.5	70.5	68.0
	22	1940	1	78.0	78.0	76.0	69.0	67.0	67.0
	5	1941	2	77.0	73.0	72.5	71.0	68.0	....
	8	1940	1	70.0	74.0	71.0	68.0	65.0	....
	19	1940	2	72.5	73.5	76.0	71.5	68.0	62.5
	13	1940	1	76.0	77.0	75.0	72.0	67.0	61.0
	10	1940	1	....	75.0	72.0	69.0	66.0	61.0
	9	1941	3	76.0	76.0	74.5	70.5	66.0	62.5
	14	1940	1	77.0	78.0	76.0	71.0	68.0	65.0
	15	1940	1	72.0	74.0	75.0	70.0	67.0	64.0
	20	1941	2	80.0	78.0	76.5	71.0	65.5	60.5
Other	4	1941	2	76.5	76.5	74.5	72.5	69.5	....
Makes	23	1939	1	77.0	77.0	77.0	75.0	72.0	....
	8 H.C.*	1940	1	74.0	76.0	76.0	75.0	74.0	....
	9 H.C.*	1941	1	77.0	77.0	77.0	76.0	73.0	72.0
	11 H.C.*	1941	2	77.0	76.0	75.5	75.0	73.0	71.0
	18 H.C.*	1941	1	74.0	75.0	74.0	73.0	72.0	67.0
Average All Std. C.R. Cars				76.8	74.9	75.3	71.3	67.8	63.8
Average Low-Price Group				78.6	77.1	75.8	71.9	67.8	64.4
Max. Spread, All Std. C.R. Cars				9.0	7.0	8.0	7.0	7.0	7.5
Max. Spread, Low-Price Group				5.0	7.0	5.0	5.0	4.0	6.0

* H.C. indicates experimental high-compression cylinder head.

Table 7 – Comparison of Ratings of All Fuels in All Cars with Ratings of All Fuels in Three Low-Priced Cars of Different Makes

Expressed in terms of the ASTM Octane Number of the Equivalent Reference Fuel Blends

	Engine Speed, rpm					
Fuel No.	750	1000	1500	2000	2500	3000
4-B*	68.6	68.9	70.6	70.8	72.0	75.0
4-B**	68.8	69.2	69.5	69.7	70.0	....
10-B*	76.8	74.9	75.3	71.3	67.8	63.8
10-B**	76.7	....	76.5	72.2	67.2	63.5
11-B*	72.5	73.0	73.1	71.3	69.5	67.0
11-B**	72.8	73.7	74.5	72.0	68.3	67.2
1-B*	61.0	61.4	62.0	62.8	63.6	63.0
1-B**	60.3	61.0	61.2	62.5	62.3	62.8
2-B*	64.0	64.0	64.5	64.5	63.5	62.2
2-B**	65.2	64.5	64.5	65.0	63.3	62.7
3-B*	70.3	69.9	70.2	70.4	70.5	70.6
3-B**	70.5	69.7	69.7	70.2	69.7	70.0
5-B*	69.9	69.8	70.3	70.8	73.9	76.9
5-B**	69.0	70.3	69.7	70.5	75.0	....
6-B*	71.1	71.7	71.7	69.9	67.4	64.7
6-B**	72.3	72.7	72.8	70.2	65.7	64.2
7-B*	74.0	73.9	73.1	70.5	67.9	64.5
7-B**	75.5	76.0	74.8	68.5	67.7	65.7
8-B*	71.5	72.6	72.6	71.4	69.5	67.6
8-B**	70.8	71.8	72.5	71.5	68.0	67.2
9-B*	72.5	73.0	73.3	71.2	69.4	66.4
9-B**	71.2	72.3	73.8	72.2	68.2	66.5
12-B*	79.0	76.6	74.7	71.2	67.6	66.0
12-B**	82.5	79.5	76.3	73.2	67.2	65.7
13-B*	74.4	74.9	74.1	72.4	70.4	68.3
13-B**	76.5	77.0	75.8	72.8	69.5	68.3
14-B*	75.9	76.1	76.0	74.7	72.8	72.0
14-B**	77.5	77.3	76.3	74.7	71.3	70.5
15-B*	79.1	79.7	79.6	79.6	81.4	81.5
15-B**	79.5	80.0	80.0	79.7	81.0	81.5
16-B*	78.8	79.0	78.6	79.0	80.0	80.5
16-B**	79.0	79.3	79.0	78.8	79.5	80.5
17-B*	79.2	79.2	79.7	79.6	79.1	78.4
17-B**	79.5	79.3	80.0	80.2	79.7	78.5
18-B*	80.0	79.4	78.4	75.5	73.2	71.7
18-B**	79.5	80.3	79.5	76.3	74.7	71.2
19-B*	82.6	83.0	79.5	75.9	72.5	69.8
19-B**	83.3	83.8	80.5	76.3	72.5	69.7
20-B*	88.4	86.8	80.8	72.0	73.2	70.3
20-B**	89.0	86.7	82.5	79.5	75.0	69.8
21-B*	81.5	82.0	82.2	81.8	84.6	85.5
21-B**	82.2	81.8	83.0	82.5	85.0	86.0
22-B*	88.5	88.0	82.6	78.4	76.3	74.6
22-B**	89.0	88.5	84.2	80.5	78.0	74.5
23-B*	91.0	89.5	86.0	84.5	84.0	83.0
23-B**	No data, too high to rate					

*Average of all available data in all standard cars.
**Average of ratings in cars 6, 11, and 24.

■ Application of New Concepts

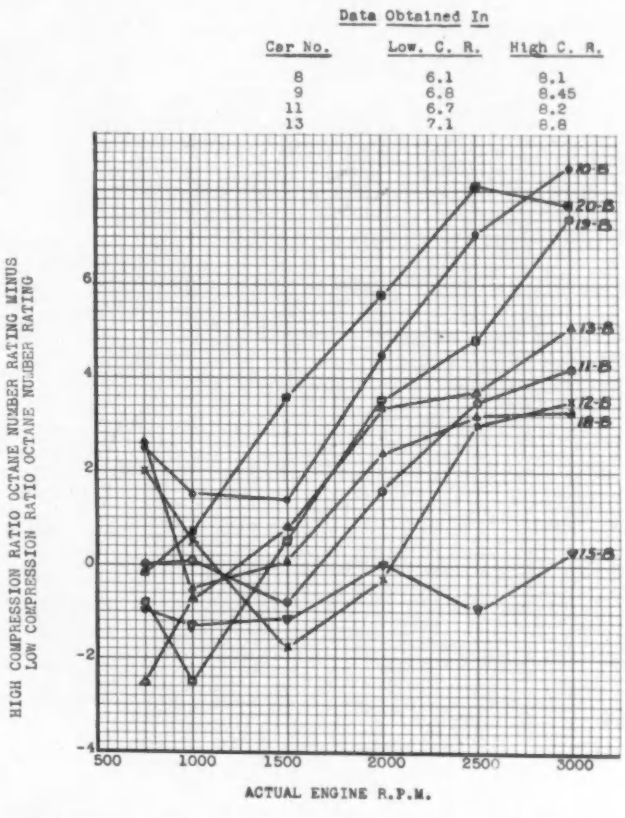
There are several ways in which the octane number-speed relationship illustrated in Fig. 3 may be applied in fuel development and in the interpretation of fuel characteristics. The octane number-speed relationship in itself is a useful means of distinguishing certain fuel characteristics, and in Fig. 5 it was shown how this relationship could be combined with the octane requirement over the speed range in a given car to indicate the knocking tendency of a given fuel in a given car.

The indications are that this principle might be extended to provide statistical information on a large number of cars representative of cars in actual service if there are available sufficient data on the octane requirements of these cars over a reasonable range of engine speeds. To illustrate how this might be accomplished, an analysis was made of the octane requirements of 19 cars used in the San Bernardino tests, having standard compression ratios.

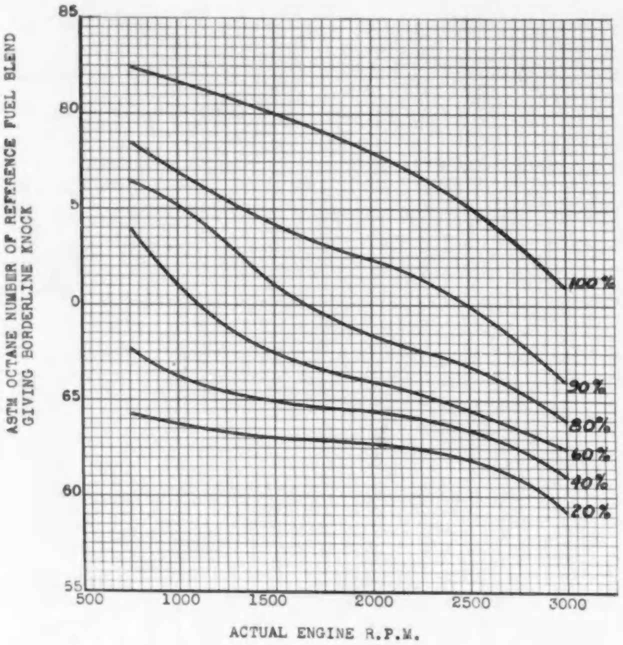
The octane requirements at each speed were determined by the procedure illustrated in Fig. 4 using distributor advance curves corresponding to the high limit specified by the manufacturer. These octane-requirement data were then combined to form distribution curves showing the fraction of the total number of cars requiring fuels of various octane numbers at each engine speed over the range of speeds between 750 and 3000 rpm. This information is shown in Fig. 10.

The relationship between fuel characteristics and car octane requirements on a statistical basis is then shown by comparing octane number-speed curves with the octane-requirement framework shown in Fig. 10. This is illustrated in Fig. 11 in which the average octane number-speed curves for three fuels, 4-B, 10-B and 11-B, taken from Table 7 have been superimposed on the octane requirement curves shown in Fig. 10.

From this figure it is possible to determine the relative performance of each fuel in a large number of cars with respect to actual car requirements and further to distinguish between the relative performance of fuels at different speeds. For example, according to this information, Fuel 10-B would be free from knock in from 80 to 90% of all the cars represented at speeds between 750 and 3000 rpm. Fuel 4-B, on the other hand, would be free from knock in only 40 to 50% of the cars at engine speeds below 1000 rpm although, at engine speeds above 2000 rpm, this fuel

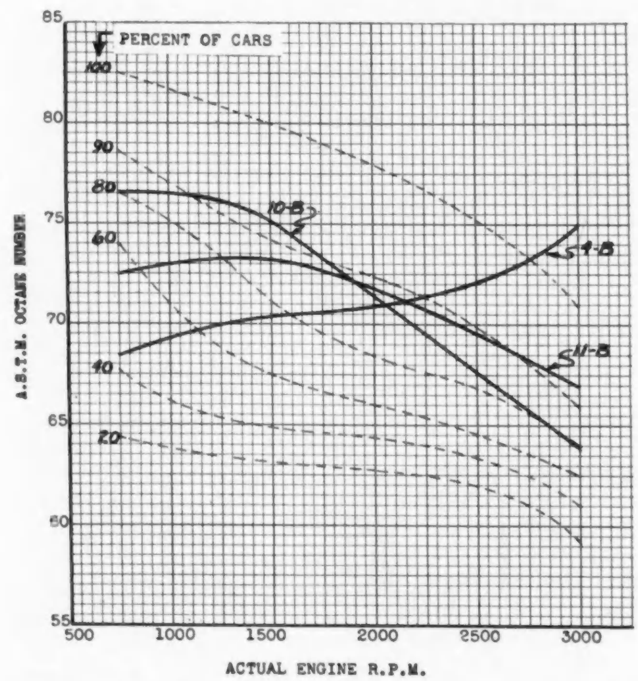


■ Fig. 9 – Effect of change in compression ratio on road antiknock value



■ Fig. 10 – Percentage of cars giving borderline knock on reference fuel blends – Data based on 19 cars having standard compression ratio, using upper limit of manufacturer's distributor curve

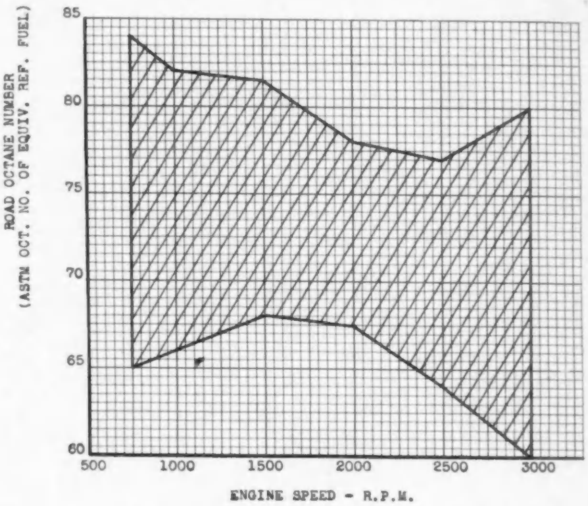
would knock in fewer cars than 10-B and would not knock in any of the cars at 3000 rpm. Fuel 11-B represents a gasoline having characteristics intermediate between 4-B and 10-B and would be free from knock in from 50 to 80% of the cars, the degree of freedom from knock being somewhat less dependent upon engine speed than that of 4-B.



■ Fig. 11 – Percentage of cars giving borderline knock on reference-fuel blends and three test fuels of varying type – Reference fuel data based on 19 cars of standard compression ratio – Data for fuels 4-B, 10-B, and 11-B are based on ratings in 15 to 21 cars

■ Application to Engine Design

The application of these new concepts to engine design depends somewhat upon the degree to which the type of reference fuel frameworks shown in Figs. 6, 7, and 8 can be reduced to representative and reproducible data which can be used as a basis for engineering development. If such information can be obtained which will be truly representative of a given engine design, it will be possible to use this information as a guide in the correlation of engine design with the characteristics of available fuels to an extent which heretofore has not been possible. Such correlation, however, is contingent upon the availability of corresponding information on the characteristics of current



■ Fig. 12 – Spread in road ratings of 11 fuels obtained in 19 cars – ASTM octane Nos. from 69.2 to 76.3 – 1939 Research octane Nos. from 73.9 to 83.2

motor gasolines. If the pattern of behavior of gasolines can be determined with reasonable certainty, then it may be possible to obtain a more satisfactory relationship between engine design and fuel characteristics.

At the present time there is a wide spread in characteristics among commercial gasolines which makes it difficult to go any further than to make a rough compromise between engine design and fuel characteristics. For example, Fig. 12 shows the spread in the octane number-speed relationship for 11 different fuels used in the San Bernardino tests having ASTM octane numbers between 69 and 76. In road tests these fuels varied over a range of from 65 to 84 octane number at 750 rpm and from 60 to 80 octane number at 3000 rpm.

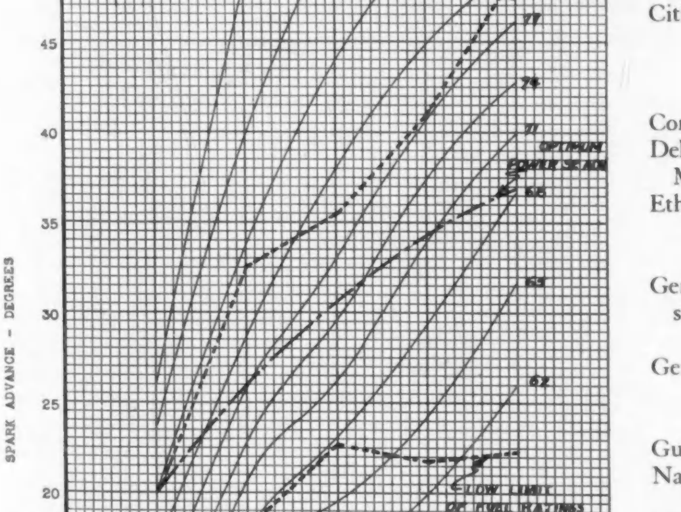
What this variation in characteristics means in terms of engine design is shown in Fig. 13 in which are represented the actual reference-fuel frameworks for one make of car and superimposed on these frameworks are the effective road octane limits for the gasolines shown in Fig. 12. It is clear from this figure that a wide range in distributor advance curves would be permissible within this range

which would be entirely satisfactory for some fuels and unsatisfactory for others. For this reason it is believed desirable to have available more definite information on the relative volumes of distribution of gasolines having various characteristics and, according to present plans, the CFR Committee will obtain such information during the current year.

Outlook

It is believed that significant advances in understanding the relationships between fuels and engines have been brought about as a result of the 1940 Cooperative Road Tests. However, the application of these relationships requires further experience with the procedures and additional data on commercial fuels and cars in normal service. To continue the work along these lines, the CFR Committee has outlined a program to be carried out in 1941. Reference-fuel frameworks are to be determined on a number of current passenger-car engines to provide engineering information on the degree of uniformity in fuel requirements irrespective of variations caused by differences in distributor settings. In order to test the usefulness of control fuels in orienting individual laboratory test cars and relating them to a significant number of cars, four fuels representing several types of gasoline are to be exchanged among the cooperating laboratories. A survey of the road antiknock performance of widely used commercial fuels in terms of road octane number ratings over the speed range is to be made in a representative group of cars. The program also includes an octane-requirement survey to provide data from which a statistical study of the octane requirements over the speed range of popular cars as found in service can be made.

The new concepts of the antiknock problem developed in 1940, supplemented with information from the 1941 cooperative program, should make it possible to adapt engines better to current fuel characteristics and modify fuel characteristics more effectively toward the requirements of current engines.



■ Fig. 13 – Actual reference-fuel frameworks for one make of car and effective road octane limits for gasolines shown in Fig. 12

Acknowledgment

The Cooperative Fuel Research Committee is extremely grateful to the many men and to the organizations that have so generously supported this project. The men who participated in these tests have given freely of their time and energy in every phase of the work and in supporting these men, the contributing organizations have supplied money, fuels, test cars, instrumentation and every facility that has been needed.

Organizations and Personnel Participating in Centralized Road Tests at San Bernardino, Calif.

Organization	Personnel
Atlantic Refining Co.	F. C. Burk
	L. J. Test
Bendix-Stromberg Carburetor Co.	V. A. Smith
Carter Carburetor Co.	L. D. Boyce
Chrysler Corp.	W. E. Drinkard
Cities Service Co., Research Division	J. D. Morgan
	P. B. Levitt
	J. Oberhollenzer
	S. G. Wilson
Continental Motors Corp.	K. E. Chantry
Delco-Remy Division, General Motors Corp.	J. T. Fitzsimmons
Ethyl Gasoline Corp.	H. J. Gibson
	Gilbert Way
	T. H. Risk
General Motors Corp., Research Division and Proving Ground	J. M. Campbell
	W. A. Snyder
General Petroleum Corp.	H. A. Mason
	J. M. Grandy
	J. D. Tobey
Gulf Research & Development Co.*	J. E. Taylor
National Bureau of Standards	C. S. Bruce
	O. C. Bridgeman
Phillips Petroleum Co.	H. M. Trimble
	K. C. Bottenberg
	F. E. Selin
Pure Oil Co.	W. B. Ross
	Kenneth Boldt
Richfield Oil Corp.	L. J. Grunder
Shell Oil Co., Inc.	R. J. Greenshields
Shell Oil Co. of Calif.	A. G. Marshall
	J. T. Ronan
	A. R. Isitt
Sinclair Refining Co.	L. E. Baker
	W. G. Ainsley
Society of Automotive Engineers	C. B. Veal

(List continued on following page)

Socony-Vacuum Oil Co.	W. M. Holaday W. S. Mount T. A. Weir
Standard Oil Co. of Calif.	J. R. MacGregor R. A. Walker
Standard Oil Co. (Ind.)	J. O. Eisinger
Standard Oil Co. of Ohio	R. I. Potter E. H. Scott
Standard Oil Development Co.	C. B. Kass V. F. Massa
Sun Oil Co.	J. G. Moxey, Jr.
The Texas Co.	W. N. Fenney E. L. Lattin L. A. Humphrey
Tidewater Associated Oil Co.	W. A. Heine G. M. Wheeler
Union Oil Co.	C. C. Moore M. S. Reynolds R. I. Stirton
Universal Oil Products	J. S. Bogen
United States Army, Ordnance Department	T. H. Nixon G. H. Schoenbaum
Waukesha Motor Co.	A. W. Pope, Jr.
Yale University	H. W. Best

* Gulf Oil Corp. contributed data to Part I, only, of these tests.

Modern Spring Materials

THE spring manufacturer is confronted with quite a list of new materials which have come upon the scene in recent years. Originally there were only the well-known carbon-steel materials such as oil-tempered wire, hard-drawn Premier and Bessemer wires, music wire, annealed spring wire SAE 1085, 0.70-0.80 and 0.90-1.00 carbon flat strip steel, and spring temper phosphor-bronze and brass in wire and strip. While a large majority of springs are still made from these types, there is an ever-increasing demand for alloy steels and corrosion-resisting materials.

Among the first alloy steels to attain general favor was chrome-vanadium SAE 6150, which by now has been developed to a very high degree of excellence, especially in the wire form for use in aircraft valve springs.

Silicon-manganese SAE 9260, used extensively in automobile leaf springs, is a good spring material, but less popular in miscellaneous spring work.

The advent of the stainless steels at first caused quite a disturbance in the spring industry. The reaction of these materials in coiling, stamping, and forming operations, as well as in heat-treatment is much different from the carbon steels and, to accomplish smooth production of springs from these materials, much new manufacturing technique has been developed. For instance, stainless-steel wires will not wind smoothly in coiling machines unless there is a lubricative coating of soft metals such as lead, zinc, or cadmium on the wire. Bare stainless wire persists in galling on the coiling points. In strain-relief heat-treatment after coiling, the reaction of the stainless-steel spring is directly opposite from the spring made of music wire, for example, with stainless steel, the spring diameter expands; with music wire it contracts.

The most commonly used stainless steel for both flat and wire springs is 18-8 Type 304 of spring temper. In some special flat-spring applications where higher tensile strength is required, Type 414, commonly called N1 hard-rolled, is employed.

When designing springs or stampings to be made from these materials, it is well to keep in mind two important factors: first, stainless steels have lower modulus values than carbon steels; second, sharp bends and small radii must be avoided on these hard-drawn and hard-rolled materials, as they do not possess the ductility of annealed steel which can be hardened after forming.

Type 420, which is primarily a cutlery grade of stainless, can be hardened and tempered, and produces fine heat-resisting springs, but it lacks good corrosion-resisting qualities after heat-treating.

Inconel, a corrosion-resisting alloy of 80% nickel, 14% chromium, 6% iron, developed by International Nickel Co., is used in special spring applications where resistance to heat while under stress, as well as high corrosion resistance, is desired. For spring purposes, it must be used hard-drawn, and is handled much the same as 18-8 stainless. With a modulus in torsion of about 11,000,000, it has in this respect, some advantage over stainless steel.

The precipitation-hardened alloys, beryllium copper, K-monel and Z-nickel, form another very interesting group of non-ferrous spring metals. The most important and useful characteristic of these materials is that they may be formed in the soft annealed state, with sharp bends and fairly deep draws, and later hardened to tempers equivalent or above those of the hard-rolled non-ferrous materials.

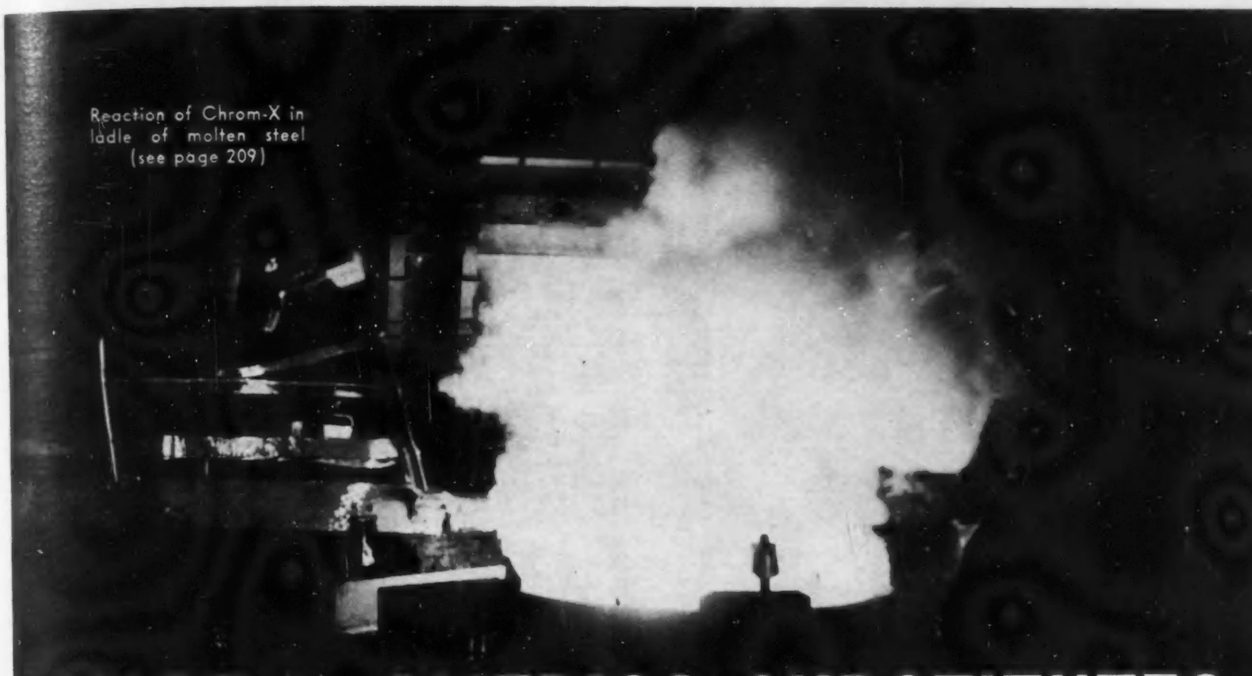
Of these, beryllium copper is the most widely used in springs, and has found considerable favor with manufacturers of electrical devices and instruments. The radio industry, for example, uses many flat springs made from beryllium copper. It has excellent electrical conductivity and, when fully heat-treated, has a higher elastic limit than hard-rolled phosphor bronze.

K-monel and Z-nickel are high-nickel-content alloys and, therefore, very useful in applications requiring springs having excellent corrosion resistance as well as moderate heat resistance.

In the heat-treatment of these new alloys, time and temperature must be controlled closely. Beryllium copper may be fully hardened by electric furnace dry atmosphere, in 60 min, taken from the furnace immediately, and left to cool in the air. The time can be reduced to 30 min by heating in liquid salt bath, with equivalent results. K-monel and Z-nickel require much longer time periods, ranging from 6 to 15 hours, depending on the hardness requirements, and must be heated in a controlled atmosphere to prevent oxidation, with the cooling regulated at a definite rate. It is obvious that proper heat-treating facilities and long furnace tie-ups are very important factors to consider when planning the use of these materials.

This problem of manufacturing variations is being attacked constantly by the springmaker, through introduction of improved machinery, and more uniform materials. The average present-day production spring is made with nearly 50% less variation, as compared with the average product of fifteen or twenty years ago. Nevertheless, it is essential when designing springs carefully to analyze the tolerance set-up. Be as liberal as possible, especially where cost is a factor.

Excerpts from the paper: "Modern Trends in Springmaking," by C. J. Bechstedt, chief inspector, Wallace Barnes Co., Division Associated Spring Corp., presented at a Southern New England Section Meeting of the Society, Hartford, Conn., Feb. 5, 1941.



Reaction of Chrom-X in ladle of molten steel (see page 209)

MADE in AMERICA SUBSTITUTES

by H. W. GILLETT
Battelle Memorial Institute

MODERN warfare, and equally so the defense preparations that we are now engaged in, depend on production of equipment and more equipment, even more than upon man-power. The materials going into war equipment and the materials for the tools that make it simply must be supplied in ample quantity and of proper quality. If these materials are not available within a country, that country must import them and, if to do this, it must keep long sea lanes open, that fact weighs heavily in naval plans. If naval forces are occupied in guarding several long sea lanes in order to secure the raw materials from friendly and neutral countries, its naval power is curtailed. This curtailment may seriously cramp the naval strategists.

To be specific, if we know and Japan knows that we can get along without tin and rubber and still make good bearings, good tires for trucks, and other good articles for which tin and rubber are used today, the effect upon naval strategy may be great.

Instead of worrying about keeping the trade route open from the East Indies, the Admirals can say "so what?" to Japanese activity around the East Indies; they can deploy their ships to guard the Panama Canal; and the diplomats need do no bluffing when they take the hard-boiled attitude that needs to be taken with aggressor nations. A complete independence of materials that have to come in over long sea lanes is the strongest hand a country can hold.

No country has all the cards in the deck. The United States has far more than any other. Our list of strategic materials – those in which we are a "have-not" country – is very short, much shorter than it was in the World War. In that War we had to keep the sea lanes open for the importation of Chilean nitrate for explosives. Germany had developed the fixation of atmospheric nitrogen to the point where she could actually use it in the War, while the other countries were only just beginning to make use of air nitrates at the end of the War. They did not lack air – they lacked the technology of its use. Similarly, we did not lack the raw materials for optical glass; what we lacked, and had to develop in a hurry, was the specific technique.

The high cards in the deck may be of several denominations. A "stock-pile" of the same material as we have been using, large enough to tide over a long period, is a high card, but not necessarily a winning one, because you never know how long a war is going to last. The very obvious precaution of stock-piling supplies of the strategic materials that we lack has long been urged by our military men but, until a few months ago, the politicians turned deaf ears to this urging. Frantic efforts are now in progress to make up for lost time in dealing ourselves these aces. The stock-piles already accumulated by provident private industry are fully as great an asset as those now being brought in by the Government. The stock-pile method has the advantage of requiring no new technology and no new equipment or, if new equipment is needed to

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 6, 1941.]

■ Fig. 1 – Tungsten high-speed steel – Normal structure at 2000X

build up the required volume of production, it is straight engineering to design and install it. You know exactly what you want. Increasing our facilities for production of aluminum and magnesium come in this category. It is straight engineering to build the new plants. A stock-pile of high-grade bauxite and of cryolite might be called for but, in a pinch, domestic lower-grade bauxite or our alunite deposits can be used and synthetic cryolite for aluminum manufacture can be made from domestic materials; because the chemical processes are thoroughly understood, we have the technology.

It often occurs that we import a high-grade variety of a given material when we have some, perhaps lots, of a lower-grade variety, because in peacetime it is cheaper to buy the high-grade material and pay the freight than it is to pay the extra cost of handling our own low-grade variety. To the logic of peacetime economics is often added the desirability of buying the material from some foreign country to which we wish to sell some of our own products. This type of barter is sensible in peacetime, and it preserves our own lower-grade materials as a potential reserve stock. But such reserve stocks are only of potential, not actual value, until there also exists the technical knowledge of how to use them and the special equipment for their processing. Seldom is the technology so developed ahead of the emergency that we know just what equipment we should use. The utilization of low-grade domestic material at best involves time and expense for construction of plants to manufacture usable finished goods from the unaccustomed raw material. At the worst, when there is no usable technologic background, it involves development from the ground up of the processes of utilization. Thus, the time lag may be very great and the pressure to produce by the first workable process that we stumble upon, no matter how inefficient, brings wasteful results compared with those that would be secured had there been a normal period of less hurried research and development. Thus, the utilization of low-grade materials is not only costly, but it is slow. Stock-piling can supply the needed time and, with a stock-pile, the possession of a low-grade supply and of technical talent to apply to the development of methods of processing and of utilization, form another fairly high card, say a king.

Both the stock-piling and the utilization of low-grade raw material are concerned with the same material that we have been using. A third expedient is the substitution of an entirely different material, but one of domestic origin, that is equally serviceable. This is a better card than either of the others. It's the joker in the deck.

The "ersatz" materials, like a German wood-pulp overcoat to take the place of wool, are not the sort of thing we mean in speaking of substitutes. A real substitute is a thing we'd just as soon have at the same cost. That we may have to pay more for it than we used to for what we formerly used, has to be charged up to the exigencies of war or defense.

THE United States is more independent of outside sources for strategic materials than any other country – and its list of "have-nots" is much shorter than it was in the last World War, Dr. Gillett brings out in this paper – an analysis of just where we stand today as regards ersatz materials. He shows clearly, however, that very definite problems face the United States as regards manganese supplies and substitutes, that tin also is a major worry, and that the situation on chromium "is not so rosy."

Detailing our position in each of the ersatz categories, Dr. Gillett points out that high cards in the ersatz deck may be of several denominations. A "stock-pile" of materials is a high card, but not

These real substitutes are not obtained by mere wishful thinking. To make them real there is required a lot of technology and user experience over a considerable time. If we want roast beef and mashed potatoes, but do not have them, we can exist pretty well on roast pork and boiled rice, but a large-scale substitution is only possible because hog-raising and rice-raising have had long periods of development. If we had only the wild razor-back hogs, if we only knew how to cook them by burning down the house, and if we had to depend on wild rice harvested into Indian canoes, our pork-and-rice diet would be merely a potential substitute, not a widely usable one. That is, the rapid development of usable domestic substitutes must be based upon the results of research *already* done, on facts ready to be utilized. Where such facts are not on record and have still to be found out, we are forced to play the stock-pile aces and the low-grade kings, but we ought to keep on dealing to see whether we can't turn up a joker.

Our "have-not" or strategic list, as prepared by our military advisers, is primarily one of metallic minerals, since modern warfare is the warfare of minerals and metals. Rubber is a notable exception, with which I need not deal since it has been discussed fully in other SAE papers. Quinine, iodine, mica, quartz crystals, and industrial diamonds are needed, but only in amounts that can be flown in, at a pinch.

Our list of "have's" is imposing. Coal, petroleum, iron, copper, aluminum, magnesium, lead, zinc, sulfur, molyb-

num, silicon, air nitrogen, potash, are sufficiently available within our borders to cause no real concern. For reasons of economics and statesmanship, it may be wise to bring part of our copper from Chile instead of expanding our own industry. Canada has nickel and platinum, and what we have is Canada's and what she has is ours for mutual defense purposes. Mexican antimony is almost in the same category, but antimony has special interest; I will come back to it shortly, since antimony is on the official list of strategic metals.

Vanadium used to be on our strategic list when it was chiefly imported from Peru. Our Rocky Mountain district is now the largest producer. Mercury is still on the list,

necessarily a winning one as no one knows how long a war will last. More powerful cards are techniques capable of bringing into use low-grade domestic supplies and/or the creation or existence of domestically available substitutes.

The metals about which the United States is chiefly concerned, according to the author, are tungsten, antimony, chromium, tin and manganese.

Stressing the need for broad research throughout the substitute field, Dr. Gillett points to the fact that cheap materials like chromium and manganese do not create a natural urge for substitution in normal peacetime – yet such substitutes may be "life-savers" in time of stress.

but domestic mercury production met the World War emergency, and seems capable of doing so again.

The metals about which there is chief concern are: tungsten, antimony, chromium, tin, and manganese.

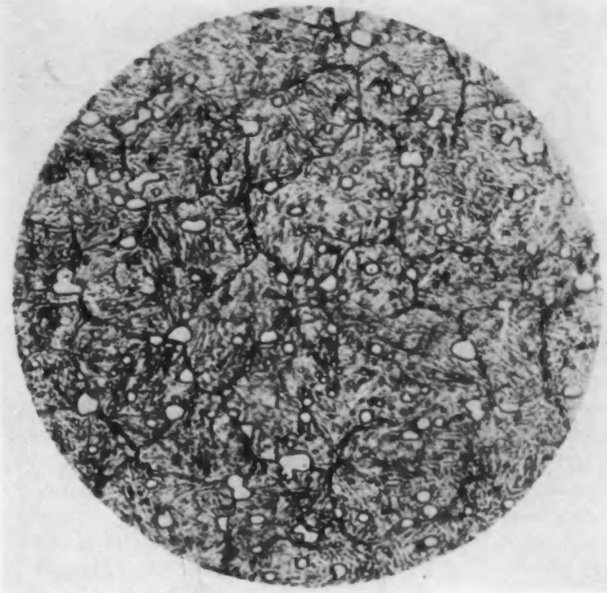
Tungsten

In the case of tungsten, we have the ace, the king, and the joker. The prospects of interruption of the supply of Chinese tungsten led to stock-piling from that and other sources. There has been active development of domestic mines due to a favoring protective tariff so that, over the last five years, domestic production has outstripped imports. There are irreplaceable uses for tungsten, as in lamp filaments and in sintered-carbide cutting tools, but these applications account for less than 2% of the total used. In all, some 20% is not readily replaceable, but this is far more than covered by domestic production.

Eighty per cent of the tungsten goes into high-speed steel. This 80% is very largely replaceable by molybdenum, of which we have a huge domestic supply.

Molybdenum high-speed steel is an outstanding case of an effective "made-in-America substitute." Its case history should serve as a model for like developments in other cases.

The War Department recognized long ago that cutting



■ Fig. 2 – Molybdenum high-speed steel – Normal structure at 2000X

tools are key materials of industry in war as well as in peace. Even at World-War prices the United States had produced only one-third to one-half the tungsten that it needed, the rest coming largely over the long haul from China. It was not then clear that domestic production could be increased materially. The War Department decided to do something about this situation. It had certain basic facts to go upon. Early metallurgical literature reported a few observations which indicated that molybdenum might be used instead of tungsten in high-speed steel. This was of academic interest, since the metal was only a chemical curiosity in those days. In 1919 Arnold and Ibbotson[1] in England reported definite findings that molybdenum high-speed steel was as good as tungsten high-speed steel. Meanwhile huge deposits of molybdenum ore had been found in Colorado and New Mexico.

This combination of facts spurred the War Department to action, and Ritchie[2] demonstrated that molybdenum high-speed tool steel could be made by commercial methods and that, in cutting tests at Watertown Arsenal, it performed well enough to be classed as an acceptable substitute for tungsten high-speed steel. This finding, valuable as it was, would by itself have had little effect, since the tool-steel makers as a whole, did not then accept it. One user of tool steel whose interest was in making tools, not in making steel, did give credence to the Army's evidence. Emmons[2] of Cleveland Twist Drill independently came to the conclusion that Mo high-speed steel would be a useful and economical steel in normal, peacetime conditions. He then spent years and lots of money in working out the "bugs" and commercializing the molybdenum steel. It happens that molybdenum has a lower atomic weight than tungsten, so it does not take so much of it to give the proper proportion of the hard carbides that give high-speed steel its cutting power. To the metallurgist, who knows that molybdenum carbide and tungsten carbide are much alike, both being hard and both being stable at high temperatures, it is logical that, when the structure of a tungsten high-speed steel is duplicated in a molybdenum high-speed steel, the performance ought

[1] See the *Journal of the Irons and Steel Institute*, Vol. XCIX, 1919, pp. 407-428: "The Molecular Constitutions of High-Speed Tool Steels and Their Correlations with Lathe Efficiencies," by J. O. Arnold and F. Ibbotson.

[2] See *Army Ordnance*, Vol. 11, 1930, pp. 12-19: "Molybdenum in High-Speed Steel," by S. B. Ritchie; see also Transactions of the American Society for Steel Treatment, March, 1933, pp. 193-232: "Some Molybdenum High-Speed Steels," by J. V. Emmons.

to be similar. Figs. 1 and 2 show that the structure is duplicated.

When the composition is properly adjusted and the tool is properly heat-treated, the performance is closely the same. The machinist who uses a high-speed tool today is unlikely to know or to be able to detect any difference between the molybdenum-steel tool he draws from the stock room today and the tungsten-steel tool he drew yesterday. That is to say, the machining of defense materials by high-speed tools can be done as effectively by the "made-in-America" molybdenum steel as by the tungsten steel.

The molybdenum high-speed steel has not reached this position of acceptance through patriotism of tool-steel users in favoring a material of domestic origin; it has obtained its standing by being good and at the same time being cheaper. That is, peacetime economics worked out to bring the molybdenum steel into such wide use and give it such background of experience that, in a pinch, there will be no hardship and little opposition to a complete change-over should war-time or defense economics demand. Nevertheless, the path of molybdenum high-speed steel has not been strewn with thornless roses. There have been "bugs" to eliminate in the production of the material. Alike as the two elements are, the steels made with them differ enough in their handling properties to require slight modifications in technique. The molybdenum steel is prone to acquire a soft skin in forging or in heat-treating. This soft skin has been variously ascribed to volatilization of molybdenum from the surface, to decarburization, or to increased surface oxidation. Whatever be the mechanism of the trouble, it is avoided only by suitable protection of the surface in forging and by heat-treating in the proper atmosphere. Special furnaces provided with gas generators to produce this atmosphere have been recently made available. This is an example of the subsidiary studies and engineering technology that are necessitated by any shift in materials.

Recently a high-speed steel with less than half the tungsten of standard high-speed steel and about half the molybdenum of the straight molybdenum steel has been found to perform like either steel, and alleged to be as free from soft skin as the tungsten steel, so as to be heat-treated without using special furnaces.[3] As a matter of fact, you could re-melt scrap tungsten and scrap molybdenum high-speed steels in practically any proportion and come out with a perfectly usable steel, provided that the heat-treatment temperatures and times and atmospheres are adjusted to the composition. Metallurgical control has so completely superseded the blacksmith in the tool room that there is no real barrier to the exclusive use of high-speed tools in which part or all of the tungsten has been replaced by American molybdenum.

But it took ten years to bring this situation about, in peacetime. Much prejudice had to be overcome and much sales effort exerted, on top of the necessary technical developments and, even now with the virtues of the molybdenum steel well proved, it has only replaced perhaps 20% of the production of the old-time tungsten steel.

Had there been a defense urge during this period of struggle, recognition of the virtues of the Mo steel could have been brought about more rapidly, but it is a slow job for the potential virtues and limitations of a substitute to be brought out in the absence of some such urge.

These shifts to substitute materials do not come about overnight. They require much research and technology, the building of great new plants, and the training of many workers. Under peacetime economics the changes occur in a very slow but rather orderly manner. Every new technology has "bugs" in it that have to be chased down and eliminated – bugs that are far better eliminated in small pilot plants than to be allowed to stay hidden until a huge plant has been built and a lot of expensive machinery installed that you finally find out isn't what you wanted at all.

Thus there can only be certainty about the utility of a substitute when there is a sufficient background of research and of pilot-plant experience to insure that large-scale production and utilization is purely engineering and not guess-work.

■ Antimony

Sometimes, though rarely, prior research affords a good deal of the necessary information. I think we have one such case, antimony, among strategic materials. The United States itself contains but little antimony ore, whereas we require large quantities, much of which was, in the World War, imported from China. A domestic smelter in Laredo, Tex., has been operating for nearly 10 years and our requirements are now chiefly supplied by Mexican ore, with a good deal of the balance from Bolivia, thus greatly shortening the trade routes. Another smelter is going in at Kellogg, Idaho, to handle a few domestic deposits in the west.

Antimony is carried on our official list of strategic metals because we need storage batteries for the cars, trucks, and so on, of mechanized warfare. But few commentators seem to realize that the 10% of antimony in the average storage-battery grid can be satisfactorily substituted by 0.1% calcium. Findings to that effect were published in 1925 both by the Bureau of Standards[4] and the Bell Telephone Laboratories[5]. There may be a few minor "bugs" but, by and large, very satisfactory batteries can be made with the use of 1/100 the tonnage of calcium compared to that of the antimony required. Calcium used to be imported, but is now made here, we have the "know how" and plenty of limestone. Since we have lead in abundance, we need not give up storage batteries even if we were shy of antimony.

Moreover, we discard every year millions of storage batteries, whose plates are normally re-melted to go into the new ones that replace them. This antimonial lead thus turned over every year contains more than enough antimony to supply a year's demands for other than battery uses. Our old batteries thus form an automatic antimony stock pile upon which we could draw.

The technique of separation of the antimony from the lead is well known; the simple and effective Harris process is in considerable use, and increasing the facilities for reclaiming the antimony by that process is straight engineering. In this case the necessary technical knowledge adds up to a very satisfactory whole; we have a usable made-in-America substitute for the major use, a stock pile, and short hauls on importations.

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[3] See Transactions of the American Society of Metals, Vol. 27, 1939, pp. 289-336: "Development in Molybdenum High-Speed Cutting Steels," by W. R. Breeler.

[4] See Transactions of the Electrochemical Society, Vol. 68, 1935, pp. 293-307: "The Electrochemical Behavior of Pb, PbSb, and PbCa Alloys in Storage Cells," by H. E. Haring and U. B. Thomas.

[5] See Transactions of the Electrochemical Society, Vol. 68, 1935, pp. 309-316: "Some Physical and Metallurgical Properties of Lead-Calcium Alloys for Storage Cell Grids and Plates," by E. E. Schumacher and G. S. Phipps.

Chromium

The chromium situation is not so rosy. The demand for chromium has increased vastly in the last few years. For our most important needs, corrosion-resistant stainless steels, heat-resistant and electrical resistor alloys, all high in Cr, there is no obvious substitute. The petroleum industry uses Cr in cracking tubes. It is required in armor plate, in special wear-resistant alloys, and to the extent of 4% in high-speed steel whether that be the W or the Mo variety. It is used in CrMo steel tubing for airplane skeletons.

Many commercial uses, in SAE steels and in alloy cast iron, can be largely substituted by other alloy steels and irons, but it is difficult to whittle away any very large proportion of the other important uses by the use of substitutes. Our supplies of chromium are imported over long trade routes, mostly from South Africa, New Caledonia, and Turkey. We have some pockety high-grade deposits and, in Montana and California, some large low-grade deposits with such low Cr and high Fe that the ordinary metallurgical treatment does not produce a ferrochromium acceptable to steel makers.

The technique of a step-wise reduction, in which a high-Fe low-Cr alloy is first made, then the balance of the Fe and Cr reduced to make a standard alloy, has been described[6]. Such a method was for a time in commercial operation by the Chromium Mining & Smelting Corp. of Canada. A later development of this company allows use of alloys directly made from low-grade ores. This is the so-called "Chrom-X" briquet for ladle addition, consisting of high-iron ferrochromium, with ferro-chromium-silicon plus calcium chromate and sodium nitrate. The reaction of silicon with the oxidizing reagents gives off enough heat to melt even the high iron ferro, so that one of the steel makers' objections to non-standard ferrochromium such as is obtained direct from low-grade ores is removed. Use of Chrom-X is illustrated on the first page of this paper. We have been working at Battelle with the Canadian company in the elimination of some of the early "bugs" in the process. The Chrom-X production has been entirely from low-grade ores corresponding to concentrates obtainable from the Montana ore. The Rustless Iron & Steel Co.'s process can also utilize low-grade ores. Such techniques, rather than widespread substitution of chromium by something else, appear to be our best bets in the long run. For the immediate emergency, worry is removed by the stock piles, especially those accumulated by provident private firms even before the Government began to stock imported chromium ore.

The tungsten, antimony, and chromium situations thus appear to be pretty well in hand. Now we come to tin and manganese – our chief worries.

Tin

We have no domestic tin ore at all, and what we use is now imported over long sea lanes. The present large stock piling of tin most certainly should be our ace, and the provision of a domestic smelter to work up the low-grade, but relatively short-haul Bolivian ores is a good king. To go with these we have a wild deuce in the form of substitute materials and technology though, in most discussions of the tin situation, this is classed as just a plain deuce, not a wild one.

[6] See *Engineering and Mining Journal*, Vol. 141, July, 1940, pp. 49-51: "Smelting Domestic Chrome Ores in the Electric Furnace," by R. G. Knickerbocker, M. B. Royer, and T. E. Evans.

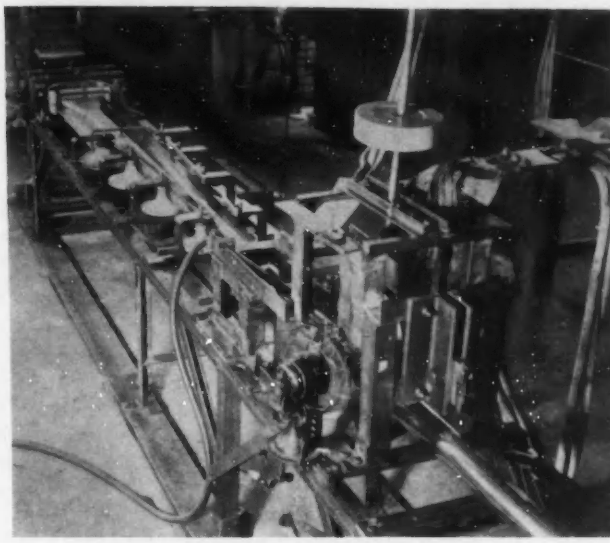

Fig. 3 – Experimental welding of black steel plate into tubes for can bodies at Battelle Memorial Institute

The things we use tin for are essential. Our use of tin divides about as follows, out of a 90,000-ton consumption in 1937:

Tin plate	39,000 tons		
Solder	20,000 tons	Radiator	9000
		Miscellaneous, including wiping	6500
		Can soldering	4000
		Auto body solder	500
Babbitt	6,500 tons		
Bronze	6,500 tons		
Miscellaneous tubes, etc.	18,000 tons		
	90,000 tons = 90 million dollars at today's price.		

Most of the tin plate goes into tin cans for food products. The suggestion has been made to replace the tin can by silver-plated cans, but this has not met enthusiastic response. There is a place for silver-lined containers that are used over and over, but there are more acceptable substitutes for the throw-away type. Aluminum coating has been studied, and a very little is in use for the few services for which it is suited, but that, too, is no general answer. The work along these lines of substitution has not been particularly fruitful. Obviously food can be packed in glass in a pinch, though the weight and fragility of the container are drawbacks and the canners' equipment is none too readily modified to deal with glass.

The most practical method appears to be the use of black, untinned plate, with lacquer linings such as are used in beer cans. Such a package would be feasible in perhaps 95% of the food cans. The tin is not there to protect the can against corrosion by the contents to anywhere near the extent that it is there merely to facilitate soldering the joint of the can body. Now it is obvious that, for this purpose, there is no real reason for coating the whole surface, a mere stripe at the joint would be enough.

With tin available, it is a nuisance to keep all but this stripe uncoated, but it could be done, and apparently is being done abroad. Some domestic firms are considering such a development.

A more logical method is to make the joint by welding, thus avoiding all need for tin or solder. It is no easy job to weld such thin material as can stock, but as Fig. 3 (experimental welding at Battelle of black plate into tubes for can bodies) shows, it can be done. The feasibility has been clearly demonstrated. The chief trouble is a normal one of any substitution, that is, that welding equipment must be installed in place of the soldering equipment in the can-making line. Such substitution would take time.

However, there is enough technologic experience to make one confident that the canning industry would not shut down if its tin supply were almost entirely shut off – if a stock pile gives it time enough to make the changes.

■ Solder Large Tin Consumer

Solder accounts for the next largest amount of tin, and of this, the most goes into auto radiator solder. The aircraft industry is already using a 95-Pb, 5-Ag solder on radiators for glycol-cooled motors. A 97½-Pb, 2½-Ag solder is usable for nearly every ordinary solder use and at present prices, volume-for-volume, is a trifle cheaper than a 45-Sn, 55-Pb solder.

Other usable solders are 82½-Cd, 17½-Zn; 95-Cd, 5-Ag; and 85-Pb, 15-Cd plus a little Zn.

Any such solder would serve for such uses as the solder drops in the base of an incandescent lamp placed there to insure electrical contact; this use accounts for 100 tons of tin annually.

The auto industry uses 500 tons of tin a year in body solder, used merely to smooth up the bodies before lacquering. A maker of one of the three "popular-price" cars uses no body solder. This use therefore can be avoided or a substitute used.

Wiping solder of 62-Pb, 38-Sn for cable and plumbing joints, off hand, would seem more difficult to substitute, but high authority[7] tells us that burning technique is entirely adequate for these joints so that, if need be, tin could be entirely dispensed with in these lead-to-lead joints.

There are other joining techniques for steel-to-steel joints that can be called upon in cases where they are applicable, especially the well-worked-out copper-brazing method. We seem recently to have hit upon still another technique at Battelle that offers possibilities of joining a variety of metals without using either tin or cadmium, that also may have application in specific cases, but we have still to work out some "bugs."

Tin bronzes, used for strength plus corrosion resistance. ordinarily can be replaced by silicon and other non-tin bronzes. It is not so easy to find a substitute for 80-Cu 10-Sn 10-Pb bearing bronze, but one with 5-Sn 5-Ni instead of 10 Sn is probably usable, and the addition of a little more lead to 85-5-5-5, made from old scrap, would give reasonably good performance in a good many cases where 80-10-10 is specified.

Tin-base babbitts have certain properties that are hard to duplicate in other alloys, but the advent of very thinly lined steel-backed bearings, with Cu-Pb or Cu-Ni between the steel back and the actual lining, permits satisfactory use of lead-base babbitt, so that the tin-base babbitt has almost passed out of the automotive picture. For more thickly lined babbitt bearings, nothing is available as yet that does not have some drawbacks. If we are not mistaken, we have in process of development at Battelle a lead-base babbitt that more nearly approaches the tin-base material than the alloys heretofore suggested.

Cadmium-base babbitts are superior to tin-base alloys in all properties save that of corrosion resistance. The alkali-hardened lead babbitts are strong but their corrosion resistance is very poor and they are mean to handle. The Cu-Pb bearings may often be used in place of babbitt, but it is so expensive to make a good Cu-Pb bearing that its use is confined to cases where babbitt won't serve.

Silver-lead makes a good bearing and some experiments at Battelle indicate that a gold plate may make quite a respectable one.

The substitution of Cd for Sn in solder and babbitt is limited by the supply of Cd and, since Cd is obtained only as a byproduct of zinc smelting, not from cadmium ores, the supply cannot be increased at will. So Cd can be relied upon only for the replacement of a small amount of tin, especially since most of this substitution has already taken place, that is, most of the Cd is already used in bearings.

Silver for solder or for bearings is at hand if needed, since the Administration's silver-buying policy, although primarily based on political motives, has given us a stock pile. We have 100,000 avoirdupois tons of silver – enough for 90 years of ordinary industrial uses of silver, or enough to replace the same thickness of all tin-base babbitt linings in bearings for 10 years at the normal rate of use of tin-base babbitt. We have well over 11,000 avoirdupois tons of gold above the ordinary requirement for backing the monetary system, and the signs are that we will continue to accumulate further stocks. This would make a lot of gold-plated bearings. We use about 1,000 tons of cadmium a year; yet most discussions of substitutes for tin in babbitt and solder feature cadmium, really a less available metal than silver or gold.

On the whole, if it were necessary, the United States could get along by making changes in technique that could be accomplished in a year or so if necessity arose, with no more than a tenth of our present importation of tin. Bolivian tin, smelted in the United States, will give us this amount and more, and a stock pile is being accumulated to give the necessary time for a change to substitutes.

The substitutes would be but little more expensive to use than tin at its present price and, once in use, might to a large degree permanently compete with tin at its normal price.

The feasibility of substitution for tin may well have a strategic effect upon our relations with Japan, since interruption of our importation of East Indian tin would merely be a temporary irritation. If the development of substitutes and the technology of their use are actively carried on to completion, the defense effort would not be noticeably hampered by a stoppage of the inflow of tin.

So our list of imported metals, of which a reasonable supply cannot be had in America, and for which no "made-in-America" substitute is in sight, dwindles down to the single case of manganese.

■ Manganese

We are using all our present steel-making capacity, and more is to be added. Each short ton of steel requires about 11 lb of manganese, so our planned 1941 output of more than 80 million tons of steel requires about 900 million lb of metallic manganese. Allowing for smelting losses, this means that we need about a million tons of high-grade manganese ore, or its equivalent, from which to make

[7] See *Metals and Alloys*, Vol. 13, p. 174, February, 1941, Letter to the Editor, by F. W. Willard.

roughly half a million tons of standard ferromanganese or its equivalent. The normal price of 80% ferromanganese over the last ten years was about $80 per ton. The present price is $120. In 1917 the average price was $325. Given time to construct the processing and smelting equipment, our low-grade ores could be turned into usable manganese by known methods for say about $60 per ton over present prices. There are deposits of high-grade ore in Montana, worked by Anaconda during the World War and just beginning to be worked again, from which we can secure some 100,000 tons a year. The low-grade ore in Cuba is amenable to concentration to a high-grade concentrate, that is, to metallurgical-grade material. From this we will get another 150,000 tons a year. This gives one-fourth of our requirements. Other smaller pockets of domestic high-grade ore, plus an increased use of spiegel and silicomanganese from low-grade ores, will supply about one-third of our needs through standard smelting processes requiring no new technology. By accepting the time factor involved in building a railroad in Brazil, some good ore could be made available, but ocean shipment is still required.

Some two-thirds of our needs must still be imported over sea routes difficult to protect, or be produced from low-grade ores by known processes that are decidedly expensive, or by suggested processes that are doubtful either as to technology or economics. The stock pile being accumulated is calculated to last two years, which period is expected to give time to select an economical method and get it into commercial production.

The situation cries aloud for a broad program of research on economical utilization of low-grade ores, not carried on, as much of the past research has been, with the idea of proving that someone's pet preconceived idea will be the answer, but with a completely open mind. We know that step-wise reduction can be applied to remove the high phosphorus of these ores and to make usable spiegel and ferro from them. We know that leaching processes can be applied to separate out a pure manganese oxide, which can be nodulized and mixed with low-grade ore for smelting to standard ferro. An embarrassing number of other more or less logical schemes have been proposed, but the technology has not yet been developed far enough to tell whether there is any more practical and economical scheme among these. Because of the large tonnage involved, it would be costly and wasteful to build operating plants without plenty of pilot-plant tests to pick out a relatively cheap and trouble-free method.

Oddly enough, the matter of substitutes has received only the scantiest of attention. We can whittle away at some of the needs for Mn in specific steels where it is used as a strength-giving alloying element. The SAE X1000 series ranging from 0.70 to 1.20% Mn, the T1300 series with around 1.75% Mn and the analogous "medium manganese" cast steels could be replaced by other steels strengthened say with Ni and Mo, but at an increase in cost. The Mn is used in these steels because it is, in ordinary times, the cheapest of the alloying elements.

In the high sulfur screw stock X1300 steels, Mn is carried at 1 to 1.65% in order to make it feasible to produce the steels with the high sulfur content, that is, we are really utilizing manganese-sulfide particles. Equivalent machinability can be had with less S and Mn through the use of lead[8].

The leaded-steel development arose from work at Battelle for Inland Steel, which stemmed from the desire for more-machinable steel rather than for saving of Mn. This could help a bit, but the tendency is to keep the S and Mn at normal levels in screw stock and add the lead to get its increment of machinability on top of the effect of Mn S.

Whittling away at the replaceable alloying effect of Mn can reduce but a small proportion of the total needs for Mn, since its tonnage use is not so much as a true alloying element, but rather as a "conditioner" of all steel, whether carbon or alloy. Concerning this use you can find oft-quoted statements in the literature that one could turn to zirconium or titanium for the production of steel in a shortage of manganese.

Functions of Manganese

One of the three important functions of manganese in steel is to combine with the sulfur, giving manganese sulfide which is plastic at forging or hot rolling temperatures and does not make the ingot crack up, while without manganese the sulfur is there as iron sulfide which is not plastic, surrounds the steel crystals, and does make the ingot crack. The subsidiary function of strengthening the steel could be handled by nickel, molybdenum or other available elements. The other action of Mn, one of partial deoxidation, is of no huge importance in fully deoxidized steels, since silicon and other deoxidizers are added anyway to complete the action, and the amount of manganese carried in by the scrap and pig iron of the steel-making charge is sufficient, with proper manipulation of the melting process, to flux the silica deoxidation product to the necessary degree. But the combining with sulfur to form a harmless sulfide is a function not performed by many other elements.

Zirconium and titanium do form sulfides in steel and there is some experimental evidence that these are of the desired harmless type that permit hot working without cracking, so the metallurgist is quite likely to accept the statement that Zr and Ti would take the place of Mn, with the reservation, however, that since both are strong deoxidizers, the substitution could only be made in fully killed steels, not in rimmed or semi-killed steels. True, killed steels could be used where rimmed steels are used today, but the recovery from a cropped steel ingot is materially less than from a rimmed ingot, so shifting to killed steel would amount to some reduction in our effective steel-making capacity.

This limitation in applicability, the present price of Zr and Ti at 40¢ per lb compared with domestic manganese from low-grade ores at less than 15¢ per lb, the low recovery in the steel from additions of these metals, and the fact that both Zr and Ti ores are imported, have held the studies of these substitutes within rather academic limits. Some of the published data[9] on the behavior of Zr were obtained in the absence of Mn, while commercial steels carry residual Mn from scrap. Other tests[10] were chiefly with sulfur contents so high as to be of interest only in respect to screw stock, although no machinability data were supplied, and with ingots so small that the freezing rate, which is important in this connection, was far from representative of big ingots for tonnage steel. Published information on Zr is quite insufficient to tell a steel maker just how it might be used practically in a pinch.

[8] See Transactions of the American Society for Metals, Vol. 27, 1939, pp. 887-922: "A Discussion of Leaded Steels," by F. J. Robbins and G. R. Caskey.

[9] See Transactions of the American Society for Steel Treating, Vol. 20, 1932, pp. 193-232: "Steels Made without Manganese," by S. B. Ritchie.

[10] See Transactions of the American Institute of Mining Engineers, Vol. 69, 1923, pp. 848-894: "Some Effects of Zirconium in Steel"; Vol. 70, 1924, pp. 201-223: "Effect of Zirconium on Hot-Rolling Properties of High-Sulfur Steels," both by A. L. Feild.

The published information on Ti[11] is even more sketchy, for it rests on two heats of 3½ lb each, one of them in a vacuum, the other in air, and the latter gave inconclusive results. These data have been cited by others in much more sweeping generalizations than the research workers who obtained them would approve. The appraisal of Ti as a possible Mn substitute rests chiefly on its several similarities to Zr.

Scanty and inconclusive as the evidence is, it should not be disregarded entirely. Elimination of Zr from consideration is probably justified, since it would have to be imported. Titanium cannot be so lightly dismissed, because we do have domestic deposits of Ti ores, and there are some in Quebec. Tyler[12] comments on the cheapness and abundance of foreign ores of Ti, the Ti ores being delivered here at a lower price per unit of Ti, in peacetime, than that of Mn ores per unit of Mn. A demand for a really large tonnage of ferro-titanium ought to lower the smelting cost. Ti can be introduced into steel by aluminothermic[13] reduction of the oxide or rutile, although the Al required costs about as much as to introduce it as ferro-titanium.

If only from the point of view that, after order is restored in the world, we might find it more desirable to purchase Ti or Zr ore from some nation with which we wish to increase our trade, than Mn ore from one with which we do not, the study of Ti and Zr as Mn substitutes might well be carried on to a real appraisal of their possibilities and limitations, irrespective of their present cost.

There is such a thing as being too cramped in the style of our experiments in the light of the economics of the present moment, and of the blocked-out ore supplies of the present moment. It is better to have advance technical information in such shape that we can take advantage of changes as fast as they occur, if they do occur, even though we hardly expect them to occur. The air nitrogen prospect at the start of the World War looked just as uneconomic in comparison with Chilean nitrate as titanium does now in comparison with manganese. An expensive material like tin creates a natural urge for substitution even in normal times. Cheap ones like chromium and manganese do not – until an emergency makes us think about what we would do in their absence. We cannot rely on thinking solely in terms of peacetime economics.

We hear plenty about the efforts of the Administration and of industry to utilize domestic low-grade ores. Every encouragement is given along this line. But we hear nothing of any corresponding effort to develop substitutes. A substitute has to bid fair to show a marked saving under normal economic conditions if its development is to roll along under its own momentum, that is, if it is to get financial backing to bring it into sufficient production and use so its commercial utility can be demonstrated. Those substitutes that are likely to be economic only in time of stress and appear to offer no noteworthy saving in normal times are red-headed step children. They may be life-savers in time of stress and, when we know more about them, they may turn to be economic in normal times.

This apathy which leaves potential future substitutes to make their own way without encouragement is not a forward-looking attitude. Research to find them should be encouraged and something analogous to "educational orders," by which promising ones might get the benefit of actual use even though for the present they cost a bit more, might well be instituted. Everyone takes it as a matter of course that a metal produced from low-grade ore will cost more, and there is tariff protection on some ores, tungsten, for example, to foster domestic production, but everyone seems to balk at the idea of a more costly substitute. This doesn't make sense. When we utilize low-grade ores, we deplete our reserve supply. A shift to a different domestic material as a substitute opens up a new reserve. Some of the fervor with which the low-grade ore situation is being attacked might well be shown on the matter of substitutes.

As time goes on and everyone gives more thought to strategic materials, individual research workers will have ideas for substitutes and individual firms, worrying about being cramped for supplies of their usual materials, will experiment with the use of substitutes as a matter of insurance. We shall probably see the real advances in this line being made by those who are not directly engaged in defense production and fear that they may be cut off from normal raw materials on the score of their product not being essential.

Pehrson of the Bureau of Mines says:[14] "the Army and Navy Munitions Board wisely has established the policy that the use of substitutes in any emergency should be confined to products whose qualities have been demonstrated under peace-time commercial conditions; to impose large-scale substitution on industry at a time when the industrial machine is expected to deliver its maximum effort merely invites disaster."

This sentiment was phrased only a bit differently in 1929 by Taylor[15] who said "even where the Army with the aid of scientific laboratories has been able to determine that a substitution can be made, its introduction during an emergency may cause loss of valuable time."

Thus it is *past* research, seasoned by time, and at least introduced into commercial development, upon which the immediate possibilities of substitution rest.

Summing up, "made-in-America" substitutes are extremely powerful factors in defense. A real substitute is as good as a battleship. All the present substitutes are the fruit of research and technology which, in nearly every case, has been aimed at economy under peacetime conditions, rather than planned for defense purposes. One notable exception, molybdenum high-speed steel, was planned for defense purposes. Its development to the point where it can be painlessly substituted is, however, due to its being cheaper under every-day conditions, so that it went ahead of its own momentum. Its success suggests that we ought to do more such planning, and that we should not rest content until the strategic materials that we must still import are all wiped off the list.

Partly by plan, partly by luck, but mostly by the bounties of nature, our list of "haves" is larger, that of "have-nots" smaller, than in the case of any other nation. Hasten the day when world peace again allows all nations to use the material that is most economic, no matter where nature put it but, until that day again arrives, we must be able to provide all the defense materials needed to guard against aggression. In this task, the research and technology that give us adequate "made-in-America substitutes" to strengthen every weak link in the chain of strategic materials are as vital as are guns, ships, tanks and planes.

[11] See Transactions for the American Society for Metals, Vol. 23, 1935, pp. 654-671: "Non-Metallic Inclusions in Steel – Part II, Sulfides," by S. F. Urban and J. Chipman.

[12] See *Metals and Alloys*, Vol. 6, May, 1935, pp. 131-133: "Titanium – Its Supply and Economics," by P. M. Tyler.

[13] See Transactions of the American Society for Metals, Vol. 25, September, 1937, pp. 788-825: "Effect of Titanium on Some Cast Ferrous and Non-Ferrous Metals," by J. A. Duma.

[14] See *Metals* (monthly supplement to *Daily Metal Reporter*), November, 1940, pp. 7-12, 20: "U. S. Buying Strategic Metals," by E. W. Pehrson.

[15] See *Metals and Alloys*, Vol. I, July, 1929, pp. 5-7: "Strategic Raw Materials," by R. Taylor.

Report of CFR Committee on Aviation VAPOR-LOCK Investigation

by O. C. BRIDGEMAN

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PROGRESS being made by this investigation is indicated by the fact that 33 progress reports have been submitted during the past year. This paper summarizes these reports. Four methods of vapor-lock control are treated: 1. By controlling fuel characteristics. 2. By fuel-tank supercharging. 3. By fuel cooling. 4. By compromise methods.

Dr. Bridgeman considers: Method 1 least promising since most considerations point to fuel of higher vapor pressure, rather than otherwise. Method 2, he believes, merits serious consideration and further study, and Method 3 has many advantages. Method 4, he explains, consists of a local and temporary expedient in the case of any given aircraft design but it does not present a fundamental solution for the problem. Further studies are being continued and progress is being made, he reports.

CONSIDERABLE progress has been made during the past year in the investigation of the various phases of the problem of vapor lock in airplane fuel systems, and 33 progress reports have been submitted during this period. Of these progress reports, 28 have been concerned with pressure drops in component parts of airplane fuel systems, 4 have been concerned with aviation fuel characteristics related to vapor lock, and 1 was a summary report covering the general problem as it was pictured approximately a year ago. It is the purpose of the present report to summarize the information contained in these various progress reports.

Vapor lock may be defined as the partial or complete interruption of fuel flow due to the formation of vapor in the fuel-feed system. Fundamentally, the vapor-lock problem involves the three variables:

(1) Vapor pressure of the fuel.
(2) External pressure on the fuel.
(3) Fuel temperature.

Control of any one of these variables, or any combination of them, can be made to ensure entire freedom from vapor lock at any altitude.

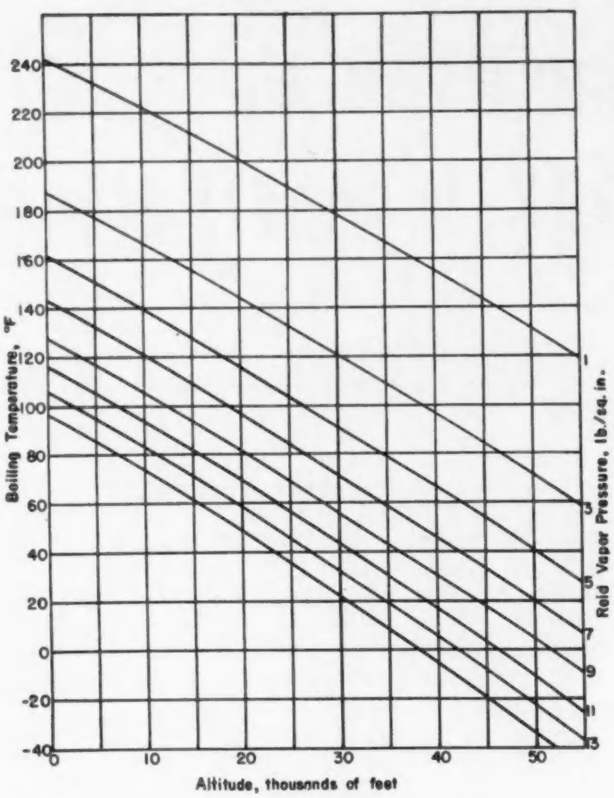
Control by Fuel Characteristics

Considering first the question of fuel vapor pressure, Fig. 1 shows the temperatures at which boiling in the tank will start at various altitudes (based on the standard atmosphere) with typical fuels of specified Reid vapor pressure. If an airplane leaves the ground with a fuel temperature of 100 F and if the temperature of the fuel does not change during the climb, it is seen that boiling in the tank would start at an altitude slightly under 20,000 ft with a fuel of 7 lb/in.^{2} vapor pressure. If now a fuel of 3 lb/in.^{2} vapor pressure were used, an altitude of about 40,000 ft could be reached before boiling started. While vapor lock has been encountered in some cases at altitudes below those at which boiling of the fuel started in the tank, in general, vapor lock does not occur until somewhat higher altitudes are reached than that at which boiling is predicted, the difference being dependent upon the design and vapor-handling capacity of the fuel system and upon the cooling of the fuel in the tank during the climb, which is usually only a few degrees.

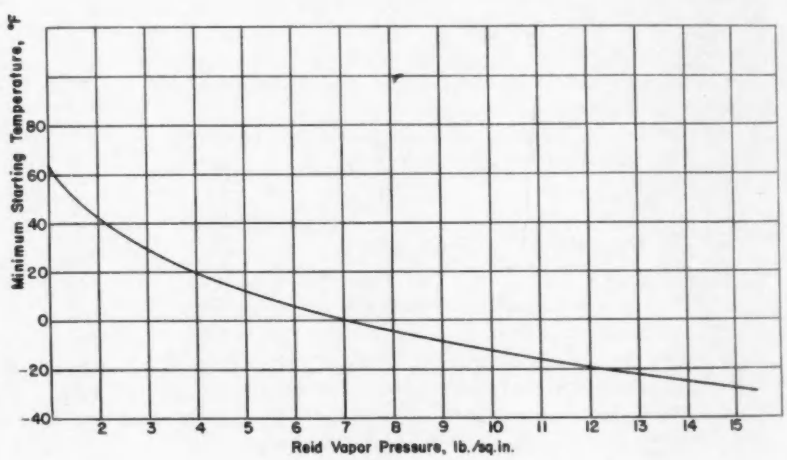
It appears therefore that, for any given set of operating conditions, it is possible to select a fuel which will prevent vapor lock. However, a number of other factors must be taken into consideration. The temperature at which the engine can be started readily decreases as the vapor pressure increases, and this condition is illustrated in Fig. 2 for typical gasolines of various vapor pressures. If a low-vapor-pressure fuel is used to avoid vapor lock during high-altitude flying, then an auxiliary fuel would be required for engine starting.

A fuel of 7 lb/in.^{2} vapor pressure is comparatively safe since, under most conditions of handling, storage, and airplane operation, the mixture of air and gasoline vapor in the tank is too rich to be explosive, as shown by curves in Fig. 3 for a system without supercharging. If now a gasoline of lower vapor pressure is used as a means of avoiding vapor lock, more hazard is introduced from an explosion standpoint. This is illustrated in Fig. 4 for a fuel of 1 lb/in.^{2} vapor pressure where it is seen that, under many operating conditions, an explosive mixture would exist in the tank. In this connection, it is of interest to note that the range of temperature throughout which an explosive mixture exists in the tank increases as the vapor pressure of the fuel is lowered. Thus, for 7 lb/in.^{2} fuel, this temperature range is about 50 F while, for a "safety fuel," the range is about 70 F. Further, the work of the CFR Committee has shown that the flash point of a fuel is considerably higher than the lean explosive limit temperature (lowest temperature for explosion), the difference corresponding to about 12 F for an average aviation gasoline

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Nov 1, 1940.]



■ Fig. 1 – Temperatures at which boiling in the fuel tank will start at various altitudes with typical fuels of specified Reid vapor pressure



■ Fig. 2 (at right) – Reid vapor pressure versus minimum starting temperature

and about 16 F for a "safety fuel." Thus, with "safety fuels" having a flash point of 100 F, the mixture in the tank may be explosive at any temperature above 84 F on the ground, and the explosive limit temperatures will decrease rapidly with altitude.

It is only necessary to mention briefly the fact that use of fuels of very low vapor pressure to avoid vapor lock will not only require a special engine starting fuel, but will also require the use of engines with solid-injection systems. Of more importance, however, is the availability of high-octane fuel. Decrease in vapor pressure of the fuel to control vapor lock decreases the availability of gasoline having a high octane number, not only from the standpoint of present production methods, but also from the standpoint of potential production. From the availability standpoint, increase in fuel vapor pressure is desirable.

Summarizing the situation from the fuel side of the problem, decrease in vapor pressure would permit operation at higher altitudes without vapor lock, but would make engine starting more difficult, would increase explosion hazards (except during tank filling where the hazard would be reduced) and would reduce availability of high-octane fuel. From every standpoint except vapor lock, increase in fuel vapor pressure is desirable. This means that all other means of controlling vapor lock should be exhausted before consideration is given to lowering of fuel vapor pressure.

■ Control by Tank Supercharging

Supercharging of the fuel tank above a predetermined altitude is a theoretically possible answer to the vapor-lock problem. The advantages to be gained are illustrated in Fig. 3 for 4 lb/in.^{2} supercharging. It is seen that, without supercharging, boiling of the fuel in the tank might be expected slightly before an altitude of 20,000 ft was reached with a fuel temperature of 100 F. With 4 lb/in.^{2} supercharging, boiling could be avoided up to about 40,000 ft. If air is used for supercharging, there will be a raising of the rich explosive limit temperatures, but these temperatures are outside the range of operation under conditions where supercharging would be required and, in any case, inert gas could be used.

By far the simplest method is to let the tank do its own supercharging. If the vent on the tank were closed just after the fuel started boiling, or at some predetermined altitude, liquid could still be withdrawn readily and the pressure in the tank would prevent the fuel from boiling at any higher altitude. During the climb, the pressure differential between the inside and outside of the tank would increase so that some redesign of the tank might be necessary in order to withstand this pressure difference if the airplane were flown at extremely high altitudes.

As fuel is withdrawn from a self-supercharged tank, the vapor pressure of the fuel and hence the pressure in the tank will decrease somewhat. Information on this point was obtained by the CFR Committee on 26 present-day commercial aviation gasolines. For the average aviation gasoline at 100 F, withdrawal of fuel from a full tank until it was only 20% full would lower the vapor pressure approximately 0.3 lb/in.^{2}, whereas draining out 98% of the fuel would result in a decrease in pressure amounting to slightly over 1 lb/in.^{2}. Considering a case in which the vent on a full tank was closed at 18,000 ft during the climb and most of the fuel was withdrawn from the tank at a

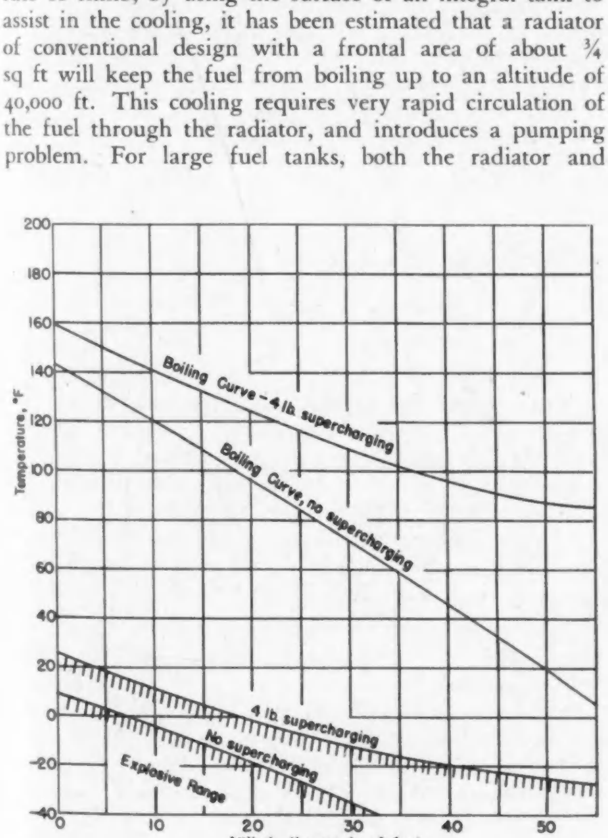
higher altitude then, during the descent, at 18,000 ft altitude, there would be a negative differential pressure of about 1 lb/in.^{2} in the tank on the average or, in extreme cases, it might amount to 2 lb/in.^{2}. Accordingly, some consideration might be necessary in the redesign of tanks to withstand negative pressures of the magnitudes indicated.

The principle of self-supercharging of tanks, equipped with automatic valves on the tank vents, is worthy of serious consideration as a remedy for vapor lock, even though there are problems involved in connection with strength of fuel tanks and other tank design features.

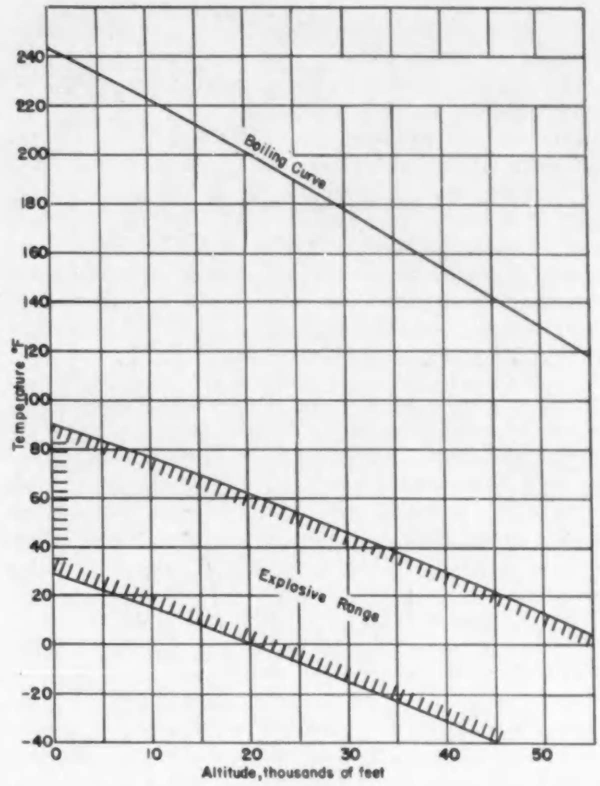
■ Control by Fuel Cooling

Cooling of the fuel appears to be the most rational answer from a theoretical standpoint. Ground cooling of the fuel tanks is by far the simplest method and the gasoline should be cooled to a temperature somewhat below the boiling point of the fuel at the highest altitude likely to be reached. Thus, if it is planned to go to 40,000 ft, the fuel should be cooled on the ground to about 40 F. However, many cases arise where ground cooling is not practicable.

The fundamental answer on fuel cooling is to cool the fuel in the tank. On the other hand, from the practical standpoint, cooling of the fuel in the tank involves considerable difficulty due to the low temperature differential at low altitudes and to the rapid rate of climb of some types of airplanes. Considering a 100-gal tank with the fuel initially at a temperature of 100 F, and a very high rate of climb, by using the surface of an integral tank to assist in the cooling, it has been estimated that a radiator of conventional design with a frontal area of about ¾ sq ft will keep the fuel from boiling up to an altitude of 40,000 ft. This cooling requires very rapid circulation of the fuel through the radiator, and introduces a pumping problem. For large fuel tanks, both the radiator and



■ Fig. 3 – Boiling curves and explosive ranges with no supercharging and with supercharging of 4 lb per sq in. – 7 lb per sq in. vapor pressure gasoline



■ Fig. 4 – Boiling curve and explosive range for 1 lb per sq in. vapor pressure gasoline

circulating-pump sizes would be serious considerations. There is a possibility, of course, of cooling only a small tank for use during the climb and for a short cruising period at altitude, until the fuel in the other tanks had become quiescent as the result of evaporative cooling. However, in such a case, the evaporation losses might become excessive, reaching 10% or even higher. Use of blast tubes in the fuel tanks would solve the circulating-pump problem, but would involve both structural and aerodynamic considerations. In large airplanes, it is possible that refrigerating systems might be employed to cool the fuel, but the problem still remains of getting rid of the heat. One point in connection with the system of cooling the fuel in the tank which should not be overlooked is the possibility of wing icing near the tanks during a descent if the airplane passes through a layer of high humidity.

■ Control by Compromise Methods

In the foregoing analysis of the high-altitude vapor-lock problem by the CFR Committee, three methods of solving the problem have been discussed, all of which involve special difficulties which are not necessarily insurmountable. While these methods have been given some consideration, the general trend in the industry has been toward minimizing vapor-lock troubles through juggling of the fuel system design and installation. It is admitted that considerable increase in altitude without vapor lock can be obtained by proper attention to fuel system design and installation if the basic principles involved are really put into practice. These principles are as follows:

(a) Install the fuel system so that vapor removal is easy and so that there are no vapor traps.

(b) Minimize the pressure drop through the fuel system.

(c) Push the fuel rather than pull it – in other words, use remotely driven pumps.

(d) Avoid bypassing excess fuel around the pump from a high-pressure to a low-pressure area.

Principle (a) has been generally recognized, and most fuel systems are free from vapor traps. However, more can be done towards proper sloping of the lines from the tanks to the tank selector valve so that vapor can pass back into the tank against the gasoline flow. The other three principles will be discussed in more detail in the following sections of this report.

■ Pressure Drops in Component Parts

With engine-mounted fuel pumps, the fuel is drawn by suction from the tanks to the engine, assisted or hindered by hydrostatic heads, and every lb/in.^{2} pressure drop in the fuel lines is equivalent to adding this same pressure drop to the vapor pressure of the gasoline at the temperature existing in the tank. For example, if two airplanes using 7 lb/in.^{2} fuel have fuel systems with pressure drops of 0.5 lb/in.^{2} and 1.5 lb/in.^{2} respectively, the first will be operating on the equivalent of 7.5 lb/in.^{2} fuel and the second on 8.5 lb/in.^{2} fuel, a difference of several thousand feet in attainable altitude before vapor lock occurs.

Since pressure drop is the basic consideration in fuel-system design from the vapor-lock standpoint, the experimental work of the CFR Committee during the past year has been concentrated on this phase of the problem. Pressure-drop data have been obtained on tubing, bends, and on practically all of the commonly used fittings and accessories covering the various sizes from ⅜ to 1 in., and using a range of gasoline flows in each case. Much of this information already has been reported in detail to the airplane and fittings manufacturers, and the remainder is now being analyzed and will be available in the very near future.

In the early pressure-drop work, data were obtained on the resistance to flow of mixtures of liquid and vapor, and it was found that the resistance to flow increased with increasing amount of vapor present, for the same mass flow, becoming twice as much for equal volumes of liquid and vapor as for solid liquid flow. Since the relative volumes of liquid and vapor flowing through any given airplane fuel system in service is unknown, and since vapor lock will probably occur if more vapor than liquid is flowing through the system, it was decided to concentrate all of the work on pressure drops with solid liquid flow, with the knowledge that a factor of safety would be necessary to take care of vapor in the fuel lines.

Analysis of the laboratory data indicated that all of the measurements on any of the units investigated could be correlated within experimental error by means of the equation

$$\Delta p = CM^{1.75} \qquad (1)$$

where Δp = pressure drop across unit in lb/in.^{2}
M = mass flow in lb/hr.
and C = constant, characteristic of the design and size of the test unit and of the gasoline characteristics.

In the correlation of the data, primary consideration was given to the additivity of C values. Thus, in obtaining the C value for a tube-to-tube fitting from the C value for the test unit, the effect of any disturbance in the exit stream from the fitting was incorporated in the C value for the fitting. This was accomplished by assuming the C value per foot of tubing to be the same in the entrance and exit positions, so that, by subtracting the C values for the tubing and the pressure taps from the experimental C value for the unit under test, a C value for the fitting was obtained which was additive. In the case of accessories, which had both entrance and exit fittings, it was necessary to go still further and incorporate into the C value for the accessory the effect of any disturbance carried over from the accessory into the exit fitting and exit tubing. The experimental data showed that the magnitude of this carry-over of disturbance from the accessory was independent of the particular exit fitting within experimental error. Further, it was found that the C value of an accessory fitting was the same regardless of whether the given fitting was in the entrance or exit position. By means of these mathematical devices to ensure additivity, it was possible without loss in accuracy to assign C values to individual fittings and accessories, for any error introduced into the individual C values would be cancelled out as soon as the total C value of any section of the fuel system was computed by summation of the C values of the individual components.

The next step in the analysis of the data involved the correlation of the C values with respect to fitting size and design. Since all data are based on a gasoline of the same pertinent characteristics, the functional relationship between C and the gasoline characteristics can be neglected for the moment. The analysis indicated that the C values can be subdivided into two general classes, namely (1) those for tubing and sections of uniform bore, and (2) those for disturbances such as bends in tubing, sharp-angle bends in fittings, constrictions caused by use of hose liners, and side outlets of tees.

For the first group, namely tubing and sections of uniform bore, the hydrodynamic equations indicate that

$$Cd^{4.75} = BL \qquad (2)$$

where d = inside diameter in in.
L = length in in.
and B = constant

and all of the experimental data obtained verify this equation.

For the second group, comprising disturbances, all of the experimental data indicate that

$$Cd^{3.75} = B \qquad (3)$$

where d = inside diameter in in.
and B = constant characteristic of the type of disturbance.

With tube-to-tube fittings, d has the same nominal value at both ends of the fitting whereas, with accessory fittings, the inside diameters at the tube or hose and at the pipe end are normally different. In this latter case, d represents the inside diameter of the fitting at the pipe end.

The hydrodynamic equations indicate that B is directly proportional to the ratio $\eta^{0.25}/\rho$, where η is the absolute viscosity of the gasoline in poises and ρ is the density of the gasoline in gm/cm^3. For the gasoline used in the present work for which $\eta^{0.25}/\rho = 0.350$, the following values were found for B:

	$B \times 10^8$
Tubing and Sections of Uniform Bore	0.099
Curvature of Tubing Disturbance	$\frac{1.00}{\sqrt{R}} \frac{(\theta)}{(90)}$
Hose Liner Disturbance	0.20
Side outlet of tee disturbance	0.20
Sharp 45-Deg Angle Disturbance	1.05
Sharp 90-Deg Angle Disturbance	4.29

in the value of B for the curvature of tubing, R is the radius of curvature of the bend in inches measured to the center of the tubing and θ equals the angle of the bend.

These values of B summarize all of the extensive experimental work done on tubing, tubing bends, tube-to-tube fittings, tube-to-pipe fittings, hose fittings and hose liners with an average accuracy corresponding to the pressure drop across less than 2 in. of tubing of the same inside diameter as the fitting. For accessories, no similar correlation has been developed, and it appears probable that individual C values will have to be assigned to each accessory.

A survey of the characteristics of 26 commercial aviation gasolines, conducted by the CFR Committee, indicated that the average value of $\eta^{0.25}/\rho$ at 100 F was equal to 0.353 and that the spread of values was less than $\pm 5\%$. Accordingly, the B values just tabulated can be considered as applicable to any commercial aviation gasoline available at present, and can be employed for the accurate computations of pressure drops across all of the commonly employed parts of aviation fuel systems, except accessories which, as pointed out previously, must be considered individually. In order to obviate the necessity for computations in each particular case, the CFR Committee has made available to the airplane and fittings manufacturers tables listing the C values of the various fuel system parts, based on the average specified dimensions.

There is no experimental evidence as yet that the pressure-drop data can be used for any other liquid than present-day commercial aviation gasoline. However, it is recognized that there is considerable interest in the application of these data to aviation hydraulic systems, and the CFR Committee has authorized a program of work on other fluids so that the B values can be evaluated as a function of $\eta^{0.25}/\rho$ over the range of interest. In this connection, it is hoped that the work will also lead to a standardization of a suitable experimental fluid for pressure-drop measurements which will be less inflammable than aviation gasoline, and which will be suitable for fuel-system mock-ups and for use in plants where fire hazards must be reduced to a minimum. This is part of the program, now under way, for the development of a standardized test procedure for pressure drops and gasoline flow in fuel-system mock-ups.

One of the outstanding points noted in connection with the experimental work on pressure drops was the lack of consideration given to pressure drop in the design of fuel-system fittings and accessories. This failure to recognize the importance of pressure drop has also been observable in some fuel-system installations. Thus the pressure drop across a ¾-in. Parker EBBT elbow is equivalent to about 40 in. of ¾-in. tubing, whereas a 6-in. radius 90-deg bend in ¾-in. tubing equals the pressure drop across 2.6 in. of the tubing. Substitution of the bend in this case reduced the pressure drop to 1/15th of its former value, and this is only an average case. In locations where tubing bends cannot be employed, a design of curved elbow should be used. Without increasing the external dimensions of an elbow, but using the maximum permissible radius of curvature, the pressure drop can be reduced to one-third or more of the value with a sharp right-angle bend. Considering the large number of 90-deg elbows used in some airplane fuel systems, development and use of a streamlined design should produce a very definite gain in critical altitude under hot-weather conditions. The situation is worse in connection with accessories and many of them have excessive pressure drops, entirely outside the range of anything that appears necessary. The CFR Committee has done some preliminary work in stimulating the development of streamlined fittings and has plans for becoming very active in sponsoring low-pressure drops in both fittings and accessories.

■ Fuel-Pump Installations and Vapor Lock

Fuel-pump installations are responsible for much of the vapor-lock difficulty encountered in airplane fuel systems. In conventional installations, the fuel is drawn by suction from the tank to the engine-mounted pump, with resultant tendency towards vapor formation in the fuel system as the result of high pressure drops. This situation is aggravated still further by the fact that the pump has a higher capacity than the engine requirements, with the result that the excess fuel is bypassed back around the pump from a high-pressure to a low-pressure area, and still more vapor is released. The net effect is that the pump efficiency falls off very rapidly as vapor-locking conditions are approached and soon reaches a state where it will not supply sufficient fuel for the engine. The theoretical answer is very simple, namely, to push the fuel and not pull it by suction up to the engine, and secondly to eliminate the bypass around the fuel pump.

Recent trends have been towards a compromise solution, involving the use of a booster pump located near the tank selector valve and incorporating a vapor eliminator. This arrangement still leaves the problem of getting boiling fuel out of the tank and through the selector valve by suction, but does eliminate the effect of suction pull throughout most of the fuel system and does reduce the pressure differential across the bypass valve around the engine-driven pump.

No work has been done as yet on this phase of the problem by the CFR Committee but, with the pressure-drop work approaching completion, active interest in pump design and installation is developing.

■ CFR Committee Program

The CFR Committee program on the investigation of vapor lock in airplane fuel systems may be summarized as follows:

(1) Completion of the present program on pressure drops in component parts of fuel systems. The laboratory work is essentially completed, but considerable work still remains in the analysis of the data and the preparation of engineering tables.

(2) On the completion of the present program on pressure drops, facilities will be maintained for testing new or modified designs of fittings and accessories under comparable test conditions, at a nominal cost to the manufacturers.

(3) Extension of the applicability of the pressure-drop data to other fluids of higher viscosity, with particular reference to hydraulic oils and to the choice of a standard calibrating fluid of low fire hazard.

(4) Development of a standardized test procedure for gasoline flow and pressure drops in fuel-system mock-ups.

(5) Development of a standardized test procedure for fuel-system performance during flight.

(6) Assistance in the development of fuel-system fittings and accessories of low pressure drop.

(7) Assistance in the development of more satisfactory fuel-pump installations.

Engineering Considerations in

AUTOMOBILE METHODS

IN recent months, due to the existing international situation and the very important part which the airplane is taking in deciding the issues at stake, attention has been called to the urgent necessity for quantity production of aircraft to build up our air arm quickly to the status required for national defense.

However, it is not public knowledge that the Curtiss Aeroplane Division, Curtiss-Wright Corp., and this is true for the aircraft industry generally, has in the last six years successfully planned its projects with mass production as the first premise.

Purpose of this paper is to outline the various elements necessary in the achievement of the high industrial efficiency which has become synonymous with mass production. This purpose is accomplished largely by 76 illustrations with descriptive captions.

■ ■ ■

THE international situation has demonstrated that the effectiveness of the airplane is a decisive factor in the outcome of the issues at stake. This potent weapon has directed attention to the urgency for quantity production of aircraft essential to national defense.

With the need for increased production quantities it is natural that there should be serious interest in the application of mass-production technique in the manufacture of aircraft. Consequently, the aircraft industry is urged to utilize the experience and facilities of the world's most outstanding mass-production enterprise, the American automobile industry. Although no single business can lay claim to mass production as being its own development, the automobile industry probably has contributed the most through its bold and realistic approach.

It is the purpose of this paper to indicate what can be accomplished in planning projects and processes, involving design, tool, production, and process engineering, and to outline the various elements so necessary in the achievement of the high industrial efficiency which has become synonymous with mass production.

1. In that the term mass production has become decapitalized the true definitions are somewhat obscured and since the subject is too comprehensive to permit more than limited treatment, it is the intention, through this résumé and accompanying illustrations, to define briefly the requisites to mass production and its expectations in the manufacture of aircraft.

To the casual observer the question arises as to why the methods and practices of the automotive industry are not utilized more extensively in the manufacture of aircraft. More than superficial observation will reveal that what appears to be reluctance on the part of the aircraft industry to accept automobile production standards is due primarily to the numerical and other differences in what constitutes quantity or mass production and the influence of contingent conditions peculiar to each field. Five hundred cars per day under normal circumstances would not represent unusually high production, whereas up to recently ten airplanes of one type per day would be outstanding. The situation is complicated further by the imperative compliance to rigid weight, performance, reliability, and strength specifications in aircraft manufacture. An empty gasoline tank or a broken gasoline line in an automobile is not a serious hazard; it is an inconvenience; while, in an airplane, a similar failure usually represents an emergency.

Special and single-purpose machine tooling and extensive conveyor systems are economically essential in the manufacture of 100,000 cars wherein certain details of the cost of 1/100 of a cent become important. In aircraft construction the application of such refinements obviously cannot be as extensive. It is not to be construed, however, that the aircraft industry is barren of opportunities to apply advanced production methods. Careful analysis supports the conclusion that much can be gained through the utilization of automobile methods and practices when combined with a liberal application of common sense.

Before proceeding further, it will be helpful to clarify some fundamental principles inherent in what is commonly known as mass production. The term is somewhat misleading. The general belief is that it involves only the manufacture of a large number of identical parts, usually by means of special machine tools. Quantity alone does not make mass production, however, mass demand is a reciprocal factor[1]. Therefore, it is necessary to differentiate between large quantity and mass production. A change in the production demand from 100 to 1000 parts, with certain time limitations, requires more than merely moving the decimal point on a shop order. It may necessitate redesign or re-tooling of the part as dictated by conditions.

Mass production results in uniformity, and this consequence is evident in the production of many things that are a part of everyone's daily activity – wearing apparel, cigarettes, even in the canned and packaged food we consume. This accurate duplication of results which changed luxuries into necessities can best be illustrated in that, although a single craftsman can produce an object of superior quality, it is unlikely – except under the most propitious circumstances – that the same craftsman can duplicate the original results exactly. Nor is it reasonable to expect that artisans of equal skill using average tools can produce identical results unless the operations in-

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Oct. 31, 1940.]

[1] See the *New York Times*, Oct. 20, 1940: "Standards Backed in War Materials."

the Application of to AIRCRAFT PRODUCTION

by DON R. BERLIN and PETER F. ROSSMANN
Curtiss Aeroplane Division, Curtiss-Wright Corp.

fluenced by individual judgment, tending to cause variations in the final product, can be controlled.

Therefore mass production may be defined as being the most efficient utilization of the human element. It can be accomplished by observing four requisites which may be developed in greater detail. Additional elements could be added; however, only the general considerations will be treated. The following are listed in sequence rather than in the order of their importance.

(a) Product Engineering
(b) Tool Engineering
(c) Cost Analysis and Regulation
(d) Production Control

There are many benefits derived by reduction of, and through the elimination of, manufacturing variations. Namely, when guesswork is removed, accurate timing and cataloging of work elements are possible, and output predictions can be made.

Interchangeability is a desirable product of reduced manufacturing tolerances that automatically eliminates the need for selective fits which represent a definite limitation on output and control. These advantages can be obtained by special machine tooling and processing, but in the automobile industry mass production begins on the drafting board when the design is in an embryonic stage.

This suggests the first consideration in that it has been found advantageous to have design personnel familiar with machining and other factory processes. This treatment of the part is production engineering and is very closely allied with tool engineering and costs.

It then becomes the problem of determining or establishing the extent to which the engineer in the development of a product is constrained to permit tooling methods and manufacturing processes to influence design. Unquestionably, in this regard, both engineering and manufacturing are vitally obligated in assuring the ultimate utility as well as achieving the most economical method of fabricating the product.

The importance of correlating the design of a product and the problem of providing tools for its manufacture, is not to be minimized. Often-times the significance of this mutual responsibility has not been demonstrated or analyzed.

Engineering is basically concerned with the creation or development of a new device or product, or an improvement on one in current use, the design of which is dictated by certain conditions – trend of the art, sales appeal, improved durability, weight and strength requirements – some of which permit no deviation from prescribed specifications.

The project of tooling, on the other hand, is that of utilizing, wherever possible, available standard machine tools, selecting, designing and providing suitable jigs and fixtures, assuring the maintenance of the specified degree of accuracy, and planning the manufacturing operations to obtain the lowest possible production cost.

Unless the product designer is familiar with manufacturing processes and production methods, the impression might be gained that the production tool engineers in their analysis of the design are unnecessarily critical and do not fully appreciate its functional requirements. The tool engineer's attitude has justification and deserves explanation in that the recommendations based on the production studies of the part are influenced, if not wholly determined, by the quantity and type of machine-tool equipment, and the magnitude of the appropriations allotted for tooling and, to a very great extent, by the number of parts to be produced.

Often a slight change in a casting or forging – a modification to simplify basic design or increased tolerances – makes appreciable savings in tool and manufacturing costs. Also, attempts to reduce cost through redesign frequently result in seriously penalizing weight and performance. They are also closely related to serviceability since liberal tolerances, and in some cases, incomplete interchangeability, might be economical from a production standpoint but the cost of subsequent service because of such conditions might be excessive, and very often the difference between profit and loss. Engine cowling is in this category.

However, it should be considered that the product engineer should not normally be fettered by precedent to the extent that the expression of ingenuity and invention in the design is restrained. Frequently the development and perfection of new and original processes necessitated by engineering design are distinctly advantageous in that, if the idea "clicks" as it were, the competitors are handicapped in duplicating or approximating the new designs if a special process is vital to its fabrication.

In general, the trend in manufacturing processes should tend to permit the engineer more latitude in designs.

Tool engineering is influenced greatly by quantity and the basic design of the part. Properly planned and executed tooling contributes much toward the fulfilment of the mass-production principle particularly:

(A) Properly planned operation sequences
(B) Simple foolproof jig-and-fixture design
(C) Gaging and locating points common to mating parts
(D) Non-fatiguing work heights and positions
(E) Reasonable application of time-and-motion studies
(F) Providing adequate inspection tools

The advantages and necessity of proper tooling cannot be over-emphasized since the manufacture of interchange-

able parts is the indispensable requisite precedent to mass production.[1]

Therefore, in the production of increased quantities of aircraft, it is insufficient merely to expand proportionally the existent manufacturing facilities.[2] Better tooling acquires added significance since the product, as influenced by mass demand, must be produced more quickly and more economically. In this regard, advantages can be derived by encouraging increased respect for the automobile industry's attitude and dependence on effective tooling.

The inspection item, although limited in its application (by quantity requirements), is of more importance than is generally conceded; it is one of the most essential factors in controlling variations in the product. Inspection expedites production, witness for example, the extensive and ingenious automobile inspection tools. Therefore, it is imperative that provisions should be made in the airplane tooling budget for inspection equipment. Good inspection tools inspire greater confidence in inspection results.

The third requisite, accurate "Cost Analysis and Regulation," is more than keeping books to record expenditures. The automobile industry is cost-conscious and the modern car would be impossible without this consideration. Accurate estimates are necessary for reliable cost predictions, and there is sufficient precedent and cataloging of aircraft manufacturing operation data to be of material assistance in compiling useful cost estimates. In fact competition in the aircraft industry should give the necessary impetus to this phase of management. Cost regulation is the applied art of intercepting costs before the money is expended.[3] All costs should be challenged and justified; otherwise there can be no confidence in the operation of the expense budget. Time studies may be considered part of the analysis. More can be said on this subject of costs, however; there are authoritative publications indicating useful data which can be consulted in great detail.

The fourth requisite, "Production Control," involves principally scheduling and material handling. The former requires accurate data on equipment and man-power capacities so that an orderly flow of production can be maintained with the minimum of congestion due to hold-ups, changes, shortages, and so on. This control in automobile manufacture is the nerve center of production and the clock-like precision with which cars are produced is significant evidence of its effectiveness. Another element of such control that has had practically no recognition in the aircraft industry is the application of economical lot size (the economical cycle) or as it is more generally known the most economical number of parts to make in a set-up. Heretofore, there has been limited opportunity to apply this science in aircraft, but now circumstances are much more favorable to its adaptation.

Material handling can be regarded as part of production control, and it is a very much neglected phase of manufacturing. Its function is more than regulating and moving material through the plant. In fact, an efficient plant layout is planned on the basis of proper routing of material between the process stations, assembly departments, storage depots, and so on. Conveyors are a means of material handling, and planned layout studies prescribe them when circumstances justify their use. Providing adequate supplies of material at the various process stations and the maintenance of shortage lists by stock chasers are also material handling.

Finally, there is the problem of designing the experimental airplane. This has always been a moot point, there being conflicting opinions as to the best solution of the problem. In the automobile industry there are different influences with regard to sales, customer's desires and requirements, trend of the art, and so on. Furthermore, there is this important difference: the automobile industry flourishes under more advantageous conditions because that type of product can be slanted toward a sales field with every assurance that the experimental model will finally go into active production. In addition, the experimental costs can be amortized over a larger number of units. With aircraft, all experimental projects are likewise considered as production possibilities but this cannot be regarded with the same degree of certainty. Therefore, designing the experimental model for production on as complete a scale as is the case with automobiles has not been found practicable. Of necessity, progress in this regard has been cautious. The Keller method of reproducing forgings, fittings, and so on, represents a reasonable approach. Primarily, designing for production is a specific problem which has many ramifications, and it is suggested as an interesting subject for special analysis.

In conclusion, it must be said that the aircraft industry is unique in its development and resourcefulness under the handicap of small quantity production which circumstance has stimulated ingenious methods and techniques adapted to its special requirements. The similarities between the automobile and the aircraft industry are greater than the differences, and the exchange of ideas is a logical development and conducive to progress if the art is not merely utilized but advanced.

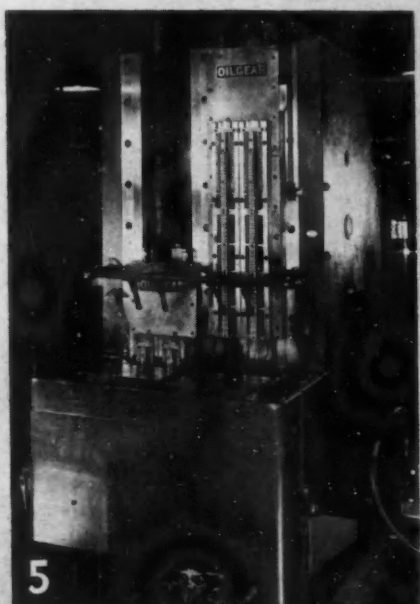
[2] See *Analyst*, August, 1940: "Mobilization of the Aircraft Industry: Mere Plant Expansion No Answer."

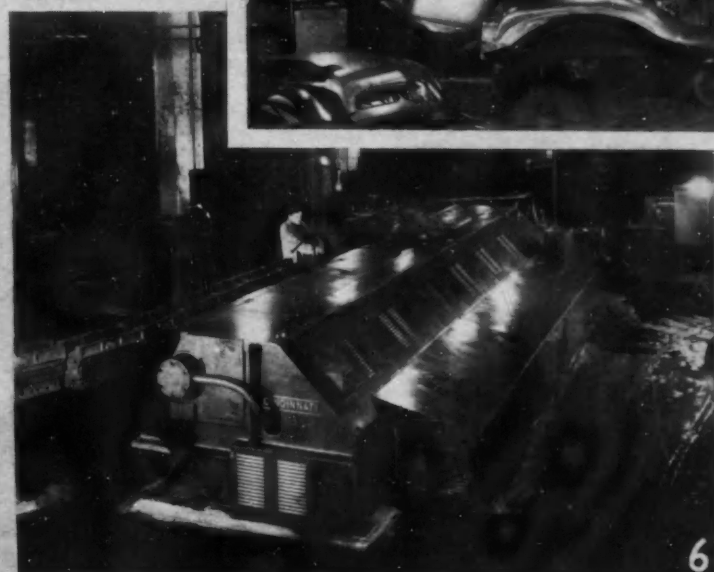
[3] See "Budgeting Expense and Cost of Handling Materials in Automotive Plants," by George Miller, presented at the Annual Meeting of the Society, Jan. 15, 1937.

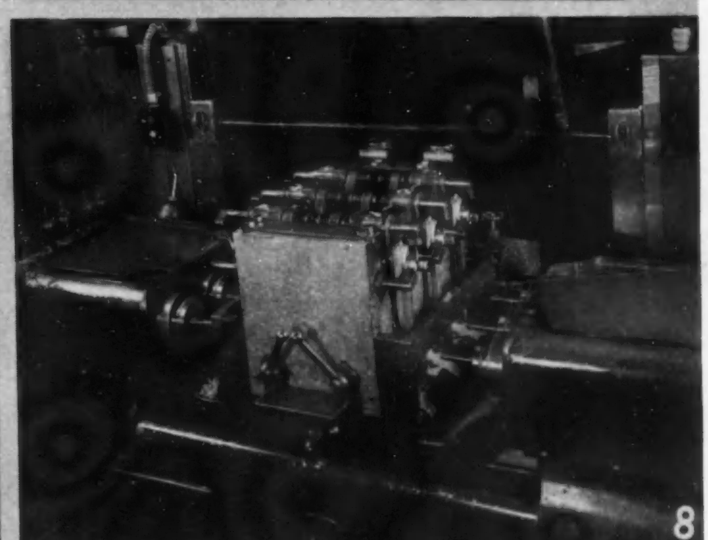
Fig. 1 (left) – The original conception of automobile body style and interior appointments independent of mechanical features is made by stylist artists

Fig. 2 (right) – The experimental car mock-up in which cars are modeled full-scale or in clay is comparable to the experimental airplane model

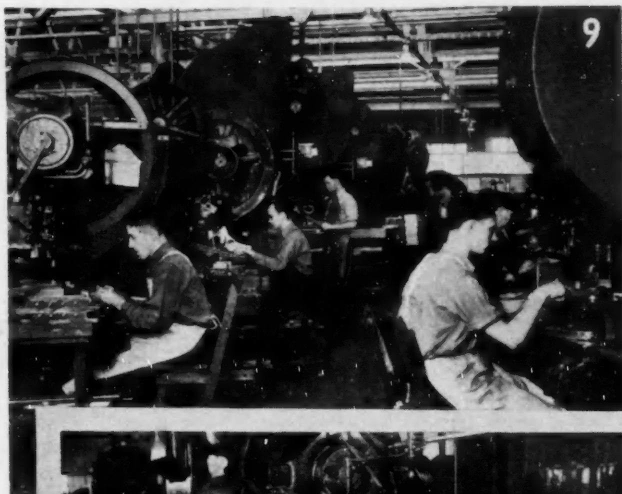
Single-Purpose and Labor-Saving Automobile Tool Equipment (Figs. 3-8)

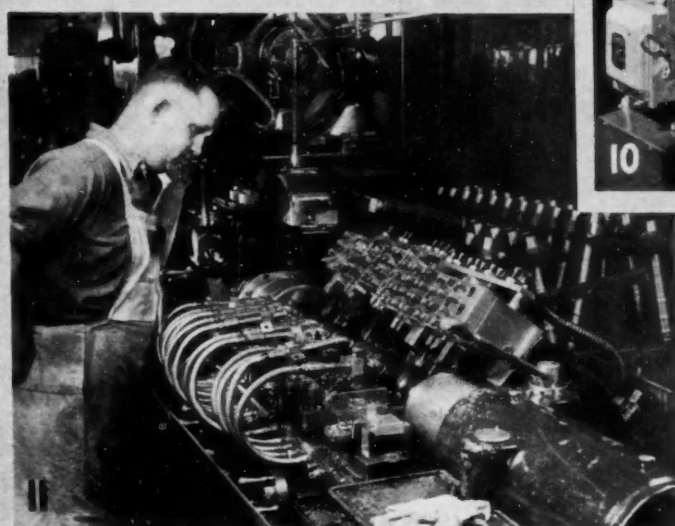


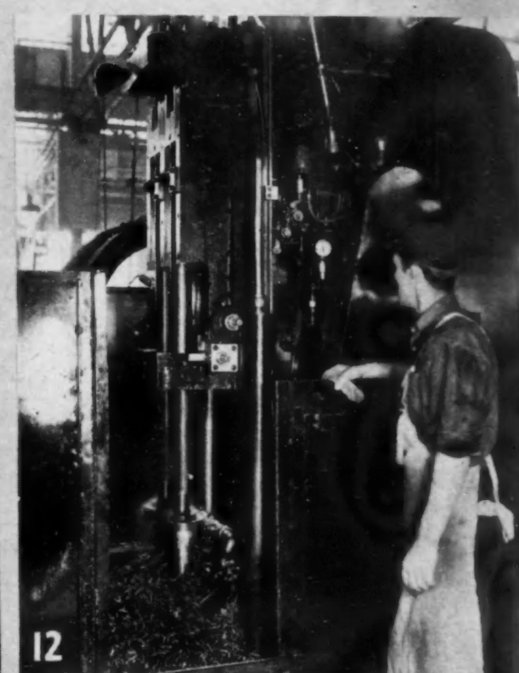


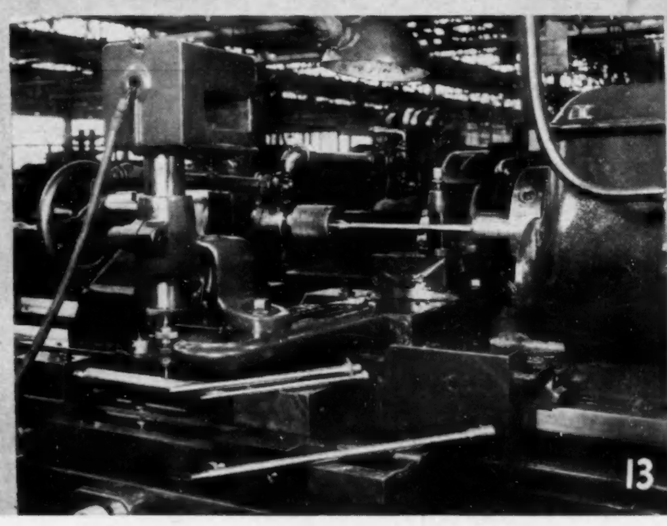


- Fig. 3 – Quantity and the trend of the art justify this large "turret-top" die
- Fig. 4 – Removing "turret-top" stampings from die
- Fig. 5 – Special tooling such as surface broaching is extensively used in autcmobile production
- Figs. 6 and 7 – An outstanding example of surface broaching applied to cylinder blocks
- Fig. 8 – Multiple precision boring of connecting rods is another single-purpose machining operation on automobile parts









■ **Fig. 9 – The Punch Press Department arrangement at Curtiss is comparable to similar layouts in automobile plants**

■ **Fig. 10 – Honing machine for loading-gear links**

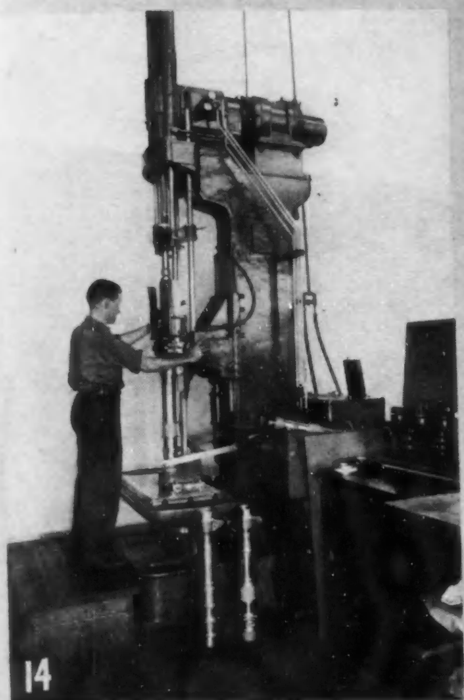
This machine has eliminated the necessity of hand-reaming holes and the subsequent variation in the finished product. This may be considered a single-purpose machine but may be accommodated to a variety of parts requiring such an operation

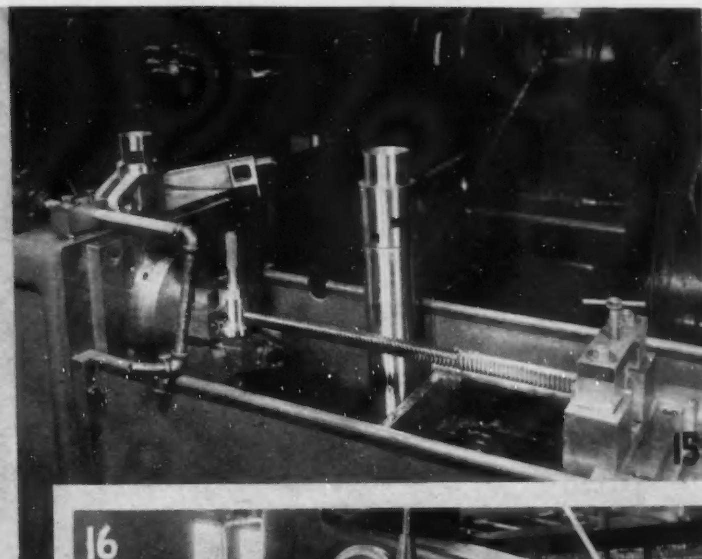
■ **Fig. 11 – Automobile practice such as the set-up shown on the Fay Automatic for machining airplane landing-gear cylinders and pistons is typical of the trend to mass-production machining**

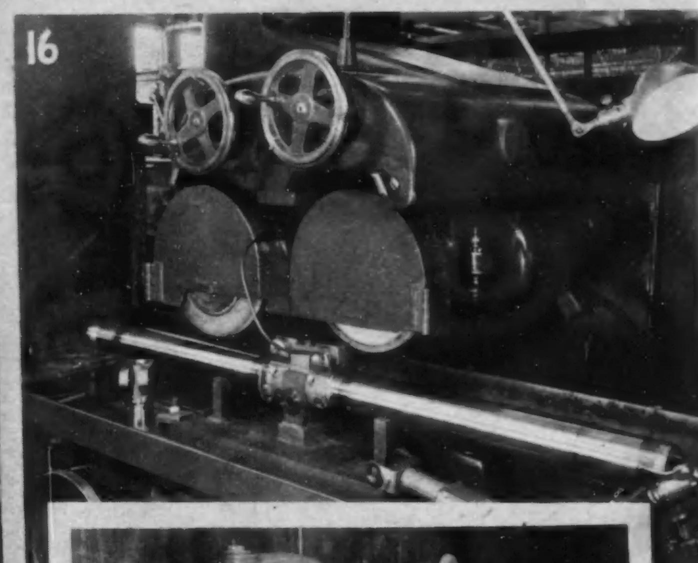
■ **Fig. 12 – The use of the Baker drill for rough-drilling the holes in landing-gear oleos and pistons completely eliminated the engine-lathe method of drilling these parts**

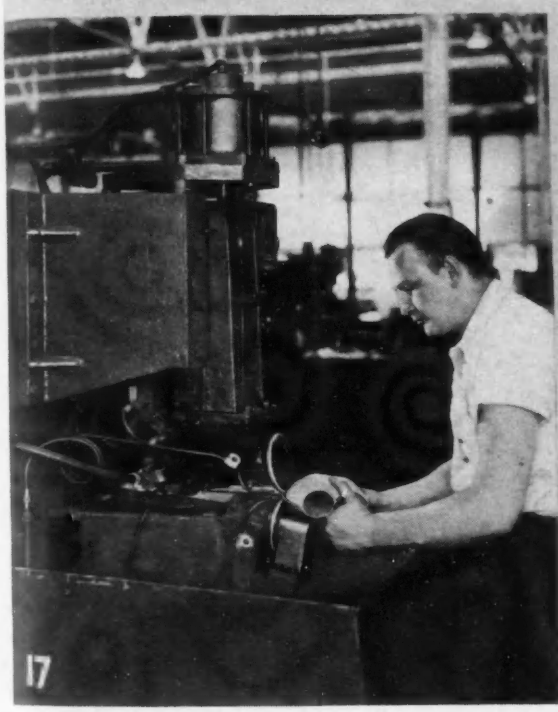
■ **Fig. 13 – The use of automatic contour-reproducing equipment is illustrated in this photograph in which a Keller attachment is used to turn a multiplicity of diameters and tapers on a landing-gear metering pin**

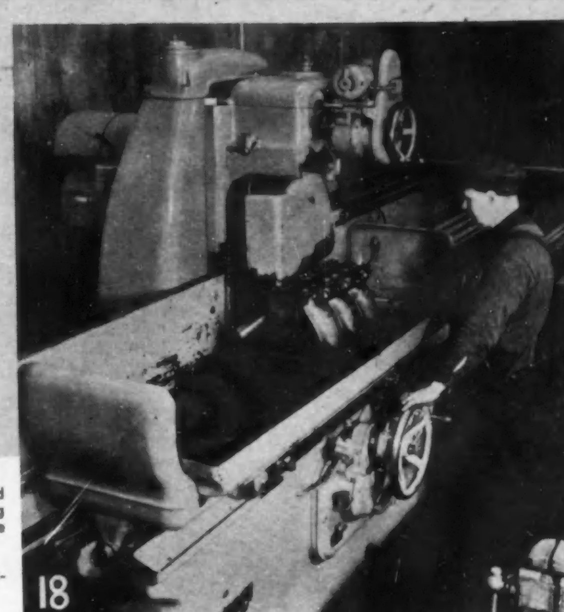
Equipment in Airplane Production at Curtiss (Figs. 9-18)











■ Fig. 14 – The honing of landing-gear cylinders is quite highly developed and it has been discovered through production tests that the grinding operations on the I.D. cdn be reduced by honing and it is now possible only to true-grind before honing

■ Fig. 15 – Broaching is a modern production technique and is extensively used on landing gears and other similar parts

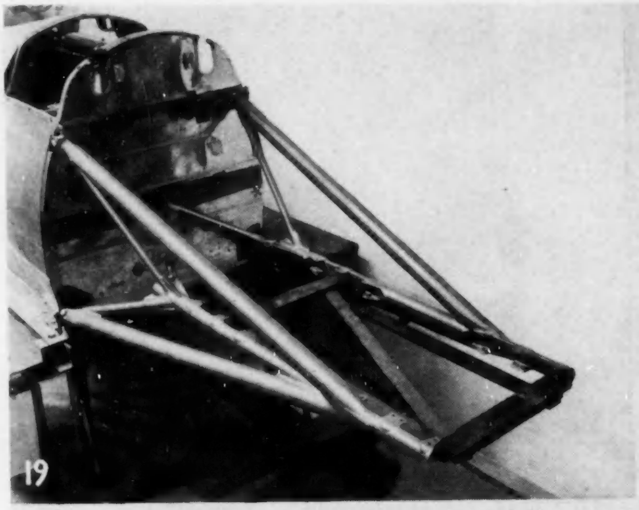
■ Fig. 16 – A special grinding machine developed for grinding both the outside and/or the inside landing-gear cylinder lugs and links

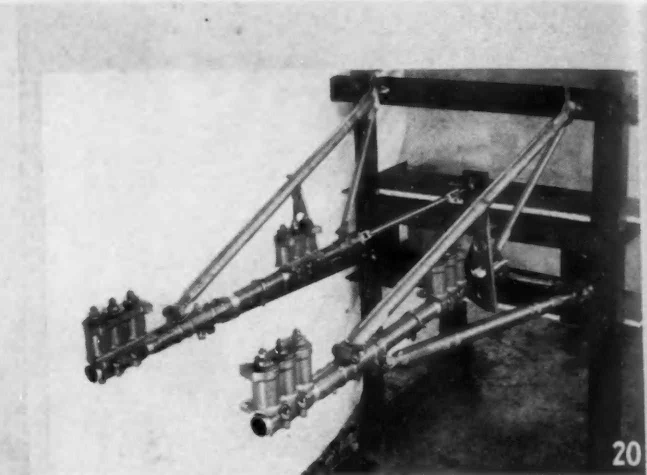
The need for extremely close tolerances as affecting interchangeability and service necessitated the development of this machine. These close tolerances prescribed by engineering design, which at first seemed unwarranted from a cost standpoint, have been subsequently justified by a complete absence of any service or interchangeability problem on these parts

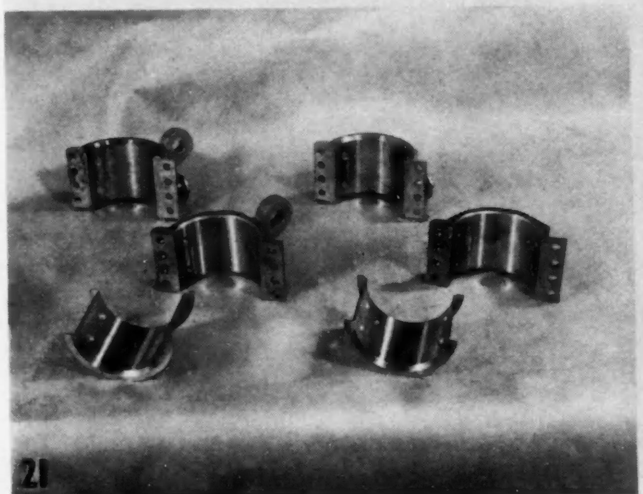
■ Fig. 17 – The seam welding of exhaust stacks is a definite trend toward high-production operations

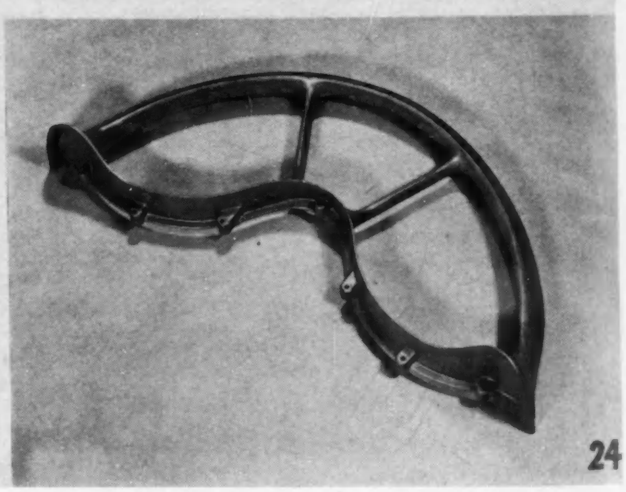
■ Fig. 18 – The multiple grinding of the joint surfaces on exhaust stacks is good production practice

Examples Representing Design Consideration in Particular (Figs. 19-24)









■ **Figs. 19 and 20 – The change from a completely gas-welded engine mount to one on which gas welding has been confined to such points where reasonable control can be exercised has resulted in an engine-mount design on which most parts are machinings involving bolted joints**

This arrangement permits the maximum concentration of man-power on detail parts and makes the sub-letting of such parts more practicable

■ **Fig. 21 – The lower trunnion bearings on the landing gear represent a case in which extremely close tolerances in the order of 0.0005 in. have been found economical in that hand fitting was eliminated and complete interchangeability was achieved**

■ **Figs. 22 and 23 – The trend from gas-welded junction boxes to die castings is clearly emphasized in these photographs; a completely interchangeable product is obtained**

■ **Fig. 24 – Die casting of the leading edge of the oil coolers and Prestone cooler scoop**

This construction resulted in a substantial saving in that parts were all interchangeable and no hand fitting was required. This die casting is one of the largest made for aircraft measuring approximately 24 in. at its maximum dimension

■ Fig. 25 – The deep drawing of automobile instrument boards after the application of synthetically grained surfaces clearly emphasizes the advance in the automotive practice of forming prefinished parts

■ Fig. 26 – The technique shown in Fig. 25 is utilized in a different way by the application of cellophane in the drop-hammer operations for airplane parts

The cellophane eliminates the scratches normally produced in such an operation and obviates the need of extensive polishing after forming

■ Figs. 27 and 28 – The old technique of forming cowling with rubber on a zinc or aluminum die as compared with the new method of drawing in steel-faced wood draw dies

By the draw-die method the need of hammer bumping to work out wrinkles to obtain the desired contours is eliminated

■ Figs. 29 and 30 (below) - Comparison of methods in making the cowl shown in Figs. 27 and 28

It can be seen readily that the rubber-forming method would never be less than $10.42, even if the tools were completely amortized; whereas the draw-die method would approach the low figure of $1.39 at complete amortization of tools. Particular attention is directed to these two figures since, although the tool cost for the draw die is almost four times as great as for rubber forming, the two methods equalize at a point as low as 49 parts. This requires a little further explanation of the use of the chart and the formula. When the total quantities to be made are known, it is relatively simple arithmetic to determine the most economical method of fabricating the part. However, it is very often desirable to know at which quantities either of the two methods would be suitable. This at first may seem unimportant, but very often quantities on an order may be reduced and this knowledge therefore becomes important since it is an accurate clue to economical processing at smaller quantities. It also can be stated that the elimination of the set-up cost from the formula in this particular case would only influence the economical number of parts by approximately one part

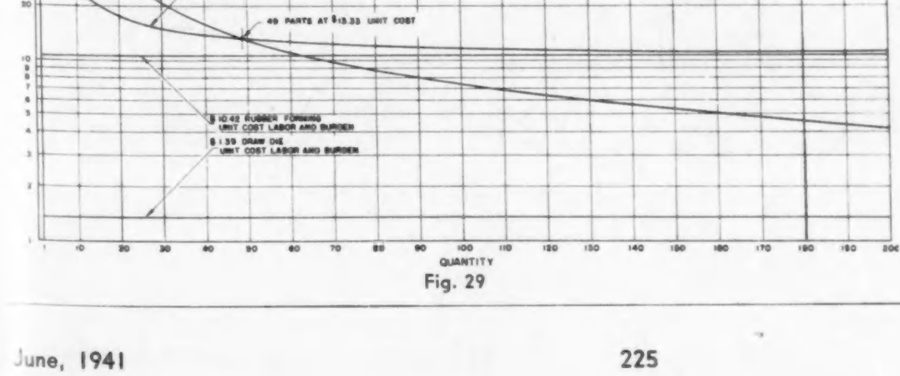


Fig. 29

PROCESS NO. 1	(DRAW DIE)
L = UNIT COST --- LABOR AND BURDEN	$ 1.39
T = TOOLS AND SET-UP COST	$585.00

PROCESS NO. 2	(RUBBER FORMING)
L_1 = UNIT COST --- LABOR AND BURDEN	$ 10.42
T_1 = TOOLS AND SET-UP COST	$142.50
N = ECONOMICAL NUMBER OF PARTS AT EQUAL UNIT COST - LABOR, BURDEN, TOOLS AND SET-UP.	

(1) $\frac{T}{N} + L = \frac{T_1}{N} + L_1$

(2) $\frac{585}{N} + 1.39 = \frac{142.50}{N} + 10.42$

(3) $585 + 1.39N = 142.50 + 10.42N$ (MULTIPLY BY N)

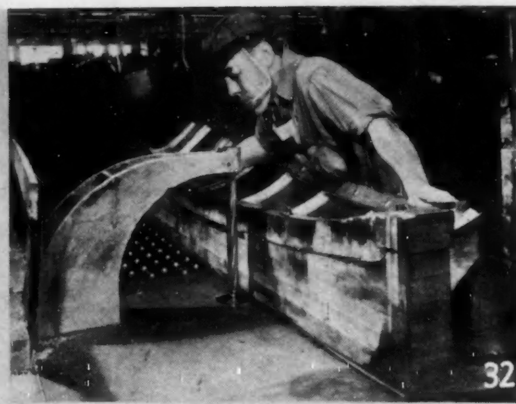
(4) $585 = 142.50 + 9.03N$ (SUBTRACT 1.39)

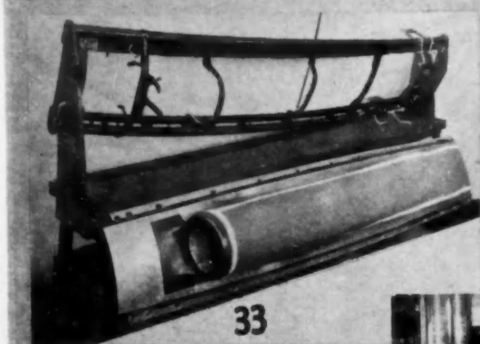
(5) $9.03 = 442.50$

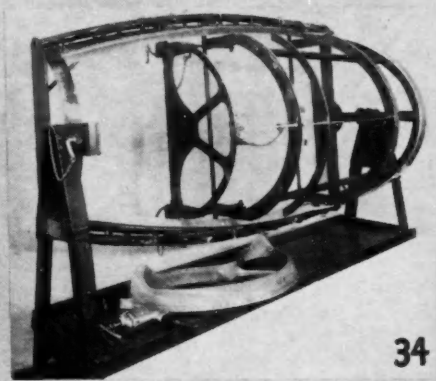
(6) $N = \frac{442.50}{9.03}$

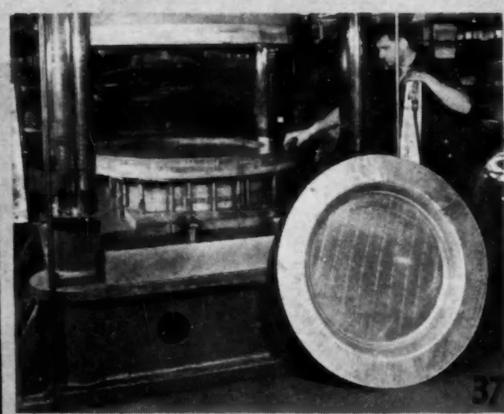
(7) N = 49 PARTS

Fig. 30









▪ **Figs. 31 and 32 – Old and new methods of trimming engine cowling**

The need of expensive trimming dies has been considered unnecessary by the use of this router form which makes such parts interchangeable

▪ **Figs. 33 and 34 – Sturdy, accessible, swinging-type cowl-assembly jigs are utilized exclusively**

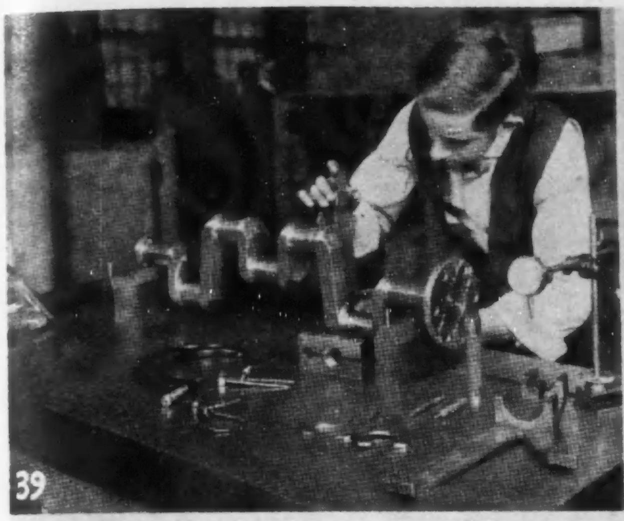
▪ **Figs. 35, 36 and 37 – The trend from zinc blocks for hand forming to steel faced wood draw dies is shown**

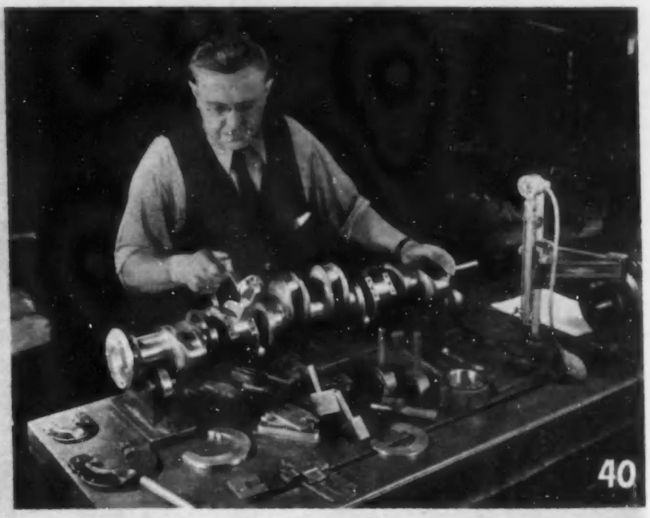
The wood used is maple and the facing is chrome-moly or carbon steel plate. Not all of these draw dies bottom but merely stretch the material over the punch and by virtue of draw beads achieve the proper shape

▪ **Fig. 38 – Gang riveting has developed high-production riveting**

▪ ▪ ▪

Typical Automobile Inspection Techniques (Figs. 39-44)





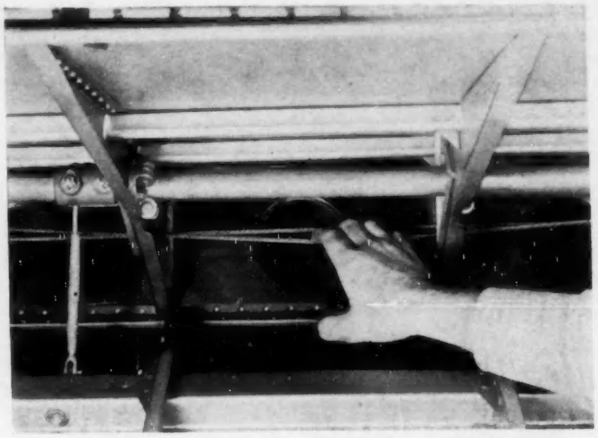
■ Figs. 39 and 40 – Comparison of the early method and the present method of inspecting automobile crankshafts

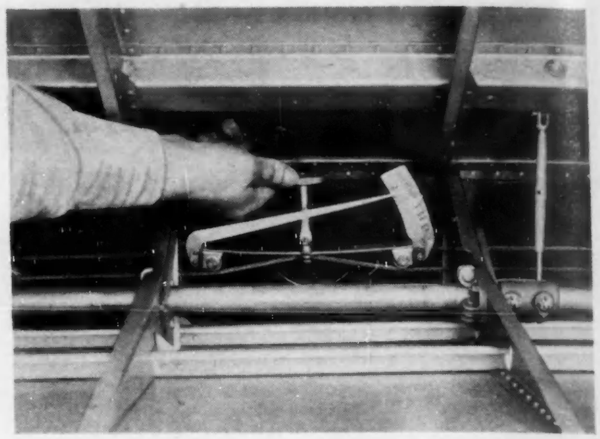
The modern trend in which all of the inspection equipment is of the direct-reading or automatic type reduces to the minimum the amount of individual judgment necessary to check such parts

■ Figs. 41 and 42 – Optical methods are extensively used in automobile inspection techniques

■ Figs. 43 and 44 – A further example of the direct-reading type of inspection on crankshafts and gears

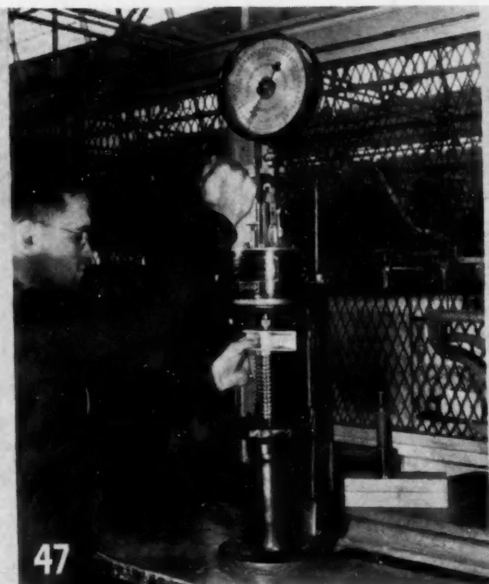
Aircraft Inspection Methods (Figs. 45-63)



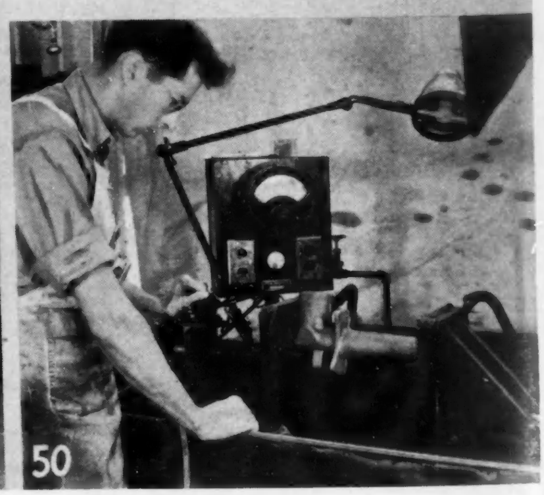


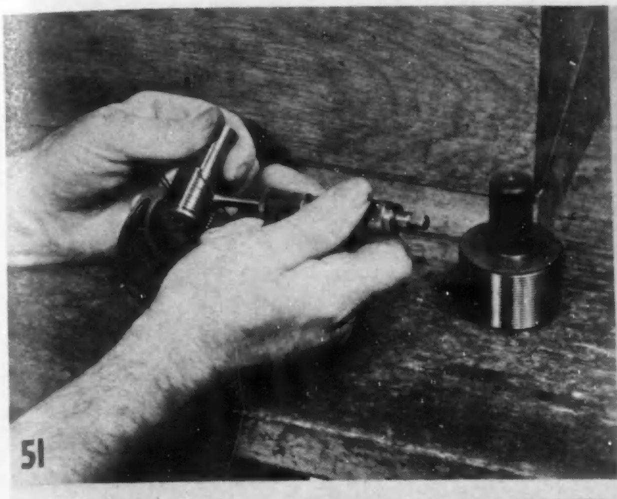
■ Fig. 45 (left) and Fig. 46 (right) – Old and new methods of checking cable tensions on aircraft

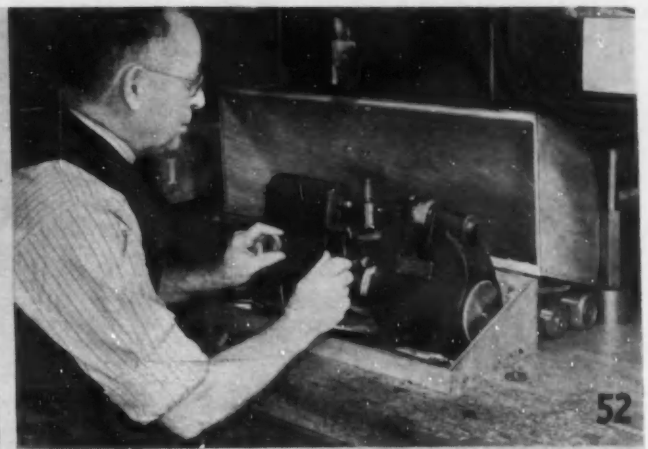
■ Figs. 47 and 48 – A comparison of the manual and automatic type of hardness testing

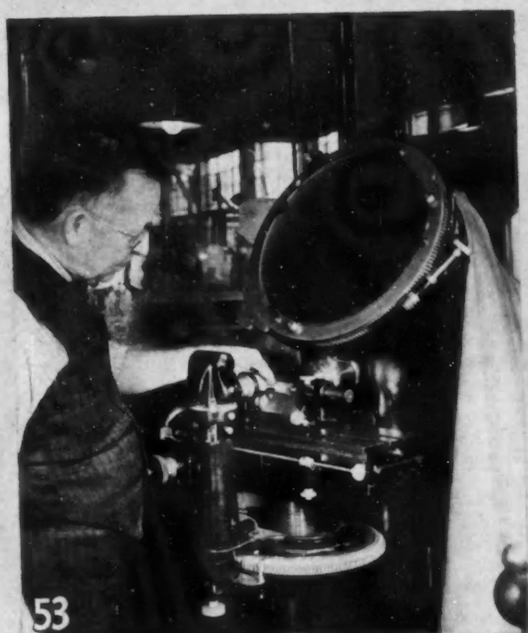


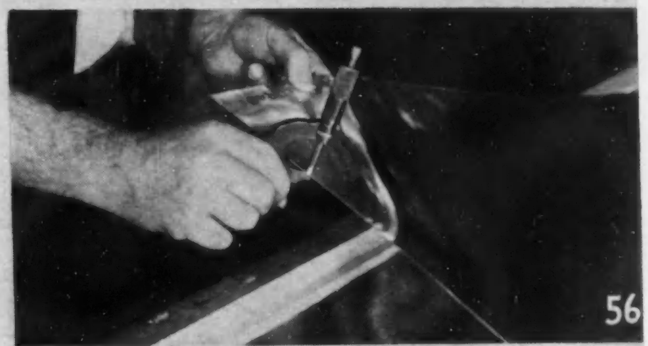
■ Figs. 49 and 50 – The evolution from visual to magnaflux testing of steel fittings is an example of the application of scientific production methods

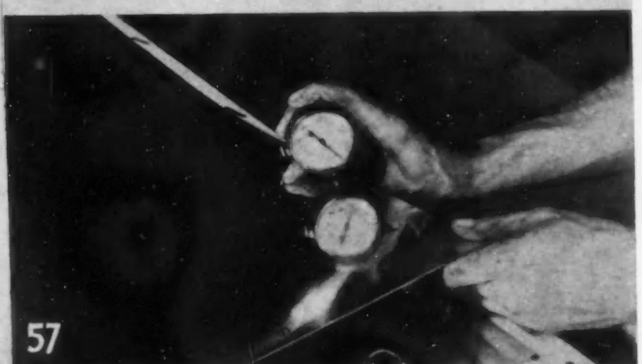


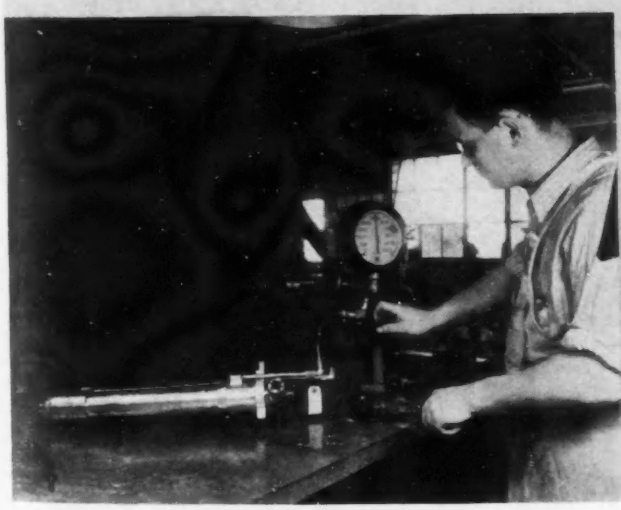


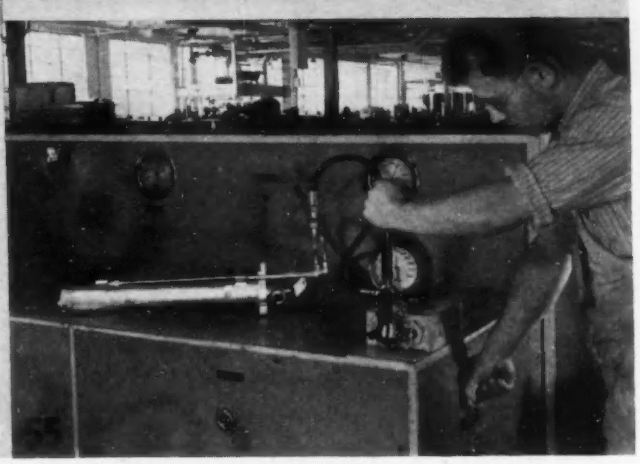








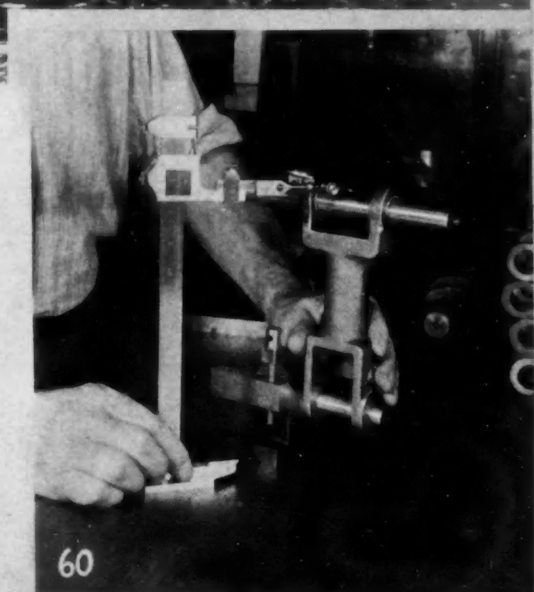


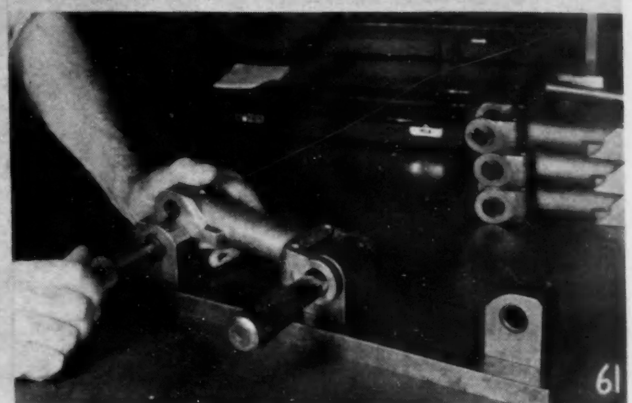


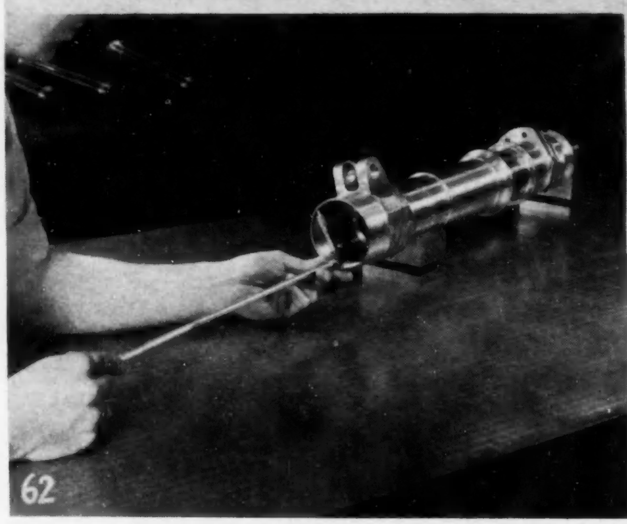
■ Figs. 51, 52 and 53 – The change from the manual measurement to the super micrometer method and the comparator method – Further evidence of improved inspection technique in aircraft

■ Figs. 54 and 55 – Old and new methods of testing hydraulic units

■ Figs. 56 and 57 – This is a rather vivid comparison of the difference in methods of checking thicknesses of aluminum-alloy sheets







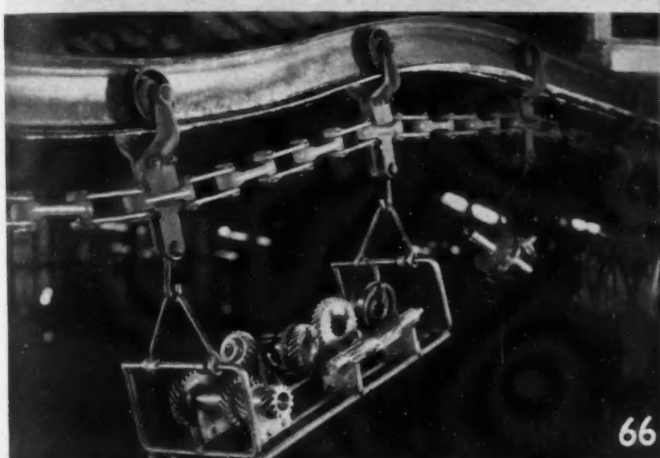
■ Figs. 58 and 59 – Change from caliper inspection to the more efficient and faster direct-reading dial methods

■ Figs. 60 and 61 – Excellent comparison between the layout type of inspection and the more efficient type of inspection fixture in which the entrance or non-entrance of checking plug gages represents acceptance or rejection

■ Figs. 62 and 63 – The measurement of internal bores is greatly facilitated by the direct-reading dial-type I.D. gage as compared with the conventional inside-micrometer type of measurement

■ ■ ■

Material-Handling Methods and Equipment (Figs. 64-68)



■ **Fig. 64 – An efficient plant requires a very comprehensive layout of machine-tooling locations and the routing of material in order that efficient and rapid material handling is possible. Conveyor systems are material-handling equipmənt**

■ **Figs. 65 and 66 – Both the roller and overhead types of conveyors are effectively used in automobile production**

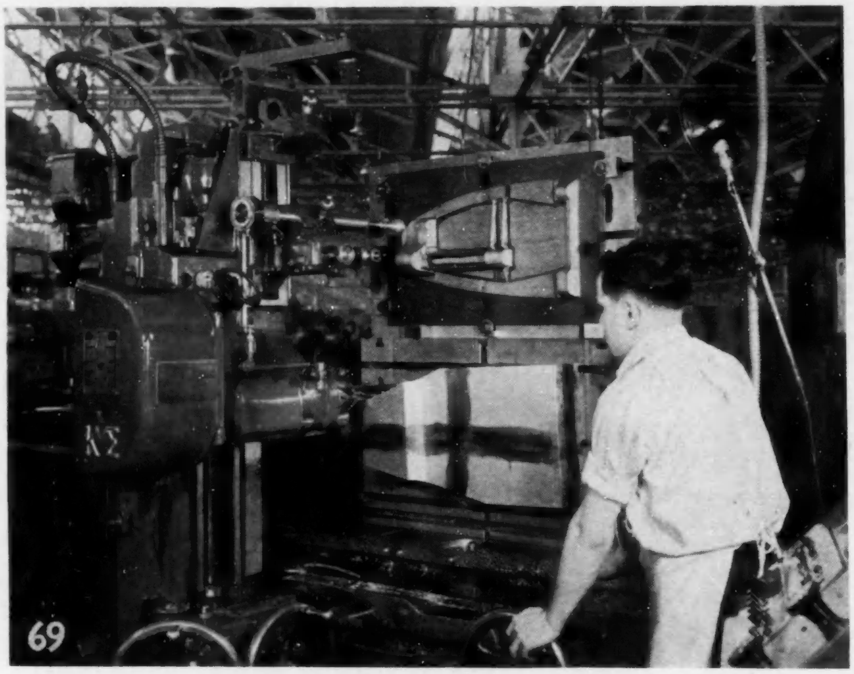
■ **Fig. 68 - Roller-conveyor system for handling landing gears**

This again has the advantage of material handling and the pull system for material supplies

■ **Fig. 67 – Conveyor system as applied to fuselage and landing-gear production**

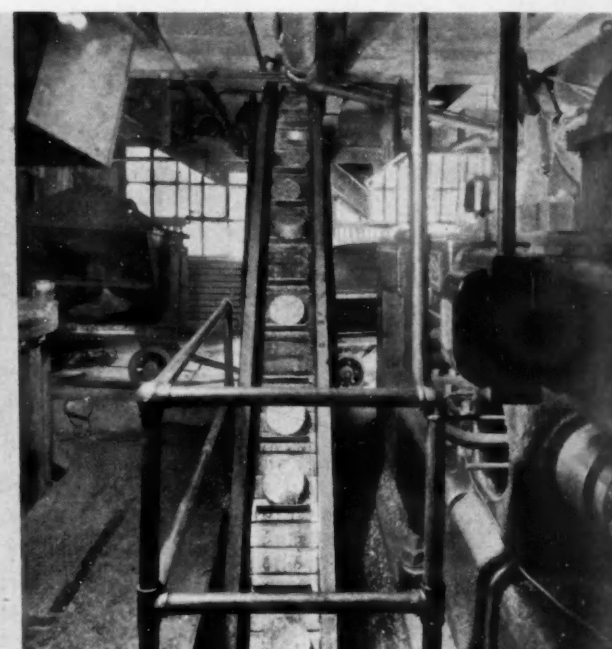
The fuselage must be designed so to permit maximum accessibility and concentration of man-power so that such an assembly line is possible. It is to be considered, however, that this type of design is also essential in small quantities, and if considered in such quantities, it is of great assistance in the event of large production

The conveyors in this figure are moved manually at regular intervals; the cart in the center is loaded with enough material to make one fuselage; and represents the second advantage of a conveyor system, that is, that it causes a pull system of material instead of a push system. The feeder departments are forced adequately to supply the assembly line to prevent a shutdown since a shortage of material on such a line causes a complete shutdown of the entire line, and does not permit shifting of man-power, so material supplies are forced automatically to be on time. The first advantage is, of course, that of material handling



■ Fig. 69 – The Keller method of making experimental forgings for prototype airplanes represents a logical step in production engineering of this type

Scrap Handling and Utilization (Figs. 70-75)



■ Figs. 70 and 71 – The problem of scrap material (not scrap due to defective workmanship but scrap resulting from clippings and wastage due to trimming and other conditions) demands recognition

Records indicate that a high percentage of all aluminum alloy used in the construction of an airplane becomes unusable clippings. In automobile manufacture, naturally, it is possible, because of the exclusive use of high-production dies, to plan punch-press operations and nest blanks more efficiently. Consequently, scrap is very low. In addition, every effort is made to obtain the highest return on the material by properly bailing such scrap in order to obtain the highest scrap value. The bailing of sheet stock and the briqueting of turnings and borings increase the tonnage scrap price and also facilitate the handling problem

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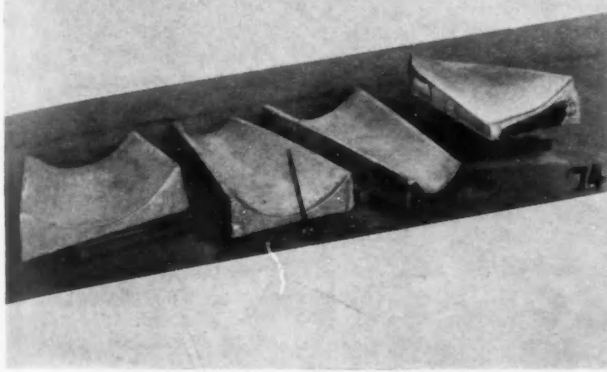

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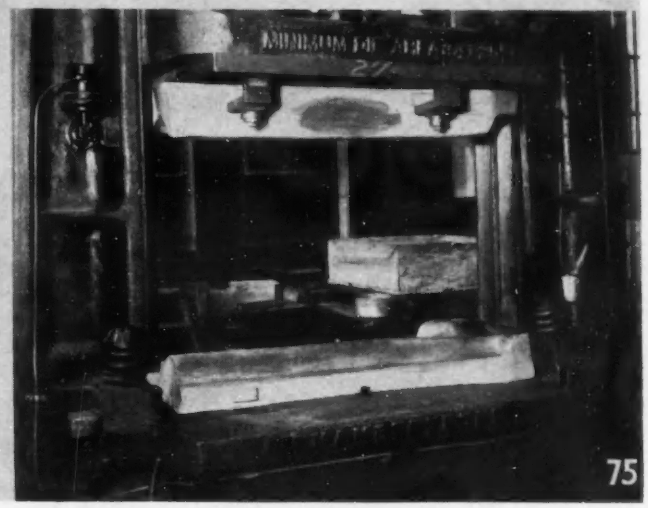


75

■ **Figs. 72, 73, 74 and 75 – Use of aluminum-alloy clippings as a source of drop-hammer material**

This use of aluminum alloy for drop-hammer dies (and for jigs and fixtures) has replaced completely the use of zinc and lead punches and dies. The change from aluminum to zinc was made without any new foundry equipment. Chills, however, are used in the molds. These chills increase the hardness of the dies, eliminate blowholes, and so on. Another feature of the aluminum die is that it weighs approximately one-third as much as zinc, is easy to handle, increases the output, for certain parts, from approximately 80 to 500 without rework, or deformation of the punch. The aluminum punch is poured directly into the die on which the face has been previously whitewashed

Another feature of drop-hammer work in this connection is the use of a drop hammer for tryout before releasing the die to production. This eliminates very costly trial set-ups and hold-ups in equipment, and permits the adjustment and fitting of punches and dies without interference with regular production activity

■ **Fig. 76 – The consideration of cost in aircraft manufacture is not as greatly advanced as it is in automobiles**

Cost control is the applied art of intercepting costs before the money is spent. To do this, it is necessary to know manufacturing costs, material - handling costs, prototype costs, and so on. Therefore, in the estimating of an experimental or production airplane, it must be borne in mind that the percentage of hand labor and the percentage of machine work on any given unit varies from department to department. Also, that the burden rate or overhead is not the same for hand labor as it is for machine labor. Therefore, this figure is shown as a tentative suggestion for arriving at a reasonably accurate production cost

UNIT	WEIGHT EMPTY	HAND LABOR	BURDEN	MACHINE WORK	BURDEN	TOOLS	MATERIAL	COMPLEXITY FACTOR	TOTAL COST	COST PER LB.
PANEL										
FUSELAGE										
TAIL SURFACE										
LANDING GEAR										
POWER PLANT										
FIXED EQUIPMENT										
MISCELLANEOUS										
TOTALS										

NEW AIRPLANE DESIGN

PROPOSED COST ESTIMATING PROCEDURE

Fig. 76

Characteristics of EXHAUST-GAS A

CONFLICTING views of experienced maintenance men on the accuracy of exhaust-gas analyzers in determining air-fuel ratio, Mr. Dilworth explains, stimulated a critical study and numerous actual tests of various exhaust-gas analyzers with a view toward securing some dependable information on the actual performance of these instruments.

The tests were all conducted at Virginia Polytechnic Institute on a single-cylinder, variable-compression test engine in order to eliminate, or permit the control of, as many variables as possible. The laboratory set-up and the method of conducting the tests are explained in this paper.

Six exhaust-gas analyzers were tested: two of the widely used thermal-conductivity type, one of the hot-wire catalytic type, one of the relative-density type, and two employing the Orsat principle. Mr. Dilworth points out that, while these do not represent all the makes on the American market, they represent every general type which is applicable to automotive service.

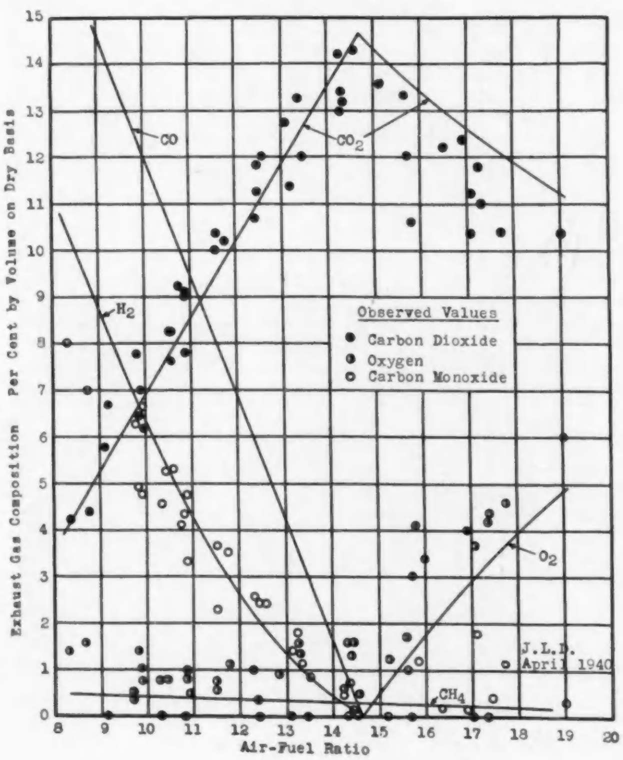
Probably the most striking thing shown, he reports, is that every instrument practically ceased to function when the air-fuel ratio became leaner than 14:1.

From the results of these tests it is concluded that exhaust-gas analyzers are not precision instruments, being likely to err as much as one-half of one air-fuel ratio, even under favorable conditions; and that all exhaust gas analyzers are calibrated for regular commercial gasolines, requiring special calibration for any fuel with a different chemical composition.

THE use of exhaust-gas analyzers as a motor tune-up aid has become steadily more popular during the past ten or fifteen years. A number of fleet supervisors and experienced maintainence men place considerable reliance on these devices, but there are also many well-informed operators who doubt the accuracy of this method of determining air-fuel ratio. With a view toward securing some dependable information on the actual performance of these instruments, the writer made a critical study and numerous actual tests of various makes and types of exhaust-gas analyzers.

These tests, which were performed in the automotive laboratory at the Virginia Polytechnic Institute, were all conducted on a single-cylinder, variable-compression test engine in order to eliminate, or permit the control of, as many variables as possible. The several gas analyzers were all connected to a manifold attached to the exhaust pipe of the engine. During each run the analyzer readings were noted as nearly simultaneously as possible, and the true air-fuel ratio was determined by accurately measuring the air and fuel supplied to the carburetor during the test period. Airflow was measured by means of a large domestic-type gas meter (calibrated against a Bureau of Standards certified gasometer), and fuel consumption was determined with the conventional fuel-measuring burette and stop watch. The air-fuel ratio was varied throughout the full operating range of the engine by adjusting the carburetor needle valve and, in this way, the accuracy of the gas analyzers could be checked over a wide range of mixtures.

While it is not necessary to go into the full details of the procedure at this time, it might be of interest to note some of the factors which could be controlled or varied at will. Beside such obvious items as speed, spark advance, and jacket-water temperature, the temperature and pressure of the exhaust and the temperature of the fuel-air mixture were subject to close regulation, thus eliminating many variables which might possibly affect the performance of the instruments, either directly or indirectly. The use of a single-cylinder engine also eliminated the chance



■ **Fig. 1 – Orsat analyses of various exhaust-gas samples compared with computed values**

Plotted points indicate experimental values; curves represent theoretical analyses (as determined by D'Alleva and Lovell[1])

ANALYZERS

by J. L. DILWORTH

Department of Mechanical Engineering, The Pennsylvania State College

of obtaining false readings caused by uneven distribution – a factor which may cause considerable error in any kind of exhaust analysis. A large exhaust pipe was utilized to aid in suppressing pulsations in the gas, but this expedient was not entirely successful.

Six instruments were tested: two of the thermal-conductivity type, one of the hot-wire catalytic type, one of the relative-density type, and two employing the Orsat principle. Although these instruments did not represent every make on the American market, they represented every general type which is adaptable to automotive service.

As an aid to understanding certain points which will be discussed later, it may be well to review briefly the operating principles employed in these kinds of instruments.

The thermal-conductivity method, which is by far the most widely used, is based on the ability of different gases to conduct heat away from a hot object (such as a wire) at different rates, depending upon the composition of the gas. For example, if electrical energy is supplied to a resistance coil at a constant rate, the temperature which the wire ultimately attains will depend upon the rate at which heat energy is carried away by conduction, convection, and radiation. Because heat lost by convection and radiation is reduced to a negligible value in gas analyzers, the maximum temperature attained by the wire depends primarily upon the conductivity of the surrounding medium. It only remains, then, to measure the temperature of the wire if we are to determine the relative conductivity of the enveloping gas, and this is what is done by the type of gas analyzer under discussion.

This principle would be of no value to the automotive engineer, however, if it were not for the fact that the composition, and thus the thermal conductivity, of the gases exhausted from a gasoline engine vary in a certain known manner with the richness of the mixture supplied to the cylinders. The relationship between exhaust-gas analysis and air-fuel ratio has been rather thoroughly investigated by a number of able experimenters and seems to hold true within fairly narrow limits under all engine operating conditions, provided ordinary gasoline is used as a fuel. The theoretical composition of the exhaust at various air-fuel ratios as determined by B. A. D'Alleva and W. G. Lovell[1] of the General Motors Research Laboratories is shown by the curves in Fig. 1. With these analyses as a basis, the theoretical thermal conductivity of the gases at various air-fuel ratios was computed. The results of these computations are shown graphically in Fig. 2.

The schematic diagram of Fig. 3 illustrates the basic components of a thermal-conductivity type of analyzer. Two gas cells, A and G, drilled in a block of metal, each contain a coil of resistance wire. These resistances, R_3 and R_4, are connected with two other resistance coils, R_1 and R_2, to form, together with a galvanometer, M, and a battery, B, an electrical circuit known as a Wheatstone bridge. When the resistance of R_1 equals that of R_2, and that of R_3 equals that of R_4, the bridge is balanced; that is, current flowing through the circuit is equally divided between the right- and left-hand sides and none flows through the galvanometers, thus causing no deflection of the needle. In the bridge employed in gas analyzers, the resistances are so proportioned that R_1 equals R_2, and R_3 equals R_4, when both resistances of a given pair are at the same temperature. It is a property of all conductors, however, that the electrical resistance will vary with the temperature. Hence, by measuring the resistance of a wire, the thermal conductivity of the enveloping gas may be determined.

Suppose that all of the four resistances are at the same temperature. Wires R_3 and R_4 are so designed that, when the current is turned on, they will attain equal temperatures of about 200 F if both are surrounded by air. However, the exhaust gas, which is led from the exhaust pipe of the engine to passage, P, and thence into gas cell G, may have a thermal conductivity greater or less than that of the air in cell A. This difference will cause heat to flow from coil R_4 more or less rapidly than from coil R_3. The temperature of R_4 will no longer be equal that of R_3;

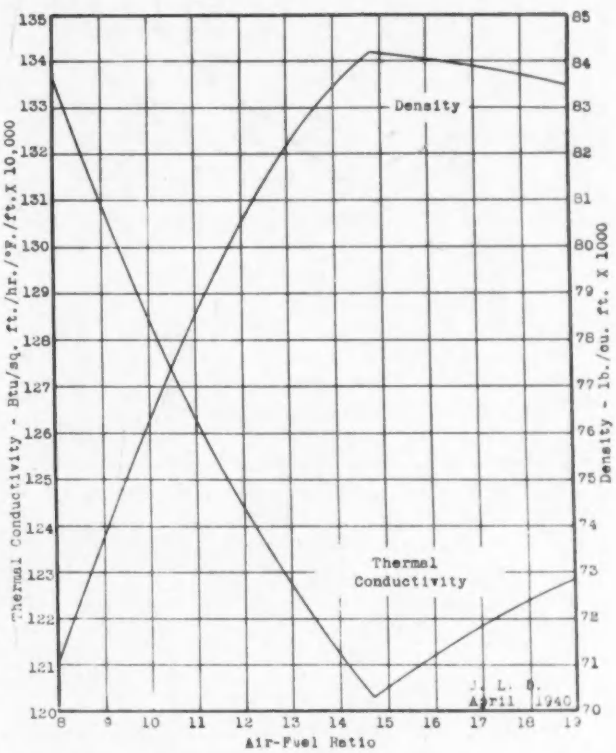
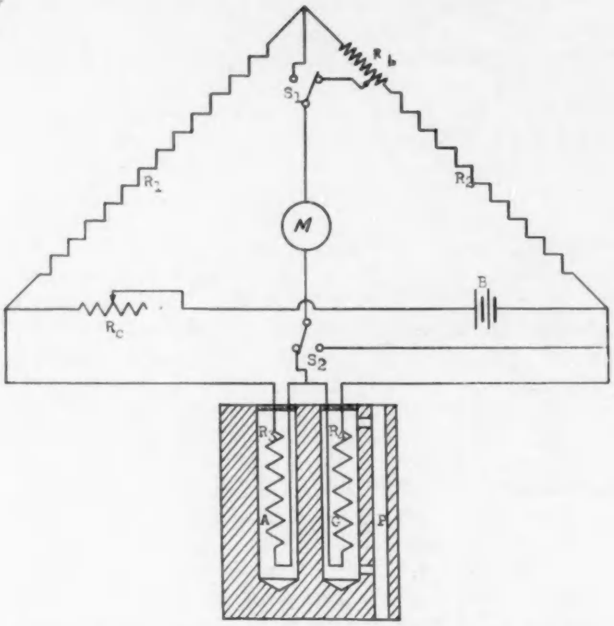


Fig. 2 – Computed thermal conductivity and density of exhaust gases at various air-fuel ratios

Based on theoretical exhaust-gas analysis as determined by D'Alleva and Lovell[1]; thermal conductivity values from International Critical Tables; all data for gases at 32 F and 14.7 lb per sq in. absolute

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 6, 1941.]

[1] See SAE Transactions, March, 1936, pp. 90-98, 116: "Relation of Exhaust-Gas Composition to Air-Fuel Ratio," by B. A. D'Alleva and W. G. Lovell; see also Engineering Experiment Station Bulletin Series No. 4, May, 1934, Oregon State Agricultural College, by Graf, Gleason, and Paul; see also NACA Technical Report No. 616, 1938; "Interrelation of Exhaust-Gas Constituents," by H. C. Gerrish and F. Voss.



■ **Fig. 3 – Circuit diagram for thermal-conductivity and catalytic types of exhaust-gas analyzers**

the bridge will therefore be unbalanced, and part of the current will flow through the galvanometer, *M,* causing the needle to move through an arc whose magnitude is related to the composition of the exhaust. By calibrating the galvanometer scale in terms of air-fuel ratio when the instrument is designed, it is possible to determine the mixture strength directly by means of the exhaust-gas analyzer.

Two other resistances and two two-way switches not heretofore mentioned are shown in the diagram. The purpose of rheostat R_c is to enable the operator to maintain a constant amperage in the circuit at all times, regardless of the condition of the battery. The second rheostat, R_b, is for the purpose of balancing the instrument (with air in both cells) in order to compensate for any change in resistance which might result from corrosion of the wires or formation of carbon deposits thereon. Both of these features are necessary for accurate work and are embodied in practically all analyzers of this type. The switches, S_1 and S_2, are necessary to permit adjustment of the current and balancing of the meter.

The catalytic type of gas analyzer is nearly identical in construction and operation with the thermal-conductivity type just described. However, whereas the operation of the latter type depends upon variations in thermal conductivity, the catalytic type is based on the principle of forced combustion of inflammable gases in the exhaust. By referring again to Fig. 1, it can be seen that the combustion of rich mixtures in an engine cylinder is incomplete and results in the liberation of varying amounts of hydrogen, carbon monoxide, and methane in the exhaust. These inflammable gases are, of course, not consumed in the cylinder because of insufficiency of air, lack of time, and inadequate mixing of air and fuel. Because of certain laws governing chemical equilibrium, the relative amounts of these several combustible components bear a rather definite relation to the ratio of air and fuel supplied.

The fundamental mechanical difference between the thermal-conductivity and the catalytic type of analyzer is that the wires in the analyzing cell of the latter operate at a red heat instead of at a relatively low temperature. In addition, these wires are made of platinum (which is also the case in certain conductivity instruments), a metal which has the property of causing or accelerating combustion which would not occur under normal conditions. When the exhaust gas, mixed with a small amount of air, passes through the analyzing cell, the catalytic property of the glowing platinum wire causes any combustible constituent in the gas to ignite and burn on the surface of the wire, thus increasing its temperature considerably. This rise in temperature is manifest by an increase in the resistance of the coil, which causes a deflection of the galvanometer as before. Since the amount of combustible material in the exhaust depends upon air-fuel ratio, the temperature rise of the wire is also proportional to this factor, thus permitting the instrument scale to be graduated accordingly. Temperature differences occasioned by variations in thermal conductivity are slight compared with those resulting from combustion; hence the effect of the former variable is negligible in this type of instrument.

It would seem to follow that the catalytic analyzer is capable of analyzing only mixtures richer than those which are theoretically correct, and such has been found to be the case.

In addition to thermal conductivity and combustible content, there is still another characteristic of engine exhaust gases which may be used successfully as an index of air-fuel ratio. This third property is density, and the manner in which it varies with air-fuel ratio is also shown in Fig. 2.

There are several methods available to the physicist for determining the density of gases but, for the most part, they require great skill and very delicate apparatus. However, a very ingenious method for determining the *relative* density of a gas (as compared with that of air) has been developed. The device employed is simple, automatic, and entirely suitable for commercial applications. Although necessarily a precision instrument, this apparatus is no more delicate than a number of other devices used by the modern tune-up mechanic.

The principle of the relative-density meter is illustrated in Fig. 4. It consists essentially of two cylindrical chambers, in each of which are located two fans or impellers. One of the fans in each housing is motor driven, and the opposing one is connected to an external system of levers in such a manner that its tendency to rotate is constrained within certain limits. Air is drawn into one of the chambers by the revolving impeller therein, and exhaust gas is similarly induced into the other. The impellers impart a swirling motion to the surrounding gas, and this effect tends to turn the semi-stationary wheels placed opposite them. The two rotating fans are so connected to the motor that they turn in opposite directions, thus exerting a clockwise torque on one of the opposing wheels and a counter-clockwise torque on the other. The stator wheels are connected to each other and to the pointer of the instrument in such a manner that they will turn through a small arc until a balance is reached, the equilibrium point being determined by the density of the exhaust gas in the one chamber as compared with that of the air in the other. Because density of the exhaust varies with the air-fuel ratio in a definite manner as previously shown, it is possible to graduate the scale of the instrument directly in

terms of air-fuel ratio as in the case of the other instruments.

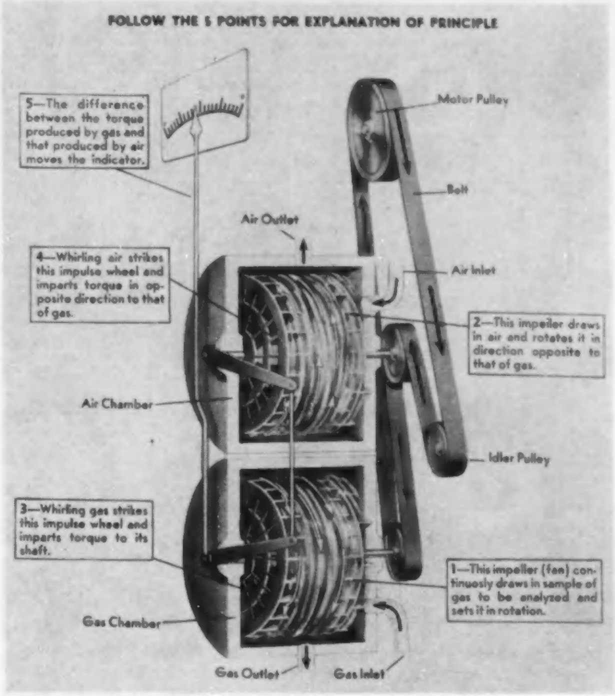
The Orsat apparatus is based on chemical principles. Since it is unsuited to automotive exhaust analysis except in laboratories, the theory will be reviewed here only in brief. Basically, the procedure consists of drawing a measured quantity of the exhaust gas into a graduated measuring burette. This measured sample is then transferred to a pipette where it is washed through a solution which has the property of absorbing carbon dioxide, but not oxygen or carbon monoxide. After the removal of the CO_2, the sample is returned to the measuring burette where the volume is again determined, the loss in volume on a percentage basis being the per cent CO_2 in the gas. The sample may then be washed in other solutions which permit determination of the per cent oxygen and the per cent carbon monoxide in a similar manner.

Figs. 5, 6, 7, and 8 show the readings of each instrument plotted against the corresponding actual air-fuel ratio supplied to the engine. The diagonal line in each case represents the locus of the points, had the instrument been correct. It will be noticed that two kinds of points are plotted; the heavy dots represent readings which were made with a fairly high exhaust pressure, and the circles indicate observations made with a low exhaust pressure[2]. The spark advance was also varied in some of these tests, but this had no effect on the readings of the analyzers. Hence, no attempt was made to differentiate between these runs.

Fig. 9 gives a rough comparison of the average accuracy of the several instruments. The curves portray the median of the points plotted on the four preceding graphs (for the higher exhaust pressure only; in no way do they indicate the "spread" of the readings of any analyzer).

Probably the most striking feature of all of these curves is the fact that every instrument practically ceased to function when the air-fuel ratio became leaner than about 14:1. This characteristic is to be expected, however, when it is remembered that the theoretical curves of both thermal conductivity and density broke sharply and reversed direction slightly at the chemically correct mixture ratio. It has already been mentioned that the theory of the catalytic type of analyzer would indicate the same limitation in its case.

This shortcoming of exhaust-gas analyzers is not as serious as might first be supposed, however, because it is unlikely that anyone would desire to operate an engine on an air-fuel ratio greater than 14:1. As most of us realize, excessively lean mixtures result in loss in economy as well as in performance, and may cause burning of the exhaust valves besides. This last point deserves considerable emphasis, however, since it is quite conceivable that some well-intentioned mechanic, in an effort to secure greater gasoline mileage, would rely too much upon his gas analyzer and set the carburetor to supply a dangerously lean mixture, completely unaware that the instrument will never read higher than about 14:1, even though the air-fuel ratio may be in the neighborhood of 16 or 17:1. Such a mistake would be entirely natural because most of these instruments have scales graduated well beyond their operating range. It is, therefore, important that anyone using this type of equipment be informed of its limitations and of the damage which might result from excessively lean mixture settings of the carburetor.

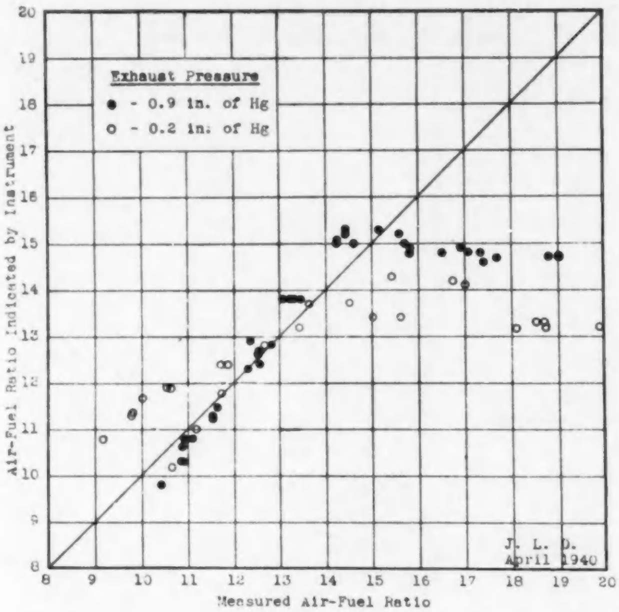


■ Fig. 4 – Operating principle of centrifugal type of relative-density exhaust-gas analyzer

The effect of changes in the exhaust pressure on the accuracy of exhaust-gas analyzers appears to be rather pronounced, as illustrated by the plots in Figs. 5 to 8. The pressure range used in these tests was probably greater than that which would be encountered in ordinary service but, even under normal conditions, considerable variations might be experienced. With the conventional method of inserting a sampling tube into the end of the exhaust pipe, the rate of gas flow through the analyzer will, of course, depend primarily upon the velocity of the gas in the exhaust pipe. Obviously, the velocity of the exhaust depends upon several factors, such as piston displacement, exhaust-pipe diameter, engine speed, and throttle opening.

In most cases, manufacturers of exhaust-gas analyzers have utilized various devices in an attempt to minimize the effect of rate of flow. A common practice is to connect the gas cell proper to the main gas passage by means of small holes through which, it is claimed, the gases diffuse into the gas cell at a fairly slow rate, regardless of the velocity of the gas through the main passage. To judge from the test results presented, this method is by no means entirely effective, at least not in all designs. Another method that is sometimes employed in an effort to provide a constant rate of flow through the instrument is to equip the sampling tube with a stopper or baffle which obstructs the exhaust pipe. This baffle incorporates a spring-loaded relief valve which permits the excess gas to escape, thus supposedly maintaining a constant pressure in the exhaust pipe and sampling tube. Unfortunately, so simple a device as this cannot possibly keep the pressure constant under all conditions. It is claimed that the relative-density type of analyzer, because it employs a fan to draw the sample into the instrument, is affected less by exhaust velocity than are those types which depend upon the exhaust pressure for obtaining a flow through the in-

[2] The upper row of points (large dots) in Fig. 8 is the result of a single series of runs which is believed to have been inaccurate. It seems probable that a little water was accidentally forced into the impeller chambers during this test, thereby causing the instrument to function improperly. Since no positive proof of this theory existed, however, these data were not thrown out.

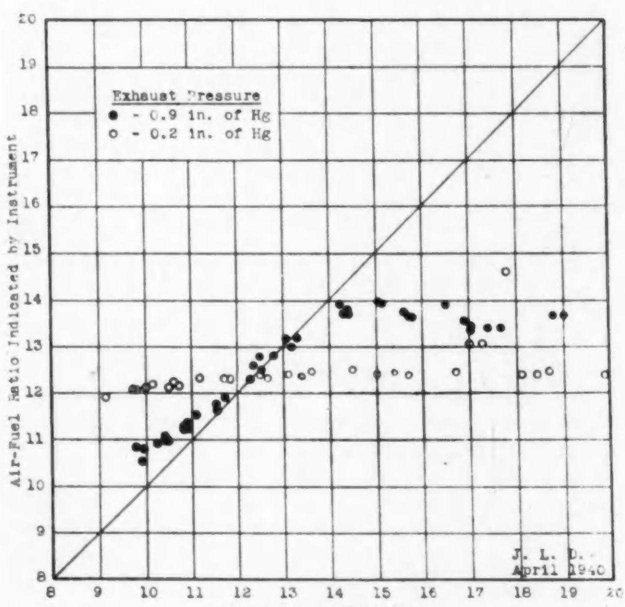


■ **Fig. 5 – Calibration curve of instrument *A***

Thermal-conductivity type of exhaust-gas analyzer; engine speed, 1400 rpm; compression ratio, 6.0:1; mixture temperature, 140 F; exhaust temperature, 130 F; spark advance varied from 15 to 40 deg BTDC; Esso gasoline

strument. This statement is supported somewhat by the results portrayed in Fig. 8, although even here some effect of pressure variation is noticeable.

Even under carefully controlled and steady conditions, the instruments tested did not give results which were altogether consistent. This shortcoming may be attributed to several possible causes. For one thing, the composition of the exhaust gases resulting from the combustion of a given air-fuel ratio may have varied somewhat. Certain analyzers seemed to be inherently unsteady in operation,

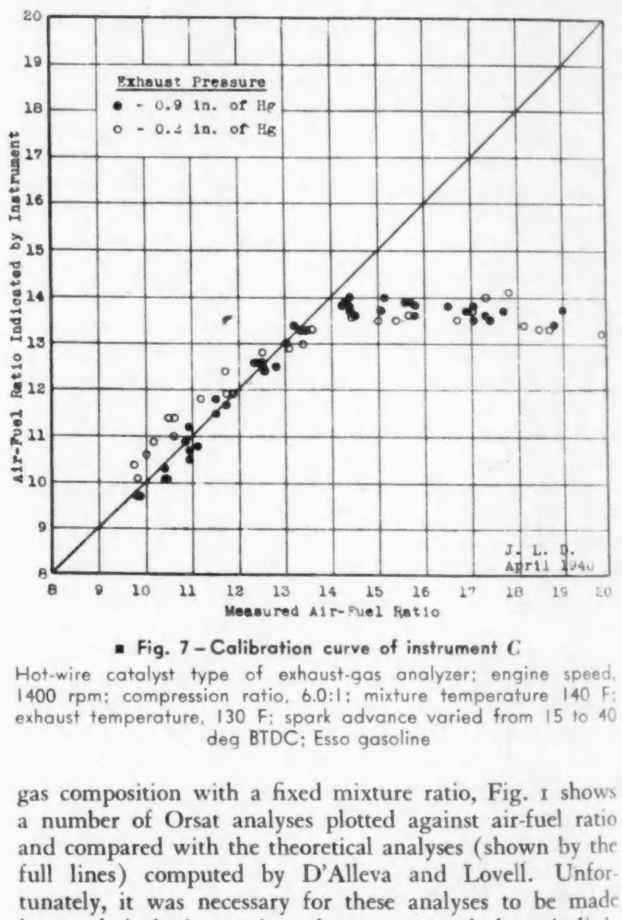


■ **Fig. 6 – Calibration curve of instrument *B***

Thermal-conductivity type of exhaust-gas analyzer; engine speed, 1400 rpm; compression ratio, 6.0:1; mixture temperature, 140 F; exhaust temperature, 130 F; spark advance varied from 15 to 40 deg BTDC; Esso gasoline

the needle fluctuating considerably even when all engine operating conditions were apparently constant, thus making a close reading unobtainable. It is possible that small droplets of water may have gotten into the analyzing cells and caused this erratic performance. In an attempt to preclude this possibility, however, the hoses were allowed to drain thoroughly before being connected to the analyzer, and each instrument was aspirated with air before a reading was taken. Another factor which might have contributed to the irregular readings of certain analyzers was the difficulty experienced in properly balancing the meter before making a reading. The usual procedure was to aspirate thoroughly the instrument with air, balance the meter, again flush with air, and recheck for balance. If a period of a minute or so were allowed to elapse following this operation, the needle would gradually creep away from the "balanced meter" index, making it impossible to adjust the instrument with certainty. This peculiarity was not characteristic of all analyzers, however.

Regarding the possibility of variations in the exhaust-

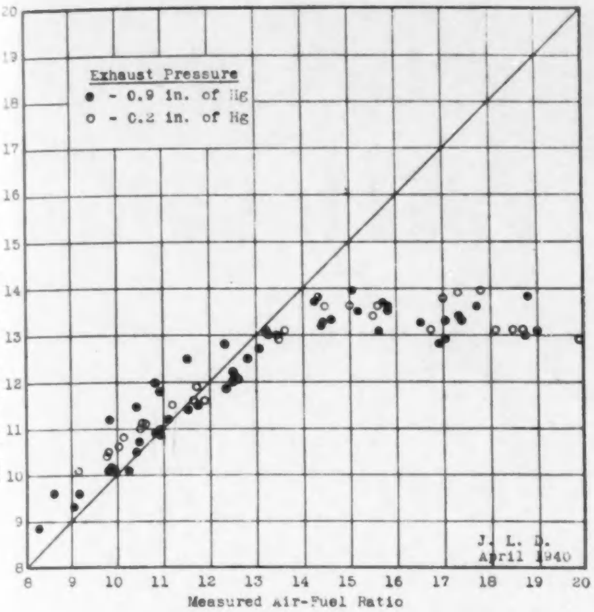


■ **Fig. 7 – Calibration curve of instrument *C***

Hot-wire catalyst type of exhaust-gas analyzer; engine speed, 1400 rpm; compression ratio, 6.0:1; mixture temperature 140 F; exhaust temperature, 130 F; spark advance varied from 15 to 40 deg BTDC; Esso gasoline

gas composition with a fixed mixture ratio, Fig. 1 shows a number of Orsat analyses plotted against air-fuel ratio and compared with the theoretical analyses (shown by the full lines) computed by D'Alleva and Lovell. Unfortunately, it was necessary for these analyses to be made by a relatively inexperienced operator, and there is little certainty as to their accuracy. However, the thermal conductivity and density of a number of actual samples obtained from different engines by the aforementioned engineers were computed, and the maximum variation for any given mixture strength was the equivalent of little more than one-tenth of one air-fuel ratio. This would indicate that variations in actual gas composition could account for only a slight error in instrument reading.

Little mention has been made thus far of chemical methods of analysis – the method that is used in all laboratories where precision is of paramount importance. The



■ **Fig. 8 – Calibration curve of instrument *D***

Relative-density type of exhaust-gas analyzer; engine speed, 1400 rpm; compression ratio, 6.0:1; mixture temperature, 140 F; exhaust temperature, 130 F; spark advance varied from 15 to 40 deg BTDC; Esso gasoline

conventional Orsat apparatus, briefly described in an earlier part of this paper, is slow, non-automatic, and intermittent in operation. In addition, it requires an experienced operator, troublesome cleaning of the pipettes, and renewal of the reagents. Also, it cannot feasibly be operated in moving cars or trucks. Although a valuable laboratory method, the Orsat is obviously not adapted to automotive service requirements.

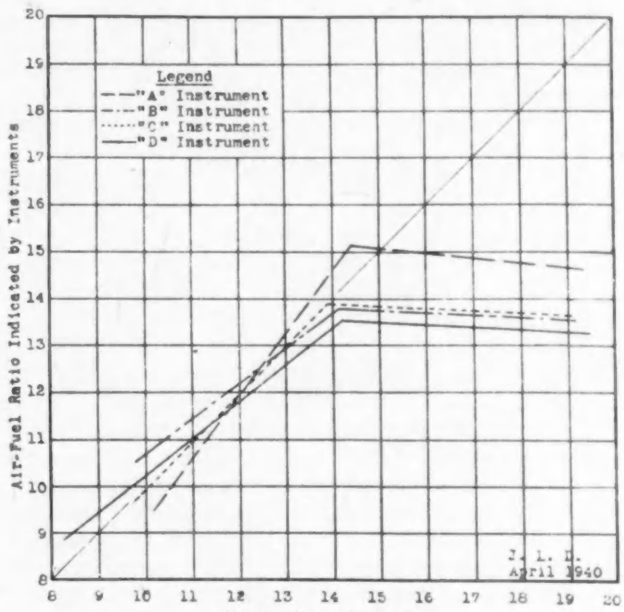
A semi-automatic instrument based on the Orsat principle has been developed which is claimed to be suitable for automotive work. This type of instrument can be used by an unskilled operator, but has several serious disadvantages. In the first place, it is intermittent in operation, each analysis requiring at least two minutes. Secondly, the apparatus is very sensitive to vibration and change of position, which renders it unsuitable for use on the road. Thirdly, it requires a periodic recharging with fresh chemicals, which is a somewhat troublesome task. As a fourth disadvantage, the scale is graduated in per cent, and a chart similar to Fig. 1 must be used to convert the readings to air-fuel ratio.[3] The automatic Orsat enjoys one big advantage over the other types of analyzers, however: it will operate equally well with rich or lean mixtures, provided it is of the two-unit type for determining both CO_2 and O_2 content. If it measures only the CO_2 concentration, it is obviously subject to the same limitations in this respect as the other types.

Generally speaking, the thermal conductivity and catalytic types are the lightest and most compact and, in addition, they enjoy the advantage of operating satisfactorily in any position. These types are also relatively unaffected by a reasonable amount of vibration and jolting, but some manufacturers take less pains to cushion the delicate galvanometer movement than do others. The relative-density instrument, while ruggedly built, is comparatively large and heavy. Certain meters of this type incorporate a humidifying chamber which is filled with water. Such an instrument must therefore be operated in an upright position and without undue bouncing if water is to be prevented from entering the impeller housings. Ordinarily, the motor in this analyzer requires more current than can be supplied successfully by dry cells, which necessitates the use of a 6-v storage battery if the instrument is to be used for road testing. If shop use only is contemplated, a motor will be supplied for 110 v a-c or other specified current and voltage.

All exhaust-gas analyzers are calibrated for regular commercial gasolines. Any fuel whose chemical composition differs appreciably from that of ordinary gasoline will necessitate a special calibration of the analyzer. While such fuels are not common at present, their use may become more widespread in the near future, a fact which users of these instruments should bear in mind.

It would appear from the results of these tests that, in general, exhaust-gas analyzers are not precision instruments, being likely to err as much as one-half of one air-fuel ratio, even under favorable conditions. It should be remembered, however, that, because the instruments tested do not represent all makes, by any means, it is possible that more accurate analyzers may be available. In the main, it is improbable that the average mechanic could secure better performance than was obtained under carefully controlled laboratory conditions, however.

The experimental portion of this investigation admittedly leaves much to be desired, and it is unfortunate that time and funds were not available for more extensive research. In spite of these imperfections, it is believed that the general conclusions derived from the tests are essentially correct and that they form, together with the various theoretical considerations presented, a reliable basis for appraising automotive exhaust-gas analyzers.



■ **Fig. 9 – Consolidated calibration curves for all instruments**

These curves represent the mean values taken from Figs. 5 through 8, inclusive, for observations made with 0.9 in. hg exhaust pressure, only

[3] A leading manufacturer of gas-analysis apparatus recently informed the writer that, as a result of these tests, modifications are being made in their future instruments of this type in order to rectify some of its objectionable features. The scale has been redesigned to read directly in terms of air-fuel ratio as well as in percentages. Other changes are claimed to speed the action of the instrument and eliminate creep of the pointer.

ALTITUDE CONDITIONING of Aircraft Cabins

by JAMES B. COOPER
Chief, Mechanical Equipment Unit, Boeing Aircraft Co.

HUMAN physiology and the inability of the human being to remain long at high altitudes without artificial provision of oxygen or pressure, or both, are emphasized in this paper.

The author reports experience with the Boeing "Stratoliners" now in regular commercial service. Ventilation standards, he explains, have permitted bleeding a minimum of cabin air and recirculating a considerable portion of the pressurized air. Rate of pressure change, he announces, has been found of considerable importance. Even at low altitudes where pressurizing is not essential to flight comfort, he reveals that passenger comfort is increased if pressurizing is used to reduce the rate of change in air pressure while climbing or descending.

The Boeing Stratoliner pressurizing system, and the test equipment with which it was developed is described in the latter part of this paper. Mr. Cooper also explains storm distribution geographically and by normal maximum altitudes.

MODERN aeronautic design now offers to both military and commercial operators the unrestricted highway of the substratosphere. Here weather conditions, except in rare instances, may be disregarded, and terrain clearance is no longer a problem. Engine supercharging, controllable propellers, and advances in aerodynamics enable us to take advantage of the rarefied atmosphere at the higher altitudes and thereby design the modern airplane to meet the demands for greater speed and increased range.

Although it is possible to take advantage of the upper air conditions in the aerodynamic design of the airplane, the human machine is so constructed that it operates properly only at the lower levels. Even persons in good health are unable to act efficiently when subjected to atmospheric conditions above 12,000 ft for extended periods, because of the lack of oxygen, and cannot remain conscious for any appreciable length of time at altitudes above 20,000 ft. Although the use of modern oxygen equipment will permit trained personnel to ascend to 35,000 or even 40,000 ft for short periods, the greatly reduced pressures often will produce severe pain or discomfort in the gastro-intestinal tract and, in many cases, too rapid decompression may result in the more serious effects of aero-embolism. The author recognizes the fact that proper de-nitrogenation, by breathing 100% oxygen while exercising for approximately one-half hour prior to an ascent, will prevent aero-embolism in nearly every case; this procedure, however, is not always possible. For example, unfavorable weather conditions might necessitate an ascent to high altitude after a flight previously scheduled for low altitude is well in progress. This method is, of course, entirely practical for scheduled high-altitude flights in military aircraft, but the use of oxygen for commercial operations has never been welcomed by the traveling public. The average airline traveler is noted for his ability to arrive at the airport only a few minutes before the time of departure, and might object to spending 30 min or more conditioning himself for a high-altitude flight.

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., Nov. 1, 1940.]

■ Pressure Cabin a Logical Solution

The most logical solution to the physical problems of high altitude operation is the sealed compartment or pressure cabin. By this means comfortable altitude pressures may be maintained within the cabin when flying at 20,000 or even 30,000 ft above sea level. The high degree of success already attained during actual service operation of the Boeing "Stratoliner" is sufficient to prove beyond any doubt the practicability of this type of aircraft.

In the design of the supercharged cabin it is necessary to consider only the variations in pressure with altitude and to disregard completely the differences in temperature, except in so far as the regular heating system is concerned. This is true because of the fact that normal body regulation maintains the air temperature within the lungs at a nearly constant value. Therefore, the true density altitude has very little bearing on the oxygen or pressure requirements of the body, and only the pressure altitude need be considered.

Based on the NACA Standards (Technical Report No. 218) the atmospheric pressure for any altitude up to the isothermal region may be determined by the equation:

$$P = 29.921\left(1 - 0.00689\frac{h}{1000}\right)^{5.256}$$

where P is the pressure in inches of mercury and h is the altitude in feet above sea level. A standard sea-level temperature of 59 F and a temperature gradient of 0.00357 F / ft up to 35,332 ft is assumed for the pressure calculations. Since the temperature in the stratosphere is assumed to remain constant at — 67 F, the pressure at any altitude above 35,332 ft is determined by the equation:

$$P = 29.921 \times 10^{(0.09759 - 0.000020742\,h)}$$

From this it may be seen that the atmospheric pressure variation with altitude is roughly 1.08 in. hg / 1000 ft at

■ Boeing Stratoliner in flight over piled-up cumulus clouds

sea level, 0.58 in. hg / 1000 ft at 20,000 ft, and 0.27 in. hg. / 1000 ft at 40,000 ft. Then, if the cabin pressure is maintained at a constant differential in relation to the atmospheric pressure surrounding the aircraft, it is possible for the airplane to climb or descend at a much higher rate than will be indicated within the pressure compartment, although the change in pressure inside and outside of the cabin will be the same. This is, of course, due to the fact that the barometric curve is steeper at the indicated cabin altitude than at the actual operating altitude.

"Overweather" operation, the primary reason for our desire to fly in the upper levels, has considerable bearing on the design of the supercharged airplane. The level of minimum frequency for various cloud formations, and the storm conditions prevailing over the area where the airplane is intended to operate, will, to a great extent, determine the normal operating altitudes of this type of aircraft.

Although much is being done in the way of air mass analysis, very little is known at this time in regard to weather conditions at the base of the stratosphere. However, by reference to meteorological charts, whereon cloud heights are plotted against average frequency, it will be seen that, during the summer months, cloud formations are at a minimum between the altitudes of 20,000 and 26,000 ft and between 16,000 and 22,000 ft in the winter. From this we may assume that an average operating altitude of 21,000 ft would be most satisfactory throughout the year.

In the continental United States thunderstorms which would affect long-range or coast-to-coast operations occur most frequently in the Central States from the East Coast to the Rocky Mountains and in the Mississippi Valley from Minnesota to the Gulf of Mexico. The average height of these storms is considered to be approximately 7000 ft although they frequently extend up to 15,000 ft. Some observations have been made of thunderheads whose tops reached to 35,000 to 40,000 ft; such heights, however, are considered extreme for this type of storm. Observers have reported that, while flying at altitudes between 20,000 and 30,000 ft, they were able to skirt occasional storms by deviating only 50 or 100 miles from their regular course.

During unfavorable weather conditions it is possible with supercharged cabin aircraft to maintain sufficient altitude to clear even the highest obstacles, such as mountain peaks, by a wide margin. In this way the last vestige of danger is removed from instrument flying, and a much greater percentage of operating schedules may be maintained, by cruising at an altitude of 15,000 ft or more, than is possible at the lower levels.

The growing demand for higher cruising speeds for both military and commercial airplanes may be met best by designing for operation in the substratosphere. According to Oswald[1], airplanes of modern aerodynamic design when equipped with turbo-centrifugal engine superchargers will permit an increase in operating speed of as much as 69% at 40,000 ft while the increase in operating costs would not exceed 6%. The Boeing "Stratoliner", equipped with two-speed geared engine superchargers, while operating at normal cruising speed (600 bhp / engine) with a gross weight loaded of 40,000 lb, attains a speed at 20,000 ft approximately 17% greater than that at sea level for the same conditions. At the same time, the cruising range is increased 6%.

If we could disregard the limitations of the human machine, operation in the substratosphere or even the stratosphere would be a comparatively simple problem. Medical science, however, after many years of painstaking

[1] See *Journal of the Aeronautical Sciences*, Vol. 2, January, 1935, pp. 1-9: "High-Altitude Flying and Its Effect on the Economics of Air Transport Operation," by W. Bailey Oswald.

research, has been able to lay down certain principles which may be followed in the design of supercharged cabin aircraft.

Of paramount importance are the oxygen requirements necessary to sustain life and to permit the brain and body to act efficiently. Paul Bert, in his studies on the effect of reduced atmospheric pressures, discovered that the amount of oxygen delivered into the blood stream depends directly upon the partial pressure of the oxygen contained in the lungs rather than the percentage of oxygen or the total pressure of the combined gases. Air at all altitudes up to 72,000 ft contains approximately 21% oxygen, 78% nitrogen, 0.03% carbon dioxide, and traces of other inert gases, while the alveolar air, because of respiratory action, is normally composed of about 15% oxygen and 6% carbon dioxide, the balance being made up of nitrogen, other inert gases, and water vapor. During ascent, the body automatically compensates to a certain extent for the reduction in partial pressure of the oxygen in the air taken into the lungs by an increase in the respiratory rate. In this manner the oxygen partial pressure is increased by a decrease in the carbon dioxide content in the alveolar air. Carbon dioxide, however, is one of the most important respiratory stimulants, and too great a reduction in the percentage of this gas may actually retard or even stop respiration.

Experience has shown that normal individuals require some additional oxygen if they remain at altitudes of 12,000 ft or more for extended periods and that, at altitudes above 20,000 ft, the breathing of only normal air will result in coma or possibly death in a relatively short period of time. Thus it is evident that, if long-range flights are contemplated at altitudes above 10,000 ft, some method of increasing the oxygen partial pressure must be employed. For this purpose we have available three alternate types of equipment. First and most common, particularly in the military service, is the oxygen mask which permits an increase in the amount of oxygen in the air delivered to the lungs. As mentioned before, this method permits operation at altitudes up to 40,000 ft for limited periods and is quite satisfactory for most military operations.

■ Limitations of Oxygen Mask

The use of the mask, however, is not entirely satisfactory for long-range flights, either military or commercial, since it is a source of discomfort to the wearer, requires a certain amount of his attention to avoid fouling of the tube and to keep the mask adjusted to his face, and limits his motions to a radius equal to the length of the tube. A further disadvantage is the weight of oxygen storage equipment that would be required on long flights.

A second method is the provision of an oxygen compartment wherein the amount of oxygen in the contained air is increased to the required minimum with an oxygen spray or similar means. It is obvious that ventilation by introducing fresh air into this type of compartment would be impracticable because of the waste of oxygen which would be carried off with the exhausted air. Therefore, it would be necessary to recirculate and "rejuvenate" the air within the compartment. The weight of the air-purifying equipment in addition to the oxygen-storage equipment would seem to overshadow the advantages of this system, although it would have great possibilities when operations are carried into the upper stratosphere, making completely sealed compartments necessary.

For operations between 10,000 and 40,000 ft or above, the presentday supercharged cabin appears to be superior to all other methods. By this means the percentage of oxygen in the air remains unchanged, while the oxygen partial pressure is increased by an overall increase in the compartment pressure. Therefore, occupants of the cabin are subjected to conditions exactly the same as though the airplane were being operated at an altitude equal to the pressure altitude maintained within the cabin. Any desired rate of ventilation may be maintained in the supercharged cabin by introducing temperature-conditioned fresh air through the supercharging system. The additional weight required for this type of equipment is entirely dependent upon the limits of operation desired, and experience to date indicates that the weight penalty is small except for extreme conditions.

A secondary advantage of the pressure cabin is the ability to regulate the rate of pressure change within the sealed compartment so that it is entirely independent of the rate of climb or descent of the airplane. This feature is most important for commercial operations, where inexperienced persons are carried, and is not necessarily limited to operations at high altitudes. In fact, the ability to regulate rates of pressure change is far more important at sea level than for any other condition. This is due to the fact that the variation in pressure with altitude decreases as the altitude is increased and, for a given rate of climb or descent, the rate of pressure change at sea level will be greater than at some higher altitude.

The physical aspects of changing pressures must be considered in two separate phases. First is the effect of pressure variations on the middle ear. The other is aeroembolism, which is similar to the "bends" commonly associated with deep-sea diving.

■ Effect on the Ear

Every air traveler is fully acquainted with the discomforting and sometimes painful effects of rapid climbs or descents on the passages of the ear. This condition may be understood best by a review of the construction of the middle ear, which is an air-filled cavity vented through the Eustachian tube to the nasopharynx, located just above and behind the soft palate. A decrease in atmospheric pressure such as would be experienced during a rapid climb will cause the pressure within the middle ear to be relieved through the Eustachian tube at regular intervals. Since this passage normally remains in a closed position, there is a slight deformation of the tympanic membrane prior to the release of pressure. Then, as the Eustachian tube opens, releasing the pressure, the ear drum rapidly returns to its normal position, producing an audible click. Armstrong[2] has found by actual test that, beginning at sea level, this cycle occurs first after an ascent of approximately 500 ft, then every 425 ft for all ordinary rates of climb. However, during an extremely rapid decrease of pressure such as would be experienced if the pressure in a sealed chamber were suddenly released, the Eustachian tube remains open throughout the initial decompression period. With the exception of those persons suffering from severe colds or an infection of the ear, nearly everyone who has been subjected to an "explosive" pressure release finds it less annoying than the more gradual change. This is probably due to the fact that the steady release of pressure

[2] See "Principles and Practice of Aviation Medicine," by Capt. Harry G. Armstrong, William Wood & Co., Baltimore, Md., 1939.

through the Eustachian tube, prevents the building up of sufficient pressure to cause the stuffy or full feeling in the middle ear, and eliminates the usual deflections of the ear drum which produce the sometimes annoying audible click at regular intervals.

From the foregoing, we may assume that a decreasing pressure, either gradual or rapid, is more a problem of annoyance rather than any physical danger. On the other hand, increasing pressures such as experienced during a descent may produce serious physical reactions which must be considered in the operation of any aircraft.

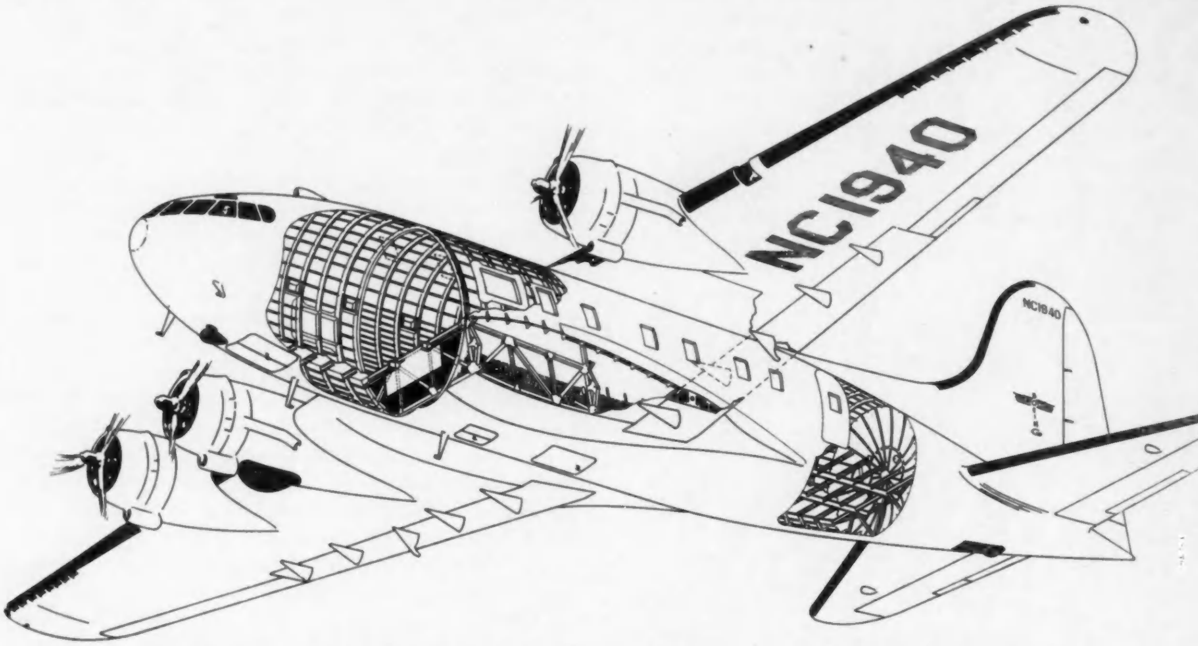
Ventilation of the middle ear during a descent is much more involved than the opposite condition. If the change is gradual, such as would be experienced while riding in an automobile down a slight grade, the pressure equalization is accomplished more or less without conscious effort. This is due to the fact that air may seep through the Eustachian tube when the pressure difference is quite low. At all higher rates of pressure change or if the passages are swollen by an infection, the Eustachian tube, already closed, tends to collapse further with increasing pressure difference, thereby preventing any equalization of pressure between the atmosphere and the middle ear cavity. This pressure difference may be relieved normally by voluntary action of the Eustachian muscles. This is usually accomplished by swallowing, yawning, or other facial contortions, except when the tube is blocked by an infection, such as a cold, or in the event that the pressure difference is allowed to exceed certain limits before any attempt is made to equalize the pressure. The maximum pressure at which the Eustachian muscles are capable of opening the tube by voluntary action is between 3.1 and 3.5 in. hg. Although experience has shown that the average person can ventilate the middle ear properly at rates of descent as great as 1000 fpm, airlines have established a practice of limiting descents to 300 or 400 fpm for the convenience of inexperienced passengers or persons suffering from colds.

Aero-embolism, caused by the release of dissolved nitrogen into the bloodstream, occurs when the partial pressure of the nitrogen dissolved in the body tissues and the blood is approximately doubled after a rapid reduction of total gas pressure to a value less than half that at which the body has become acclimated. During a steady climb from sea level the release of nitrogen begins at about 18,000 ft, or one-half an atmosphere and, because of normal elimination through the respiratory system, does not usually become dangerous until an altitude of approximately 30,000 ft is reached. The critical altitude at which the effects of aero-embolism become dangerous, however, varies with the absolute pressure at which the body has become saturated with nitrogen before starting the climb, the rate of pressure decrease, the altitude at the end of the climb, and the physical condition of the person involved. In general, any abnormal rate of ascent which results in a final pressure considerably less than half of the original or starting pressure might be considered dangerous. For the purpose of comparison we might assume an unsupercharged cabin airplane starting at sea level (29.92 in. hg absolute pressure) and climbing to 30,000 ft (8.88 in. hg) at an average rate of climb of 500 fpm, and a supercharged cabin aircraft designed to maintain a 12,000-ft (19.03 in. hg) altitude pressure within the cabin at 30,000 ft, climbing from sea level to 30,000 ft at the same rate. In the former case, the personnel on board would be subjected to a pressure change of 29.92 — 8.88 or 21.04 in. hg (a final pressure less than 30% of the starting pressure) over a period of 60 min. The change in the latter case would be only 29.92 — 19.03 or 10.89 in. hg (a final pressure of more than 60% of the starting pressure) in the same period of time. Then, assuming the remote possibility of a complete failure of the supercharging system with an average rate of cabin leakage which would produce an indicated rate of climb of 1000 fpm within the cabin, the personnel would be subjected to a pressure change of 19.03 — 8.88 or 10.15 in. hg (a final pressure nearly 47% of the starting pressure) over a period of 18 min, provided no attempt is made to descend to a lower altitude. This would allow ample time to don and adjust emergency oxygen equipment. The very remote possibility of a failure of the cabin structure, resulting in a sudden release of the cabin pressure, would have no serious consequences unless the pressure drop resulted in a final pressure considerably less than half of the starting pressure during operation of the airplane under conditions which would prevent a rapid descent to an altitude where oxygen could be administered conveniently. Although very little is known of the effect of sudden pressure releases at altitudes above 30,000 ft, numerous tests have been conducted at altitudes below this level with very satisfactory results. Armstrong[2] has shown that, because of the lag in the formation of nitrogen bubbles, a sudden release to altitude pressures between 30,000 and 40,000 ft would be dangerous only when neither an immediate administration of oxygen nor a rapid descent to an altitude sufficiently low to prevent anoxia could be effected.

From the foregoing, it is obvious that the ability to regulate the rate of pressure change in a supercharged cabin is of considerable advantage in that much greater rates of descent are permissible without inconveniencing the passengers. Comparatively high rates of ascent necessary to climb above storms may also be permitted with much less danger of any person suffering from aero-embolism.

Types of Cabin Superchargers

No attempt is made in this discussion to analyze the relative merits of the various types of cabin superchargers, although some mention should be made of available equipment, of which the Roots blower and the centrifugal compressor are most suitable. For either type the source of power may be either the main engines or an auxiliary powerplant. The proper choice of a blower and power source is entirely dependent upon the particular requirements for the aircraft in question. If the airplane is not normally equipped with an auxiliary powerplant and accommodations are available for driving the superchargers from the main engines, this type of installation is usually favored. The superchargers, when driven from the main engines, may employ either a single or two-speed drive, with or without declutching control, where the impeller speed is directly proportional to the engine speed unless it is disengaged. Variable-speed drives, which permit the impeller to be operated at speeds more or less independent of the engine speed, are now in the development stage and will soon be available for military and commercial aircraft. The type of supercharger and drive combination most suitable for a particular installation will depend, in general, on the engine speed range, the cabin pressure requirements, and the airflow required either for ventilation or to offset



■ Cutaway view of the Boeing Model 307-B, showing the main structural features of the pressure cabin

cabin leakage. Temperature rise through the blower also must be considered and may influence the selection to some extent. If the pressure and flow requirements are extreme, it will be necessary to provide coolers to reduce the temperature of the air before it enters the cabin, while in some cases auxiliary heating equipment will be required. By way of comparison, one supercharger installation was found to have as much as 200 F temperature rise through the blower during take-off and only 60 F rise at 20,000 ft. Preliminary estimates for another installation with considerably higher cabin pressures, indicated the maximum rise at sea level to be 10 F and as much as 300 F rise at 30,000 ft. In the case of the former, a single-speed supercharger was used and the blower discharge air bypassed at low altitudes. The estimates in the second case were based on a single-stage supercharger with a variable-speed drive.

The problem of ventilation with a supercharged cabin is somewhat more complicated than with the standard type of aircraft in that air flow may be governed by the power available to drive the blower, by the ventilating requirements for the passengers and crew, or by the rate of cabin leakage. As a general rule, the total flow will depend on the ventilation requirements for large commercial airplanes, and be governed by the rate of cabin leakage for military or small commercial aircraft. Although no set rule can apply for the amount of cabin leakage to be expected in any new design, one important fact has been gleaned from the multitude of tests conducted with the Boeing "Stratoliners," and that is, after all seams, joints, and door seals have been inspected and the leakage brought to a minimum, the variation in leakage with pressure change follows the curve for a standard orifice. The only way in which we may account for this is that, as door and window seals are made more effective by increasing pressure, the skin seams and other joints become less effective because of normal distortion.

An analysis of the preceding discussion indicates that the ideal supercharged cabin aircraft should be designed for normal operation between 15,000 and 25,000 ft to avoid a major percentage of adverse weather conditions, or to provide ample terrain clearance in the event that instrument flying is necessary. Within this range the cabin pressure should be maintained at a value not exceeding the equivalent of 8,000 to 10,000 ft for maximum passenger comfort and safety. Provision should also be made for emergency operation at altitudes up to 30,000 or 35,000 ft with a cabin "altitude" of approximately 12,000 ft. Commercial aircraft should also be equipped to permit regulation of the rate of pressure change within the cabin during rapid descents prior to landing. Before definite conditions can be established for any particular aircraft, many other factors must be considered, and the final design will depend to a great extent on the economics of operation.

During preliminary investigations leading up to the design of the Boeing "Stratoliner," a thorough review was made of all preceding works, and the results of laboratory and flight tests conducted by both the Army and the airlines were analyzed carefully. It was found that an operating altitude of approximately 20,000 ft would permit "overweather" operation on more than 90% of the proposed schedules, while only a slightly greater advantage could be gained by operation at 30,000 ft or above. The lower limit of supercharging was determined by the fact that this airplane would be operated out of airports as high as 7000 ft above sea level. Since it would be undesirable to rapidly increase or decrease the cabin pressure prior to take-off or after landing at the higher levels, it was decided to begin supercharging at approximately 8000 ft. Another factor which influenced this decision was the possibility of excessive temperature rise through the cabin superchargers at low altitudes during the summer months. This altitude therefore appeared to be the most satisfactory compromise for maximum passenger comfort and safety.

In the interests of economy it is of course desirable to

limit the differential pressure between the cabin and the atmosphere to a minimum. Therefore, on the assumption that operation at 20,000 ft would be required only during adverse weather conditions and could be compared to operations at 12,000 ft in an unsupercharged airplane, it was decided that an altitude pressure equal to that at 12,000 ft be maintained in the cabin at the upper limit of normal operation. The cabin differential pressure necessary to maintain an altitude pressure of 12,000 ft within the cabin at an operating altitude of 20,000 ft would be 19.03 — 13.75 or 5.28 in. hg which is approximately 2½ lb per sq in. Such a differential pressure is not at all unreasonable from a structural standpoint and therefore would not introduce any severe weight penalty.

Since this airplane would normally be operated between 8,000 and 15,000 ft, the limiting differential pressure of 2½ lb per sq in. would permit a constant pressure altitude of 8,000 ft to be maintained within the cabin throughout this range.

The production model of the Boeing "Stratoliner" is a four-engine, low-wing monoplane with accommodations for 33 passengers and a crew of 6. The wing has a span of 107 ft 3 in. while the overall length is 74 ft 4 in. Four Wright GR-1820-G102 or GR-1820-G105A engines provide a total of 4400 hp for take-off or 3600 hp at normal rated power. The standard gross weight of the Model S-307 is 41,000 lb, or 42,000 lb for the Model SA-307-B. Provisional gross weight for both models is 45,000 lb. The Model S-307 has a maximum speed of 236 mph at 7600 ft at standard gross weight, and cruises at 208 mph at 69.5% of power at 10,000 ft. The maximum speed of the Model SA-307-B is 246 mph at 17,300 ft and 238 mph at 6,700 ft at standard gross weight. Cruising velocity at 69.5% power at 15,700 ft is 220 mph.

The main cabin arrangements of both the Model S-307 and SA-307-B are such that nine reclining chairs are provided at the left side, while the right side is divided into four compartments, each equipped with two sets of triple seats. For sleeper operation each triple seat may be made up into an upper and lower berth. A dressing room, toilet, and galley are located in the aft part of the cabin. In the SA-307-B a second dressing room and toilet are located between the main cabin and the cockpit.

For maximum strength and economy the fuselage is circular in cross-section throughout. An ellipsoidal nose and conoidal rear section are faired into the cylindrical portion of the body which extends almost the entire length of the main passenger cabin. The pressure compartment, which is continuous from the nose to the pressure bulkhead located just forward of the horizontal stabilizer, is constructed with circumferentials, spaced on approximately 16-in. centers, longitudinal stringers, and a covering of 24ST alclad skin. Skin seams are sealed against internal pressure with 1/64 in. thick PAW tape inserted between the laps and are jointed with a double row of rivets diagonally spaced approximately ⅝ in. apart.

The windshield consists of six panes of laminated glass ⅝ in. thick, and eight panes of Plexiglas ⅜ in. thick arranged to provide adequate visibility in all attitudes of flight or landing. The 29 windows in the passenger compartment, toilets, and galley are ⅜ in. thick Plexiglas and are arranged to provide suitable vision in either the day or night arrangements. The main entry door, two cargo hatches, accessory compartment hatch, control cabin emergency exit, three passenger compartment emergency exits, and an access hatch in the pressure bulkhead all open inward and are provided with gaskets which seal under the influence of cabin pressure.

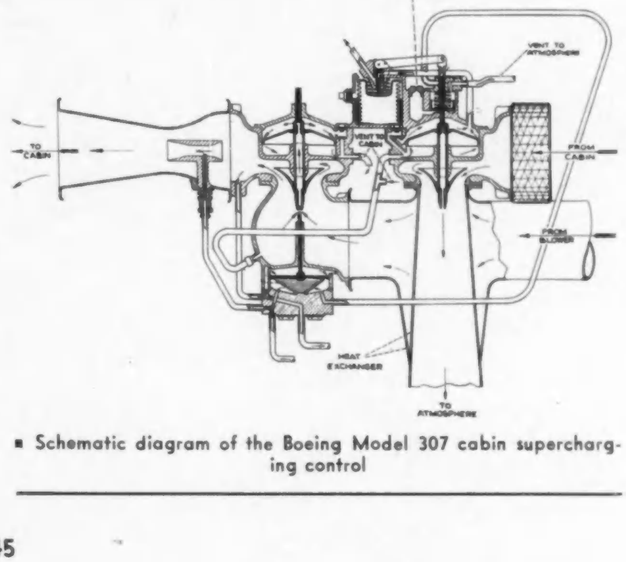
The entire pressure cabin structure is designed to withstand an internal pressure of 6 lb per sq in. and proof-tested to 3.75 lb per sq in., which is 150% of the normal operating pressure. Each body is leakage-tested with compressed air prior to final assembly of the airplane and again before delivery to the customer. The maximum allowable leakage rate of 150 cfm at 2½ lb per sq in. at sea level is based on the specified requirements that cabin leakage at 19,000 ft with both superchargers out of action should not produce a rate of pressure change within the cabin in excess of an amount equal to a 1000 fpm rate of climb at sea level.

Two General Electric, Type B-1, cabin superchargers are installed in the Boeing Models S-307 and SA-307-B airplanes, one in each inboard nacelle, and are driven from the main generator drive through a flexible shaft. These units are capable of delivering 225 cfm at 19,000 ft at 150 engine rpm while maintaining a cabin altitude pressure equivalent to 12,000 ft. No restriction is offered to the blower inlet flow, and regulation is accomplished by means of an automatic valve located in the blower discharge duct between the blower and the cabin duct system.

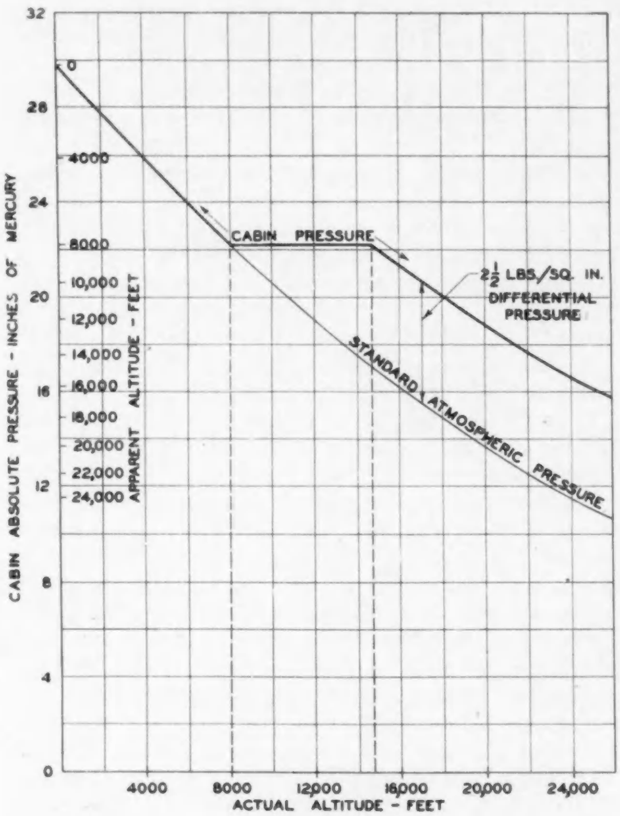
The entire cabin-supercharging control unit, which consists of the supercharger flow-control valve and the cabin pressure-regulating valve combined in a single compact unit to conserve weight and simplify installation, is located inside of the pressure compartment at the point where the blower discharge duct enters the body. This type of installation offered an excellent solution to the problem of defrosting the outlet diffuser in that the junction of the two ducts could be made in the form of a heat exchanger. In this way the warm incoming air is passed around the outlet diffuser, thus transferring a portion of its heat to the expanding exhaust air.

The supercharger flow control is entirely automatic and is operated by forces which it regulates. This unit is located in the transition between the blower discharge duct and the cabin distribution system. Regulation is by means of a poppet valve which offers a restriction to the flow of air from the blower in the amount necessary to limit the flow to a predetermined rate.

The pressure drop across the valve produces operating



■ **Schematic diagram of the Boeing Model 307 cabin supercharging control**



■ Boeing Model 307 cabin pressure events superimposed on the standard barometric curve

forces in an upward direction. These forces are balanced by opposing forces supplied by the difference in pressure between the blower discharge duct and the cabin, acting downward on an air-motor piston attached to the valve stem and having an area somewhat larger than that of the valve. The upper side of the air-motor piston is vented to blower pressure through a port in the valve stem, and control is accomplished by metering the flow of air through this port. The differential pressure between the inlet and the throat of a venturi located in the duct system is transmitted to the upper and lower sides of a flow piston in such a manner as to produce a downward acting force. This force is balanced by a spring acting upward. A metering pin attached to the flow piston is so located that its motion opens or closes the port connecting the upper side of the air motor piston to blower pressure.

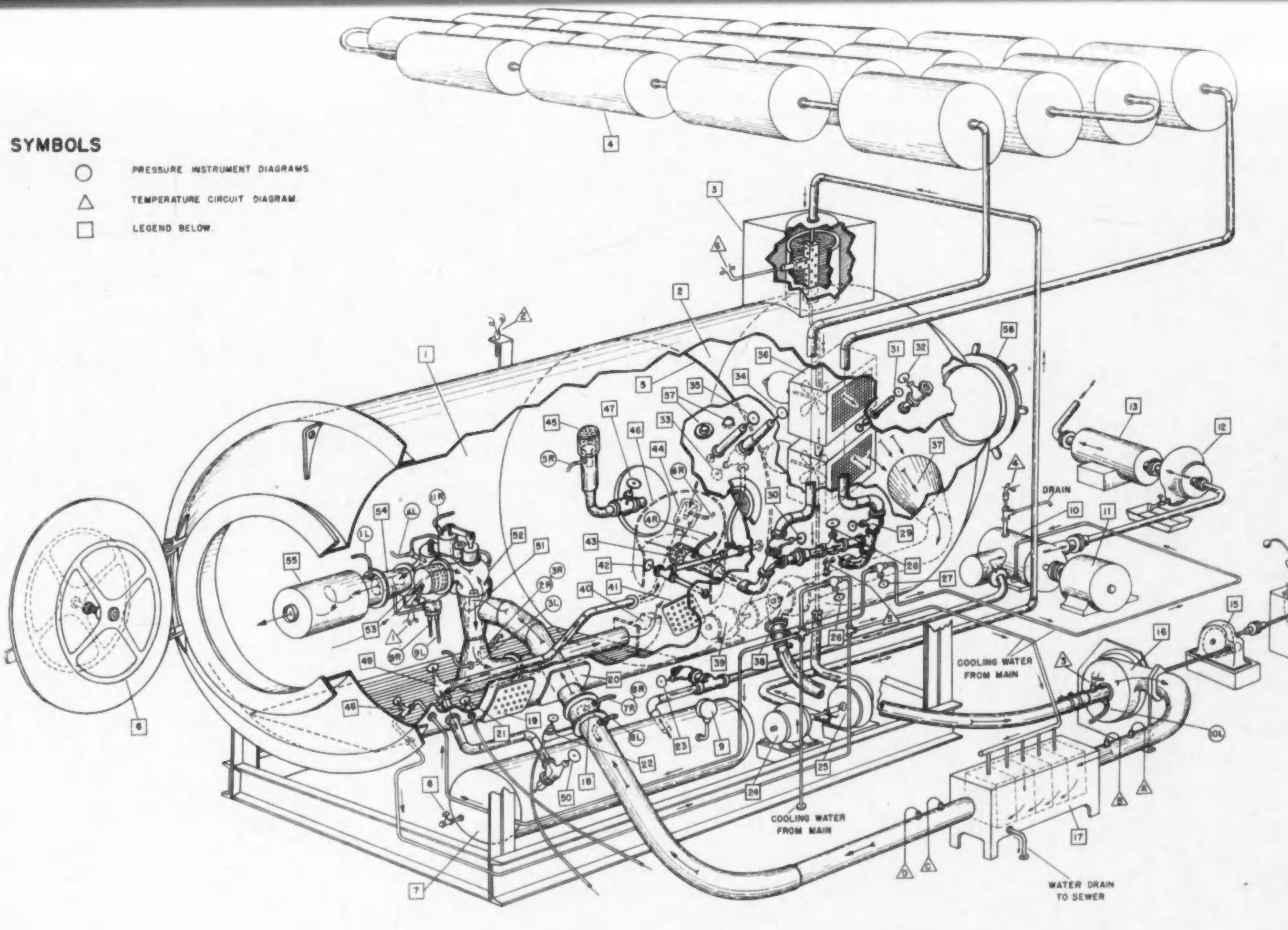
Any change in the pressure balance acting on the valve and piston assembly results in a change in the position of the valve. From this it is seen that an increase in the flow of air through the venturi will cause the flow piston and metering pin to move downward, opening the port to the upper side of the air-motor piston with the result that the valve will follow the motion of the metering pin and restrict the flow of air entering the cabin. For any decrease in flow, the reverse action will be true. Actually, the forces acting as just described will reach a balance with the force of the flow spring, thereby maintaining a constant rate of flow to the cabin.

This control, as installed in the Boeing "Stratoliner," is adjusted to maintain a constant-weight flow of approximately 18 lb per min within the capacity limits of the blower. A total of approximately 36 lb per min is supplied by the dual blower installation. An additional feature of this valve is that, in the event of failure of the blower or duct system, the valve will automatically close and prevent reverse flow.

Automatic regulation of cabin pressure is accomplished by restricting the discharge of ventilating air from the cabin to the atmosphere. The cabin pressure control consists of a valve and air-motor assembly similar to the inlet valve, an absolute pressure-sensitive element, and a differential pressure control. In this case, the pressure drop across the outlet valve produces downward acting forces which are balanced by the resultant force of cabin pressure acting upward on the under side of the air motor piston against a reduced pressure on the upper side. At altitudes below 8000 ft the upper side of the air-motor piston is vented directly to atmosphere and, since a small amount of pressure exists in the cabin because of the normal ventilating flow, the resulting forces hold the outlet valve in a full-open position. When an altitude of 8000 ft is reached, the vent between the upper side of the air-motor piston and the atmosphere is restricted by action of the absolute-pressure-sensitive element. Then, because of leakage between the piston and the cylinder walls, there is a tendency for the pressures on the upper and lower sides of the piston to equalize, with the result that the forces acting downward on the valve move it toward a closed position. The flow of ventilating air from the cabin to the atmosphere is, in this way, restricted. From this it is seen that regulation of the pressure on the upper side of the air-motor piston by metering the flow of air therefrom, will produce a balance between the forces acting upward on the piston and those acting downward on the valve, thereby maintaining a constant pressure in the cabin.

The absolute-pressure-sensitive element consists of an evacuated sylphon bellows which expands with decreasing pressure and actuates a metering valve located in the line connecting the upper side of the air-motor piston to the atmosphere. The bellows assembly includes a spring which resists the forces tending to collapse the bellows. This spring is adjusted so that the bellows begins to expand at an absolute pressure equal to that at 8000 ft. As the bellows expands, the metering valve is moved toward a closed position, thereby increasing the pressure acting on the upper side of the air-motor piston. From this it is seen that a reduction in cabin absolute pressure to an amount less than that at 8000 ft tends to restrict the discharge of cabin air until the cabin altitude pressure is returned to its normal value. Any increase in cabin altitude pressure to an amount greater than that at 8000 ft will have the opposite effect. The resulting action of this control is such that it tends to maintain cabin altitude pressure equal to that at 8000 ft at any altitude between 8000 ft and the ceiling of the airplane.

The cabin differential pressure control consists of a piston vented on the under side to cabin pressure and on the upper side to atmospheric pressure. The resulting upward force is counteracted by a spring whose force acts downward on the piston. The spring is adjusted to restrain motion of the piston until the limiting differential pressure of 2½ lb per sq in. is reached. This pressure difference normally occurs at an altitude of approximately 14,700 ft. Any further increase in this pressure which might occur at higher altitudes forces the piston upward against the spring pressure and engages a shoulder on the differential control plunger. This plunger normally closes a vent



LEGEND

1 - CABIN COMPARTMENT OF CHAMBER } B.A.C. 99-2225
2 - ATMOSPHERIC COMPARTMENT OF CHAMBER }
3 - COOLING LIQUID RESERVOIR
4 - AUXILIARY VOLUME DRUMS
5 - PRESSURE BULKHEAD
6 - ACCESS DOOR TO CABIN
7 - VACUUM TANK B.A.C. 1-17177
8 - VACUUM CONNECTOR VALVE (AUXILIARY)
9 - VACUUM GAGE
10 - VACUUM PUMP
11 - VACUUM " MOTOR
12 - OIL SEPARATOR
13 - EXHAUST SILENCER
14 - FORD ENGINE
15 - GEAR BOX (REVERSING) RATIO - 1 TO 1.545
16 - CABIN SUPERCHARGER (TYPE B 1) GENERAL ELECTRIC CO.
17 - INTERCOOLER B.A.C. 99-2572
18 - FLOW MEASURING NOZZLE - 2 IN. DIA.
19 - ADAPTER PLATE (PRESSURE INSTRUMENTS) B.A.C. 99-2308 #1
20 - ADAPTER PLATE (ELECTRICAL INSTRUMENTS) B.A.C. 99-2308 #2
21 - FORD ENGINE THROTTLE CONTROL (INSIDE)
22 - FORD " " " (OUTSIDE)
23 - VACUUM CONNECTOR VALVE (AUXILIARY)
24 - COOLING LIQUID PUMP MOTOR
25 - COOLING " "
26 - INTERCOOLER WATER REGULATOR VALVE (OUTSIDE)
27 - VACUUM PUMP WATER SHUT OFF "
28 - CABIN LEAKAGE ADJUSTING VALVE (OUTSIDE)
29 - ATMOSPHERIC AUXILIARY VOLUME SHUT OFF VALVE
30 - CABIN " " " " "
31 - ATMOSPHERIC COMPT. VAC. RELIEF VALVE
32 - ATMOSPHERIC " AUXILIARY "
33 - ATMOSPHERIC " BLEEDER " (INSIDE CONTROL)
34 - ATMOSPHERIC " " " (OUTSIDE ")
35 - CABIN " " "
36 - COOLING RADIATOR UNIT
37 - AUXILIARY BLOWER INLET DUCT
38 - AUX. BLOWER
39 - AUX. " MOTOR
40 - CABIN OUTLET DEFLECTOR
41 - PRESSURE BULKHEAD ADAPTER PLATE B.A.C. 99-2308 #3
42 - ATMOSPHERIC COMPARTMENT AUXILIARY VALVE
43 - CABIN LEAKAGE FLOW NOZZLE (OUTSIDE) 1.5 IN. DIA.
44 - VACUUM PUMP INLET NOZZLE .625 IN. DIA.
45 - CABIN LEAKAGE FLOW NOZZLE (INSIDE) 1.5 IN. DIA.
46 - WARM AIR BAFFLE
47 - CABIN LEAKAGE ADJUSTING VALVE (INSIDE)
48 - INTERCOOLER WATER REGULATOR VALVE (INSIDE)
49 - VACUUM PUMP INLET THROTTLING " "
50 - VACUUM " " " " (OUTSIDE)
51 - HEAT EXCHANGER B.A.C. 14-1670
52 - CABIN SUPERCHARGER CONTROL B.A.C. 56-5909
53 - CABIN OUTLET SCREEN SEE " " "
54 - INLET VENTURI " " " "
55 - CABIN INLET MUFFLER
56 - ACCESS DOOR TO ATMOSPHERIC COMPARTMENT
57 - UTILITY PLUG

■ Schematic diagram of the Boeing "Strato-Chamber" showing the equipment set up for a test of the Model 307 supercharging controls

through the valve stem between the upper side of the air-motor piston and the atmosphere and is so located that upward motion of the plunger opens this port, causing a decrease in the pressure on the upper side of the piston. This unbalances the forces acting on the piston and valve assembly and the valve moves toward a more open position, thus relieving excessive cabin pressures and overriding the absolute pressure control.

The apparent cabin altitude which the absolute pressure control maintains, and the limiting differential pressure, may be varied to suit any range of operating conditions by merely altering the bellows spring or the differential control spring or both. The flexibility of this type of control is limited only by the structural requirements of the airplane and the capacity of the blower.

Cabin absolute pressure in the Boeing Model 307 follows the standard barometric curve from sea level to 8000 ft, is isobaric from 8,000 to 14,700 ft, and approximately 5.1 in. hg in excess of atmospheric pressure at altitudes above 14,700 ft. See accompanying curve. This action has the effect of raising the barometric curve for altitudes above 14,700 ft as though an additive constant had been introduced into its equation.

Rates of pressure change within the supercharged cabin, as expressed by the ratio of

$$\frac{\text{Indicated cabin rate of climb}}{\text{Indicated airplane rate of climb}}$$

would have the following values for the conditions noted: unity from sea level to 8000 ft, zero between 8000 and 14,700 ft, and approximately 0.8 at all normal operating altitudes above 14,700 ft.

■ Instruments and Controls

In the Boeing "Stratoliner" the Flight Engineer is provided with a cabin supercharging instrument board and various manual controls. These instruments, located within a green border on the Flight Engineer's instrument panel, consist of a rate-of-climb indicator vented to cabin conditions, a cabin altimeter, blower pressure gage and selector valve, and a vacuum gage which, by means of a selector valve, may be used to indicate the flow-control venturi suction of either unit or to show the cabin differential pressure in relation to atmospheric pressure. A third selector valve located on this board permits manual selection of cabin pressure control by either or both of the duplicate installations. Manual control of the flow regulating valves is provided by levers located on the floor to the left of the Flight Engineer. Auxiliary ventilation controls regulating a fresh air inlet and a foul air outlet are also within reach of the Flight Engineer. This auxiliary ventilating system may be used at low altitudes during warm-weather operation.

Each supercharged airplane is tested thoroughly before delivery to the customer to insure the proper functioning of the entire supercharging system. This test consists of operation at various engine rpm during level flight, climbs, and descents, throughout the normal supercharging range.

Preliminary tests of the cabin supercharging system were conducted in the laboratory with the aid of a small pressure chamber. This chamber, however, did not permit operation at the actual pressures for which the control was designed. Subsequent tests therefore were conducted both during flight of the airplane and by means of a large "Strato-Chamber" especially designed for testing cabin supercharging equipment. (See accompanying illustration.)

The Boeing "Strato-Chamber" consists of a 3-ton steel tank 12 ft long by 5½ ft in diameter, fitted with pressure-tight doors at each end and divided into two compartments by a pressure-tight bulkhead. One compartment represents the cabin of the airplane while the other compartment represents the outside air. In this chamber observers can "fly" in a few minutes time to any desired altitude, either with comfortable supercharged air conditions or with oxygen equipment.

Complete instrument boards consisting of a bank of manometers, all the normal flight instruments necessary to indicate cabin and atmospheric altitudes, rates of climb or descent, engine rpm, and pertinent temperatures throughout the supercharging system, are located both outside and within the cabin chamber.

A high-capacity vacuum pump connected to the atmospheric chamber and regulated by valves is used to control the pressure within the chamber. The vacuum pump is capable of evacuating the chamber to a pressure equal to that at 40,000 ft or above, and at a rate equal to a rate of climb of more than 3000 fpm at 40,000 ft. A refrigeration system is provided to reduce the temperature in the atmospheric chamber to as low as 30 F or more below zero. During actual tests it has been possible to maintain standard density altitudes within the chamber during a simulated climb from sea level to 25,000 ft at a rate of climb of 1000 fpm.

Other equipment includes a dehydrator and steam jets for regulating the humidity, a small blower used to simulate ram pressure in the duct system during flight, and valves arranged to regulate leakage between the cabin and atmospheric chambers equivalent to the rate of cabin leakage which may be expected in the airplane.

During tests of a supercharging system the blower receives air from the atmospheric chamber and discharges to the cabin chamber. The blower is driven by an automobile powerplant which is mounted on a cradle and so arranged that it may be used as a dynamometer to measure horsepower absorbed by the blower. The cabin supercharging control unit is normally located in the cabin compartment in a manner similar to the installation in the airplane. In this way the air drawn from the atmospheric chamber by the supercharger and delivered to the cabin under pressure is returned to the atmospheric chamber through the regular cabin pressure controls and through the cabin leakage-regulating valves.

Complete controls for the powerplant, the vacuum pump, and the temperature-regulating apparatus are duplicated both outside and within the cabin chamber. Contact with operators inside of the chamber is maintained by observation windows and a telephone system.

Although this chamber has been used mainly for the adjustment and calibration of cabin pressure-control equipment, a great deal of work is being done at this time in connection with the advanced development of cabin supercharging and heating systems.

The Boeing "Stratoliners" are now being operated under actual service conditions both by Transcontinental & Western Air, and by Pan American Airways. This innovation in air transportation has been widely acclaimed both by the operators and by the traveling public, and much may be expected of the future of altitude conditioning of aircraft cabins.

Designing for ALTERNATE MATERIALS

by THOMAS A. BISSELL
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THIS paper was prepared in response to a request of the Passenger-Car Activity Committee of the Society for a report of the work done by engineers in automobile, equipment, and accessory companies, in substituting plentiful alternate materials for critical shortage materials urgently needed in defense. Most of the data presented were contributed by these engineers either in interviews or correspondence with the author. Since the alternate-materials picture is constantly changing, this paper can only present a "still"—as of May 1, 1941.

The major part of this paper reports the work done in replacing aluminum, nickel, zinc, and Neoprene—all on the OPM's mandatory priority list or under partial control by this body.

Superimposed on this activity, is the development of alternates for many imported "strategic" materials, such as chromium, manganese, rubber, tin, antimony, and cork. Objectives of these substitution programs are two: to cooperate in our defense effort by releasing large tonnages of critical materials to defense industries, and to guard against every material shortage, real or possible, that threatens to interrupt passenger-car production.

In his conclusion, the author points out that substitutions of alternate for critical materials, made or definitely planned to date, will necessitate no lowering of the standards of safety, durability, performance, and comfort established in the 1941 passenger cars; and that, in virtually all cases, parts of alternate materials cost more than do the original parts of critical materials.

DESIGN and development of automobile parts to accommodate plentiful alternate materials in place of critical materials started voluntarily long before acute shortages arose or mandatory priorities were invoked. From that time on, this work has been expanding until today it requires the full-time efforts of many engineers in automobile, equipment, and accessory plants. This substitution program has two important objectives:

1. To cooperate in our national defense effort by releasing large tonnages of critical materials, formerly required to build passenger cars, for use in defense industries.

2. To guard against every material shortage, *real or possible,* that threatens to interrupt passenger-car production (This also strengthens national defense, since millions are kept at work building products that have grown to be a vital part of our industrial and economic life and since it insures against loss of the substantial taxes paid to the Government by the industry and the motorist.)

Both objectives are being achieved by immediately replacing many parts of critical materials; the second also is being reached by developing parts of alternate materials and setting up stand-by equipment to produce these parts, in preparation for shortages which *may* develop.

Passenger-car executives and engineers agree, however, that there are certain basic materials used in automobiles

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 5, 1941.]

for which satisfactory alternates cannot be worked out. Most automobile engineers list as irreplaceable: steel, copper, rubber (natural or synthetic), lead, some kind of upholstery material and, possibly, glass. Serious shortages in any of these materials, they contend, would shut down our plants or curtail production to the levels common before mass production came into being.

Of the critical materials under industry-wide mandatory priorities or monthly rationing when this paper was prepared – aluminum, nickel, nickel-bearing steel, Neoprene, magnesium, and tungsten – only the first four are used in significant quantities in automobiles. In addition zinc, widely used in automobile die castings, was under partial control, requiring that zinc producers set aside for June, 1941, 22% of their April production for an emergency defense pool. Late in May, 1941, copper was placed under similar control, suppliers being required to pool 20% of their April production for June, 1941. For these reasons, virtually all substitution work that already has been done has been that of replacing parts of aluminum, nickel, nickel-bearing steel, Neoprene, and zinc with those of alternate materials. Superimposed on this activity, has been the development of alternates for many imported "strategic" materials – manganese, chromium, tin, antimony, rubber, and cork, the first four of which appeared on the OPM's "priority critical list" under inventory control as this paper went to press. With steadily darkening war clouds, this latter work recently has been expanded and accelerated.

Since new priorities are constantly being invoked or old ones relieved and new alternates are constantly superseding previous ones, it should be emphasized that, at best, this paper can be but a "still" of a rapidly moving picture. It, therefore, attempts to give a cross-section of the work on alternate materials as of May 1, 1941.

Replacing Aluminum

About half the passenger-car engines being produced in America today carry aluminum-alloy pistons. To guard against the possibility of drastic curtailment of all piston aluminum alloys, development of cast-iron or cast-steel substitutes either has been completed or is in process for the majority of these engines. This is the most important design problem to date in the entire substitution program of many companies because of the basic effect of the piston on the design of related engine parts.

The cast-iron pistons are approximately 50% heavier, have less than half the thermal conductivity, and about half the thermal expansion of the aluminum-alloy pistons which they are designed to replace. Increase in reciprocating weight means that piston pins, connecting rods, and connecting-rod bearings must be checked for strength and rigidity against the higher stresses to which they will be subjected. The cooling system must be checked against the lower thermal conductivity, and the piston-pin fit, against the lower thermal expansion. Often, too, the compression ratio must be lowered and the spark timing altered to suit the new piston set-up. The extent of the redesign necessary, of course, varies according to the design of the particular engine. One fortunate aspect of the problem has been the adoption on many 1941 models of thin-babbitt main and connecting-rod bearings of greatly increased fatigue life. In most cases, these bearings have adequate excess capacity to handle the higher stresses imposed by the heavier cast-iron piston assemblies.

Heavier, stiffer connecting rods have been found to be necessary on many engines, the added weight being employed to increase the section through the connecting-rod shank and the thickness of metal around the big end.

On one line of cars, approval has been obtained on a switch to a connecting rod about 5% heavier and to bearings with longer fatigue life on both connecting-rod and main bearings. These bearings are of the type, introduced on some 1941 passenger cars, that employ a porous nickel-bronze matrix to bond the thin babbitt lining to the steel backing. The program of this company calls for incorporating the heavier connecting rods and new bearings into the engine designs as soon as the stock of present parts is exhausted. Then, if and when cast-iron pistons must be substituted, the only changes necessary in the piston assembly will be the piston change plus certain detailed adjustments of the piston-pin fit. In addition, the distributor curve will have to be modified to retard the spark along most of the speed range.

The chief obstacle in a second design was in attempting to cast the dome of a dome-type piston in iron without machining, and to hold its profile to as close limits as when cast in aluminum. Engineers at this company report that the problem has been worked out satisfactorily through developments in foundry technique. Instead of increasing the size of the connecting rod, they plan to use a stronger steel, changing from SAE 1045 to SAE 1340. Since the wristpin is of the type that is held in the connecting rod, bushings of bronze or other suitable material will be incorporated in the cast-iron pistons. This company will not be forced to change bearings because it already had adopted the new thin-babbitt bearings with porous matrix, just described, in its 1941 models.

The V-type engines of another line of cars will require heavier crankshaft counterweights to compensate for the increased reciprocating weight of the piston assembly if cast-iron pistons are employed. No changes in the connecting rod are believed necessary for this particular V-type design.

Since production of cast-iron pistons requires different and more extensive machining and grinding operations, a complete new machine-tool equipment set-up is required. To insure against a possible future shutdown of automobile production because of curtailment of piston aluminum, many manufacturers already have production equipment for cast-iron pistons set up and ready. Executives of one large corporation report that it could swing into production on cast-iron pistons on a month's notice. This same corporation already has spent about $750,000 in *getting ready* to replace aluminum-alloy pistons with those of cast iron. Another idea of the cost of design and development, of new equipment, and the additional production cost of the change can be obtained by the estimate of another large corporation – that the substitution of cast-iron pistons will raise the cost of each car by approximately $0.70.

Whether or not aluminum-alloy pistons will ever have to be replaced by those of cast iron depends upon future availability of low-grade aluminum alloy. The OPM's definition of low-grade alloy is "any copper type of aluminum alloy reduced from scrap containing at the most 87%

aluminum, the remaining constituents of said alloy being composed of at least 6.5% copper and 1.5% of zinc; or, if said remaining constituents do not contain 1.5% of zinc, they shall contain at least 1% of either nickel, manganese, tin, lead or bismuth, or at least 1% of a combination of nickel, tin, manganese, lead, or bismuth." In short, low-grade aluminum alloy contains no virgin aluminum – all the aluminum in it is remelted from scrap. At the time that this paper was prepared, low-grade aluminum alloy had a B-4 Priority Rating which allowed passenger-car manufacturers 90% of their 1940 requirements on aluminum pistons.

Whether this restriction soon would be tightened enough to force a shift to cast-iron pistons, or whether it will be relaxed to a 100% or even an unlimited priority rating is a moot point. Aluminum suppliers point out that the Government to date has required little low-grade aluminum alloy for defense since it cannot be sheeted or forged, the amount required being used mostly for miscellaneous die castings; therefore, virtually all the low-grade aluminum alloy will be available for civilian use. Furthermore, they argue, with the tremendous increase in aluminum forging and casting for defense, the aluminum scrap available for low-grade alloy may increase to the point where the supply of low-grade alloy may even exceed the defense and civilian demand. Engineers who prophecy further curtailment, on the other hand, point to the growing shortage of high-grade aluminum alloys and predict that means will be found, such as modifying defense specifications, to utilize more and more of the low-grade aluminum in defense production to relieve this shortage. In addition, it is reported from abroad that the British are utilizing 85% of their aluminum scrap for defense purposes. As a result of this growing uncertainty, as this paper goes to press, all indications point to a shift to cast-iron pistons on a number of cars with the 1942 models.

Before the aluminum shortage, many automobile pistons were made from high-grade aluminum alloys containing virgin aluminum. In these cases pistons of low-grade alloy either have been substituted for those of high-grade alloy, or are being substituted as soon as present stocks of high-grade aluminum pistons are exhausted.

In one such substitution, pistons of a special low-expansion high-grade alloy have been replaced by those of low-grade alloy. Since the low-grade alloy is heavier and has higher thermal expansion than the piston of this high-grade alloy, the low-grade piston weighs about 1 oz more; is fitted 0.0005 in. looser in the cylinder; and has a longer slot in the side of the skirt. Heat-treatment of the low-grade-alloy pistons also is different from that used for high-grade-alloy pistons. Another company that shifted its pistons from high-grade to low-grade aluminum alloy was able to design its low-grade piston so that it weighed exactly the same as the replaced piston. Since this company employs V-type engines, this equalizing of weights obviated the necessity for changing the crankshaft counterweights. A third company was more fortunate: it found that it could substitute trunk-type pistons of low-grade aluminum alloy used in one engine of its line of the same bore and stroke for the strut-type pistons of high-grade aluminum alloy used in another model.

Brake Pistons – Selecting an alternate material for brake wheel pistons of extruded high-grade aluminum alloy has been a knotty problem in most automobile plants.

The number of different materials of which sample brake wheel pistons were made and tested by one company illustrates the characteristic thoroughness of passenger-car engineers in running down every possibility in their efforts to find the most satisfactory and economical alternate material. This company tried powdered metal, cast iron, screw-machine steel, welded steel forgings, and special glass before approval was obtained on a low-grade aluminum casting. The powdered metal was found unsatisfactory without some form of plating or coating because the oil in the brake system formed a deposit caused by electrolytic action. The screw-machine steel, welded steel forging, and cast-iron samples of brake wheel pistons also formed scale unless tin-coated or chrome-plated. Low-grade aluminum cast in permanent molds proved to be cheaper than the other alternates considered, since it did not have to be plated or coated. However, the low-grade aluminum castings proved more costly than the original high-grade extrusions because the low-grade castings require the extra operation of facing the back side of the piston, necessitate more grinding, and take longer to produce. It should be pointed out here that any of the alternate brake pistons considered, if properly plated or coated with chrome or tin, would function in the brake wheels as well as the original high-grade aluminum extrusion. Selection among these, therefore, was made solely on a basis of cost. This same company also has developed injected plastic brake wheel pistons as an optional alternate, and to insure against a shut-off of low-grade aluminum alloy.

One large brake company, which formerly used aluminum alloy in both brake wheel cylinders and master cylinders, is experimenting with coated or plated pulverized iron and plastics. This pulverized iron is the same material used in the gears of the oil pumps of many cars. Since the plastic pistons are reported to be satisfactory physically, the development work is being directed toward increasing their production rate.

Cylinder Heads – The low-grade aluminum-alloy cylinder heads employed originally on two 1941 models are being replaced with cast-iron heads. In both cases compression ratios are being lowered to suit the lower heat conductivity of the cast iron, the reduction being from 7.2:1 to 6.6:1 in one of these engines. In addition, the aluminum-alloy cylinder heads which were carried as optional at extra cost on many lines, will be discontinued. In their place, some companies will offer as optional special cast-iron cylinder heads with higher compression ratios and combustion chambers designed for premium fuels.

Miscellaneous Parts – Aluminum-alloy die castings have certain special properties which make them very difficult if not impossible to replace in certain critical parts without increasing costs to an excessive degree.

One of these parts is the main control valve body for a well-known automatic transmission. This is a very intricate casting with tortuous oil passages and cylinder and valve-stem bores. If made of zinc alloy, the chief engineer of this company pointed out, it would be more likely to corrode and would not give such close fits, besides zinc alloy is also a shortage material. If made of cast iron, the amount of machining necessary would be out of all proportion, and it is doubtful if the result would be so accurate. "If I were told that I could have only one aluminum part in our cars," he declared, "this would be it."

The aluminum-alloy rocker-shaft bracket on one line of valve-in-head engines is another such critical part because of the difference in expansion between the aluminum and the alternates considered, such as malleable iron, and be-

cause of the effect on valve lash. The difficulties of machining a cast-iron substitute for the aluminum-alloy directional-switch housing located on top of the steering gear are responsible for the reluctance of this company to make this substitution unless forced to do so.

Following are other typical aluminum-alloy parts with the alternates which have been approved to replace them: engine front-end oil seal – plastic alternate; timing gears – plastic gears with steel center; valve-chamber cover – cast-iron substitute; distributor body – cast-iron alternate; sheet aluminum horn resonator discs – sheet steel; aluminum foil in electrical system – tin foil.

Most zinc-alloy die castings used in passenger cars contain about 4% aluminum which zinc manufacturers do not want to replace. Without it, they claim, the zinc has a high affinity for the iron in their melting pots and, therefore, has a solvent action; also, the aluminum strengthens the alloy.

Eliminating Nickel

Work being done to eliminate or minimize the amount of nickel used in passenger cars, in general, consists of substituting non-nickel steel alloys for those of nickel; replacing other nickel alloys used in miscellaneous parts; taking out the nickel used in certain cast irons; eliminating or reducing the amount of nickel used in chrome plating; and dispensing with the plating entirely. Several companies report that 80% or more of the nickel will be eliminated from their cars. Replacing the nickel steel alloys is proving the biggest job because of the modifications necessary in machining and heat-treating, and sometimes in the design of the part itself.

Substituting for Nickel Alloy Steels – Nickel alloy steels were adopted originally for the transmission gears, rear-axle gears, and many other parts of the power-transmission system and steering system in many cars because of their toughness, hardness, and resistance to shock. Recent trends in design of the power-transmission system, such as helical gears, synchromesh, fluid flywheels, and also in design of the steering system, however, have all been in the direction of eliminating jerks or shocks. For this reason, a trend toward replacing the nickel alloy steels with those of carbon-molybdenum, chrome-molybdenum, manganese-molybdenum, carbon-manganese, or high-tensile low-alloy steels and variations of these, started some time ago, so that some cars today have few nickel-steel parts left. Other cars have retained many of the nickel steel alloys, the nickel in which amounts to from 1 to 2 lb per car. Engineers of these companies explain that the nickel steels were retained more for their production economy than because of need for their extra shock resistance and hardness, even though the nickel steels themselves usually cost more. They say that nickel steels permit more latitude in production control – such steels have enough factor of safety to take care of heat-treatment and other operation tolerances that cannot be permitted with some other alloy steels, such as chrome-molybdenum. For example, when a change is made from nickel-chromium to chrome-molybdenum steels for helical gears, extreme care must be exercised to insure that the helix angle is not changed, thereby producing noisy gears, when quenching during heat-treatment. In short, nickel-alloy steels do not have to be controlled and supervised so closely as do most other steels, and suitable modifications in machine-tool speeds and feeds, annealing, case-hardening, or other heat-treatment must be made when a switch is made to alternate alloy steels.

The accompanying table gives the steel substitution program for one line of passenger cars which is being shifted

Name of Part	SAE No. of Nickel Steel	SAE No. of Substitute Steel
Axle Shafts (semi-floating type)	X-3150-A	4150
Pitman Arm Studs	3135-A	1340 & 4140
Steering Arm Ball Studs	3115-A	4115
Front Axle Kingpins	3115-A	4115
Steering Knuckles	3135-A or X-3140-A	4140
Steering Knuckle Support	3045-A or X-3140-A	4140
Above are "Safety Items"		
Axle-Shaft Bolts (full-floating)	3140-A	4140
Differential Pinion Shafts	3115-A	4115
Differential Side Gears	3130-A	5130
Rear Axle Ring Gear and Pinions	4615-A	C-4120
Universal Joint Yoke Trunnions	3115-A	4115
Differential Pinion Spiders	4615-A	4115
Differential Lock Screw	3135-A	4140
Propeller Shaft Couplings	3135-A	5145

from SAE 3100 Series nickel-chromium steels and SAE 4600 Series molybdenum-nickel steels to SAE 4100 Series chrome-molybdenum steels, SAE 1300 Series manganese steels, and SAE 5100 Series chromium steels.

Nickel steel substitution programs of some other companies retain the SAE 4600 Series nickel-molybdenum steel (1.65 to 2.00% nickel) for gears but eliminate all other nickel steels, substituting SAE 4100 Series chrome-molybdenum steels for SAE 2300 Series nickel steels and SAE 5100 Series chromium steels for SAE 3100 nickel-chromium steels. In addition, many companies are experimenting with so-called "low-alloy high-tensile steels" that contain small amounts of almost all the best-known alloying elements – carbon, manganese, silicon, copper, chromium, nickel, molybdenum, zirconium, sulfur, and phosphorus. These steels are reported to have good machinability and hardenability. Consequently, the possibilities of getting as good finish at high machine speeds as, for example, with nickel-molybdenum SAE 4600 steels, are being explored. Engineers of one company which retains nickel steels for connecting-rod bolts and rear-axle gears have tested and approved SAE 1320 manganese steel for the rear-axle gears and SAE 4140 chrome-molybdenum steel or low-alloy high-tensile steel for connecting-rod bolts so that the substitution can be made quickly if necessary.

Corrosion- and heat-resistant steels containing about 18% chromium and 8% nickel have been used on many lines of passenger cars for exterior moldings and trim. Because of their high nickel content and purely decorative

function, all these steels are being eliminated. In their place some companies are substituting corrosion-resistant steels containing about 18% chromium; others are replacing them with carbon steel chrome-plated; and still others are giving serious thought to eliminating almost all the moldings and trim. For small moldings, substitution of plated copper is contemplated on a number of lines. Replacement of nickel-corrosion-resistant steels with those of chromium is planned by fuel-pump manufacturers for the fuel-pump arm.

Valves – Chrome-nickel exhaust valves (containing from 1½ to 12% nickel) are used in a majority of passenger cars. Many passenger-car engineers are of the opinion that, if they were permitted to have nickel-steel alloys in only one part, they would retain them in the exhaust valves. The reason, as explained by valve engineers, is that the combination of nickel and chromium shows the greatest resistance to the excessive heat and oxidizing and corroding influence of the exhaust gases. It is true that exhaust valves with little or no nickel containing 7 to 18% chromium and 2 to 3½% silicon are performing satisfactorily in many passenger cars. Valve engineers point out, however, that the engines of many cars were designed for nickel-chromium exhaust valves and that substitution of silicon-chromium valves either would have far-reaching effects on design out of all proportion to the weight of the nickel saved, or would result in a loss in performance and lower valve life.

To illustrate the critical nature of the valve problem, one engineer describes the change in physical design required for the elimination of nickel as follows:

"Valves were made from a nickel-chrome alloy. With the elimination of nickel, the chrome was increased to 18% and the silicon to 3½% with molybdenum added to control the fine grain structure. This alloy, however, had a different *hot strength* than the former alloy so a deeper cantilever section was required in the valve head. The angle of the under side of the valve head had to be increased from 8 to 15 deg. Naturally, this design change added weight, and added it to a part where inertia effects are very important. But that is the best that can be done today."

Assuming exhaust valves of austenitic non-hardenable steel alloy containing up to 12% nickel, and assuming that a substitution were required, another valve engineer indicates that he first would change to hardenable steel alloy containing 1½% nickel, 18 to 20% chromium, and 2% silicon. If not permitted 1½% nickel, he would shift to a silicon-chromium alloy of 8 to 18% chromium and 2 to 3½% silicon.

Before the nickel limitations, most passenger cars employed inlet valves of SAE 3140 or similar nickel-chromium steels containing 1 to 1.5% nickel. Since the inlet valve does not have to resist excessive heat and corrosion as does the exhaust valve, valve engineers point out that SAE 4140 chrome-molybdenum steel, SAE 5150 chromium steel, carbon-molybdenum steel, or even SAE 1050 carbon steel can be substituted to give equivalent performance. Such changes are being made in many companies.

Roller Bearings – Many of the roller bearings used in passenger cars are made of case-hardening SAE 4600 Series of molybdenum-nickel steels. Substitution in this case is unlikely for two compelling reasons: (1) More than half the production of roller-bearing companies is now utilized in defense equipment and (2) Change to a non-nickel steel alloy would necessitate changes in physical properties, tolerances, and so on, large enough to require extensive retooling and revision of engineering standards now in wide usage. Consequently, to produce non-nickel roller bearings for civilian use, an additional plant would be necessary. The new plant would require machine tools, dies, die makers, and skilled machinists urgently needed for defense.

Miscellaneous Parts – Throughout the automobile are many other parts that contain small amounts of nickel for which a satisfactory substitute cannot be found without going to very expensive material or extensive redesign.

The electrical system contains many such parts using nickel alloys – spark-plug wires; distributor wires; distributor springs; compensating shunts for voltage regulators; heating elements; and bi-metal elements of electrical indicators, such as temperature gages, and so on. It is estimated that such parts call for only ¼ of 1% of all the nickel available to industry[1]. In some cases, however, molded carbon resistances are being substituted successfully for nickel resistance wire.

Another critical nickel-alloy part on one line of cars is the tube passing through the exhaust manifold which connects to the automatic choke. The chief engineer of this company points out that only nickel alloys have been found to be corrosion- and heat-resistant enough for the purpose. If forced to shift to non-nickel material, he explains, he would face the alternative of using carbon steel and having it burn out frequently or of redesigning the hot box for the choke, which latter expedient would require a change in carburetor calibration.

Cast-Iron and Cast-Steel Alloys – Of recent years the practice has been to add nickel in small quantities from 0.25 to 1.25% to the cast-iron and cast-steel alloys used in cylinder blocks, cylinder heads, and pistons to toughen and harden the iron, thus permitting thinner, stronger sections of uniform hardness.

If necessary this nickel can be taken out, metallurgists report, and virtually the same properties obtained by increasing the silicon content or by substituting other non-nickel alloying combinations. In cast crankshafts and camshafts also, they point out, the work formerly done by a small amount of nickel in the alloy could be done by adding copper.

Plating – Before the shortage in nickel, passenger cars had as much as 3.6 lb of nickel in their plating, between 1½ and 2 lb being an average amount. Through recent developments, this amount will be reduced to a value ranging from one-half to less than one-third of the former weight so that the average car will have only about ½ lb of nickel in the plating. In one line of cars the nickel thickness has been reduced from 0.0007 in. to 0.0002 in. This decrease in the thickness of the nickel coat has been made possible by using bright copper, which is three or more times thicker than the dull copper used previously, in conjunction with a special bright nickel. The bright nickel over the bright copper produces a practically flawless surface with high gloss so that the buffing operation formerly required before applying the final chrome coat, can be eliminated. Extensive salt-spray tests on this new plating have proved that it is equal or better than the old plating in corrosion resistance. Most passenger-car com-

[1] See *Automotive News*, April 28, 1941, p. 18: Report of Address by Dr. J. S. Laird, Symposium on "Changes in Materials Due to Defense Requirements," Spring Meeting, American Society for Testing Materials, Detroit, Mich., April 18, 1941.

panies are planning to use this new plating in 1942 models when the extensive changes in equipment required are expected to be completed.

Nickel already has been eliminated from the interior plating used on many cars, and money saved, the copper being made three or four times as thick. One plating authority reports that exterior chrome plating without nickel has been tested in Germany and that, when tested in the rain, bronze stains kept appearing on the plating which had to be wiped off. These stains, he explained, were caused by the copper compounds coming through the outer chrome coating.

In case of shortage of nickel anodes for plating, one authority[2] points out that scrap nickel can be utilized for anodes by packing the scrap in a non-metallic basket in close contact with a small anode, or that insoluble anodes of carbon or graphite be used while the solution is fed from an increased quantity of nickel salts.

What to do if the supply of nickel for plating is cut off entirely? One company is experimenting with indium as a substitute. This material is now very expensive, but its price is reported to be dropping; it also requires heat-treatment and buffing. Cadmium is suggested as another substitute, but such plating would lose much of the former brightness. Silver is mentioned as another possibility. Silver plating on automobiles may not be as expensive as many might think; according to one plating engineer,[2] a 0.0002 in. deposit on brass costs 5½¢ per sq ft. To inhibit tarnish, he suggests a coat of clear lacquer.

The consensus seems to be, however, that exterior plating will be eliminated if nickel for this purpose is cut off. Surfaces now plated would be painted. One large company is now seriously considering elimination of plating regardless of whether or not nickel is available for plating. Such a trend, whether enforced or not, might not be so undesirable in the opinion of some engineers. With no large areas of glittering metal to divert attention from their graceful lines, modern motor cars might be revealed to the buying public for the first time as forms of inherent beauty.

Taking Out Zinc

Although preliminary estimates state that about 50% of the zinc formerly employed will be taken out of the 1942 passenger cars, engineers have been studying substitutes for the literally hundreds of die castings that are scattered throughout modern cars in preparation for a possible complete shut-off of zinc supplies. According to recent estimates, however, many passenger-car engineers are prepared to remove more than 80% of the former zinc weight from their 1942 models. Zinc alloys generally used for these die castings contain about 96% zinc, 4% aluminum, and traces of other alloying elements, such as copper and magnesium. In other zinc alloys, the copper content is stepped up to a value between 0.75 and 3.5%.

Decorative Parts – Zinc-alloy die castings that serve merely to decorate the car should logically be the first to go, engineers believe, and this has been the policy pursued by all makers. Fortunately it is in these parts that the greatest weight saving can be made. The hope is that these savings will be great enough to permit retention of zinc-alloy die castings for important functional parts.

Elimination of zinc die castings for radiator grilles makes by far the greatest single weight saving. Weighing as much as 30 lb apiece, their substitution lops a big chunk off the total weight of zinc die castings in the average car. This total weight of zinc-alloy die castings ran over 80 lb in some models. Plated stamped steel or antimonial-lead die-casting alternates will appear on virtually all 1942 models. So closely do some of the new plated steel grilles resemble their zinc-alloy predecessors that it will take an expert to tell the difference in material. Tooling costs and car production have been an important factor in the choice between steel stampings and antimonial-lead die castings as radiator grille substitutes. The dies necessary for stamping steel radiator grilles cost from $100,000 to $300,000, whereas dies for die-casting these grilles cost only about one-tenth as much – from $10,000 to $20,000. The per-car cost of this change can be excessive unless it can be distributed among a great many cars. On the other hand, the steel used for stamping a radiator grille costs only about one-fifth as much as the zinc alloy necessary to die-cast the same grille. As a result, in many cars which are produced in relatively moderate quantities, antimonial-lead die castings are being considered to replace the zinc-alloy die castings in radiator grilles since, in these cars, the cost of the steel stamping die outweighs the saving in material cost.

Antimonial-lead die castings cost about 25% more per part than zinc-alloy die castings. The former costs about 25% less per pound but has a specific weight more than 50% higher. Although its tensile strength is only about one-third that of zinc alloy, engineers who plan to use it as a substitute material contend that its strength is adequate for the unstressed decorative parts for which it is to be used. An important advantage of their use is that the antimonial-lead die castings can be made from the same dies used previously for zinc. On the other hand, antimonial-lead die castings must be made with care in buildings which are separated from plants in which zinc alloys are being cast; otherwise the lead may get into the zinc and spoil it. Antimonial-lead also is difficult to polish. Radiator ornaments, trim strips on fenders, lamp bodies, and other ornamental strips are also slated to be made of antimonial-lead die castings on some cars when zinc is no longer obtainable. Furthermore, antimonial-lead die castings can be made with steel reinforcements in them in parts where additional strength is required.

A number of alternates are being considered to replace zinc-alloy die castings in interior hardware and outer door handles. Plastics reinforced with spring steel and cast-iron bushings have been approved as optional on a number of lines. Plated plastics are also being considered for these parts. These plastics are plated by spraying them with metallic bronze and then adding chrome plate. No undercoating of copper and nickel is required to prevent rust. Plated or painted cast iron is also being considered as an alternate for these parts. A large decorative die casting that is being universally replaced by a plated steel stamping or plastic is the instrument-board grille which weighs as much as 2¾ lb as a die casting in some cars.

The foregoing examples are typical of numerous substitutions being planned in decorative parts. These replacements are comparatively easy to effect when compared with making those for important functional parts now

[2] See *Iron Age*, May 15, 1941, pp. 39-45: "Priorities and the Plater," by Adolph Bregman.

produced as intricate zinc-alloy die castings. In the latter type of parts, the entire design usually is built around use of the zinc alloy.

Carburetors and fuel pumps are examples of these critical designs. Companies producing them, nevertheless, preparing for any contingency, have gone ahead and developed carburetors and fuel pumps in which all but a few small zinc-alloy parts have been replaced by those of cast iron. One of these retained parts is the main discharge nozzle of the carburetor. Substitution of another material for this vital part, carburetor engineers explain, would spoil the accuracy of the carburetor. Stems and needles, now of brass, can be replaced by plated steel stampings or cold-rolled steel. In one carburetor model the design of the float was changed in the cast-iron alternate to conform to a desirable change in the float housing casting. One company estimates that cast-iron alternate carburetors and fuel pumps will cost 25% more on an emergency basis than those of zinc alloy, the difference being explained in the greater tooling and machining cost, and in the higher scrap loss because of blowholes and core shifts. Other companies have developed carburetor and fuel-pump covers of steel stampings with brazed-in fittings.

Considerable work is being done by a number of companies on the development of plastic alternates for zinc-alloy die castings. The extent of this work can be best explained in the words of the plastics engineer of one company who announced that "plastics ultimately can be released to replace 90% of existing zinc-alloy die-cast parts." The plastic alternates will be as strong or stronger, he contends, especially in impact strength. Plastic alternates have many advantages over those of cast iron. For example, the die-casting molds often can be adapted to handle the plastics, and the tooling and machining usually required are negligible by comparison.

Although there is general agreement as to the suitability of plastics for decorative parts such as ornaments, escutcheons, instrument panels, and glove compartments, their adequacy for important and intricate functional parts such as carburetors and fuel-pump bodies is questioned by some authorities. These skeptics contend that, compared with zinc-alloy die castings, plastics are less stable dimensionally and thus might impair the accuracy of certain parts; that holes cannot be cored as deep or as close; that their impact strength is less than that of zinc; and that production rates will be lower, even with injection molding of thermosetting plastics using accelerators to increase the rate of cure, unless expensive multiple-cavity molds are resorted to. Furthermore, if used for carburetors or fuel pumps, plastics must withstand high engine temperatures, zero atmospheric temperatures, and the shaking caused by engine vibration. With reference to temperature, they contend that most plastics become brittle at low temperatures and expand unevenly at high temperatures and, therefore, their suitability for exterior parts is questionable. The plastic carburetor body must be strong enough to support the air cleaner, and the fuel-pump body must stand the strain on the inlet and outlet connections and the shocks caused by the cam lever arm. For this reason, some engineers believe that reinforcement may be necessary in such applications. Another difficulty to be overcome, they point out, is the solvent action of the fuel, especially benzol fuel, on certain plastics. Plastic manufacturers themselves are warning over-enthusiastic users of the limitations of their materials.

That many of these problems, nevertheless, may be solved is indicated by the amount of development work being done by several companies. One plant is reported to be testing batteries of plastic fuel pumps. Another company, which operates a plastic department with a capacity of 25,000 plastic pieces a day, has developed injection molding to a point where the finished piece of certain parts can be removed from the die 35 sec after injection. Additional advantages of plastic parts, this company points out, are their excellent corrosion resistance, and that they need no buffing or polishing after manufacture. A 32-in. garnish molding, claimed to be the longest piece ever molded in an injector press, is made in this plant. Beautiful and novel effects are being obtained in decorative plastic parts made in this department, by painting colored designs on the back of transparent plastics.

Since automobile brass contains about 30% of zinc, alternate materials are being worked out for parts made of this material. In this respect the most important changes are being effected in the radiator, where brass water tubes and upper and lower tanks are almost universally being changed to copper because of a growing scarcity of brass sheet. At the time this paper was prepared, copper cost less per pound but, since more weight per part is required, the substitution will cost a few cents more per radiator. Terne-plated steel or steel coated with special corrosion-resistant paint also could be used for these parts if copper gets scarce, but radiator engineers point out that they would not stand up as well because of the electrolytic action between the copper and the steel, and that the heat conductivity and corrosion resistance would be lower. Increasing the gage might alleviate the corrosion difficulty to some extent, they suggest, since it would take longer for the steel parts to corrode through. In some radiators a small amount of silver is being added to the copper used for the fins of fin-and-tube radiators to make them stiff enough after they are baked to permit a hole to be punched in them.

An aluminum-plated glass substitute is now ready to replace the silver-plated brass reflectors on the composite units of sealed-beam headlights. The piston of the master cylinder in many brake cylinders formerly has been made of brass; it is being replaced by pistons of low-grade aluminum or tin-plated screw-machine steel. Stamped and plated steel hub caps will replace those of brass that were spun over, in many cases eliminating the necessity for zinc-coating the inside surface. Many brass model identifications will be eliminated in the 1942 models.

Several examples will illustrate the amount of work often required to find satisfactory alternate materials for zinc-alloy die castings. At one plant, cast iron, steel stampings, impregnated wood, and powdered metal samples were made and tested before a plastic was finally approved as alternate material for the clutch torque shaft pivot bearing. The plastic was selected mostly for reasons of economy; it proved cheaper because machining was eliminated.

Before releasing pressed steel as alternate material for horn projectors, another company tried a lower grade of zinc alloy, injection-molded plastic, compression-molded plastic, two-piece plastic with steel insert, one-piece cast iron, two-piece cast iron, and antimonial lead.

Not so long ago long trumpet-type horns were used generally in passenger cars. Although these horns were perfectly satisfactory in performance, they were replaced

Zinc-Alloy Die-Cast Parts	Alternate Materials Approved
Radiator grilles	Steel stampings or antimonial-lead die castings
Instrument-board grilles	Steel stamping or plastic
Radiator ornaments	Plastics or antimonial-lead die castings
Lamp bodies	Steel stampings or antimonial-lead die castings
Trim strips on fenders	Antimonial-lead die castings
Interior hardware	Reinforced plastics, plated or unplated, or plated or painted cast iron
Exterior door handles	Reinforced plastics, plated or unplated, or plated or painted cast iron
Carburetors	Cast iron or plastic
Fuel pumps	Cast iron or plastic
Clutch torque shaft pivot bearing	Plastic
Horn projectors	Pressed steel
Horns	Steel stampings
Speedometer gears	Plastic
Speedometer pinion sleeve	Cast iron
Defroster funnel	Steel stamping
Steering-wheel hub and brackets	Steel stamping or malleable casting
Horn button	Plastic
Horn ring	Steel stamping, plated and welded
License bracket	Cold-rolled steel
Shock-absorber guide	Cast iron
Transmission cover	Cast iron or plastic with steel reinforcement
Transmission shift levers	Machined steel forgings
Medallions	Back-painted injection-molded transparent plastic
Door lower channel roller assembly	Steel
Outside mirror assembly	Steel
Rear window light frame (convertible)	Pressed steel
Lock switch housing	Cast iron or antimonial lead
Directional light switch housing	Plastic or steel stamping

by spiral snail-shaped horns made of zinc-alloy die castings, mostly because of the space saving effected. In the present emergency, therefore, most manufacturers are planning to go back to the old trumpet-type stamped-steel horn.

Largely because of the high percentage of zinc oxide employed in their white side walls, white-walled tires are being discontinued by virtually all makers, and the amount of zinc oxide in other rubber compounds is being reduced.

To give a more complete picture of the work being done, the accompanying list of approved alternate materials is given. The list is not intended to be complete, but merely to give some typical examples of alternates released by some companies.

Neoprene

Inclusion of Neoprene in Government defense specifications and resultant mandatory priorities on civilian use of this synthetic rubber have made its substitution an important part of alternate-materials programs. Neoprene is the only non-metallic substance under priority to date. With a few notable exceptions, substitution of available synthetic rubbers with approximately comparable physical properties, natural rubber, or leather for Neoprene has been proceeding satisfactorily in most passenger-car applications. The amount of Neoprene formerly used in the average car is not high – only a few pounds. Most substitution programs call for reducing this weight by more than 50%. In addition, factories are replasticizing Neoprene and recovering about half of their supplies as well as mixing it with natural rubber and other synthetics to effect further economies.

The bottleneck in synthetic rubbers, according to one expert, is in the production of butadiene. With new plants soon coming into production to make this important raw material, he explains, supplies of synthetic rubbers shortly should be more plentiful.

At one plant, gaskets of lignin and another synthetic rubber have been released as alternates for Neoprene-lignin in crankshaft seals, and independent front-wheel suspension seals. At another company, spring-backed leather gaskets have been approved in place of Neoprene for oil

and dirt seals for shafts and front-suspension parts. In these applications the spring backing provides the automatic compensation for wear that is effected by the swelling of the Neoprene. One carburetor manufacturer reports that another synthetic has been approved in place of Neoprene for the "balloon cloth" (silk impregnated with synthetic rubber) used on carburetor diaphragms. A similar substitution of cloth impregnated with another synthetic for Neoprene-treated diaphragm cloth is being made in certain vacuum units.

Satisfactory substitutes for gasoline lines seem hard to find. One car manufacturer is testing gasoline hose lined with four prospective alternate synthetics. Flexible metal tubing is being tested by another company, but it is expensive and contains brass.

Water-pump seals and radiator hose present another critical Neoprene application as substitutes must be unaffected by radiator compounds containing large amounts of oil and various chemicals.

A third substitution problem arises in the Neoprene boot that seals the ball-and-trunnion type joint in the universals of one line of cars. Engineers at this plant report that Neoprene is the only synthetic rubber of many tested that has the chafing and flexing resistance required in this part.

In line with their determination to make no substitutions that will lower performance or quality, passenger-car engineers are hoping that they will be spared enough Neoprene for these critical parts, at least until the time when satisfactory alternates can be developed.

Typical Neoprene parts for which alternates have been developed at one plant are: rear main bearing crankshaft seal, axle drive pinion carrier gasket, valve spring cover gasket, radiator pressure cap gasket, and thermostat gasket.

Chromium

Although mandatory priorities had not been invoked on chromium when this paper was prepared, material substitution programs have taken into account the tightening of supplies of this material imported from Turkey, Africa, Greece, and other foreign lands.

The greatest amount of chromium used in passenger cars is in alloy steels. In case of a tightening of this material, the first parts to be replaced would be the 18% chrome corrosion and heat-resistant steels used for body molding and trim. Some companies plan to replace them with plated carbon steels or plated copper; others are slated to remove them entirely in such a case, and use striped painted moldings and trim.

Exhaust valves would be definitely critical in event of a chromium famine as valve engineers cannot conceive of a satisfactory production exhaust-valve alloy containing neither nickel nor chromium.

Virtually all ball bearings and many roller bearings employ through-hardening chromium steels – usually SAE 52100, or modifications of this steel. These parts are also critical for the same reasons given for the nickel-molybdenum roller bearings: Most of the output of ball-and-roller bearing manufacturers is now going for defense equipment and any change in steel would require revolutionary revisions in production processes, tolerances, and engineering standards.

Shut-off of chrome and chrome-molybdenum alloy steels for gears, transmission parts, and steering assemblies would cause difficult but not insurmountable problems of substitution if nickel also were not available, according to some metallurgists. If manganese, molybdenum, vanadium, zirconium, and silicon were still available, they believe, satisfactory alternates could be worked out, although more generous sections might be required to compensate for possible deficiencies in impact strength. Metallurgists of one company contend that carbon-molybdenum steels could be substituted throughout the entire car with exception of the exhaust valves and roller and ball bearings. Small amounts of chromium are used in many cast irons for such parts as cylinder blocks, pistons, and cylinder heads. These could be replaced by other available alloying elements to give cast irons and cast alloys of the virtually same properties, it is reported.

The small amount of chromium used in so-called "chrome" plating is amazing – only about 0.36 oz per average car, although about the same amount is wasted in production and application. The reason is that the coating of chrome is only 0.000015 in. thick in many cases. For this reason it is logical to conclude that chrome plating is much more likely to be abandoned because of a shortage of nickel or copper, than of the insignificant amount of chrome employed. It is safe to predict, however, that chrome plating will be eliminated from passenger cars if the supply of either copper, nickel, or chromium is cut off.

Tin

Tin is another "strategic" material, most of which is imported from the East Indies and other foreign nations. Tin is used in passenger cars in babbitt bearings, bronze bushings, coatings for pistons and other parts, radiator solders, body solders, terne plate for mufflers and gaskets, and tin foil.

So thin have the babbitt linings of the main, connecting-rod, and camshaft bearings become (about 0.003 in.) in the average car today that only about ½ lb of babbitt is required. The tin in this babbitt ranges from 0.45 to 0.045 lb per car, depending upon the type of babbitt used. Tin-base babbitts contain about 90% tin; SAE 14 lead-base babbitts, 10% tin; and SAE 13 lead-base babbitts, 5% tin. Although the bearings of many cars recently have been shifted from tin-base babbitt to the less costly lead-base babbitt, tin-base babbitts are still widely used. In the event of tin shortages, bearing authorities recommend a substitution program in the following steps, depending upon the amount of tin available for bearings:

1. Replace all tin-base babbitts containing 90% tin with SAE 14 lead-base babbitts containing 10% tin.

2. Replace SAE 14 lead-base babbitts containing 10% tin with SAE 13 lead-base babbitts containing 5% tin.

3. If impossible to get the 0.045 lb of tin per car required for SAE 13 babbitt, go to a cadmium-silver-copper bearing alloy with indium added to inhibit the tendency of the oil to corrode these bearings. As the cost of such bearing linings would be excessive, they would be employed only under emergency conditions.

The tin used in bronze piston-pin bushings amounts to from 4 to 10%. Suggested alternates for this material

include aluminum bronzes that contain as little as ½% tin. The valve on one of the automatic clutches is being shifted from bronze to cast iron. Other bronze bearings and bushings are being replaced by powdered metal or molded plastics.

Protective tin coatings are used on a number of passenger-car engine pistons today, whether they are of aluminum or cast iron. Tin is very effective for this purpose, metallurgists point out, especially in the prevention of scuffing during the break-in period because tin itself is a lubricant. For this reason, the tin coating tends to be distributed automatically so that a close fit is maintained between piston and cylinder wall. Nonetheless, tried and proved alternates are available, and engineers are planning to use these alternates in event of a tin shortage. For aluminum-alloy pistons, if such are still in use, "anodizing" – which has long been used to protect the aluminum alloy pistons of many models – will be substituted. In this process a thin film of aluminum oxide is anodized on the piston surface by means of a sulfuric acid or chromate bath.

For cast-iron pistons – and there may be a lot more of them in passenger cars before tin gets short – an insoluble phosphate alternate coating is planned. This coating is reported to be crystalline in character and to absorb and hold oil in the same manner that an ink blotter absorbs ink. The coating increases the diameter of pistons from 0.0004 to 0.0006 in. One company plans to use this coating if required to shift from aluminum-alloy to cast-iron pistons, regardless of whether or not tin is available when the change is made.

About 4½ lb of radiator solder per car are now employed to build radiators. Some years ago these solders contained 40% tin, but this percentage has been gradually reduced until today radiator solder contains about 10% tin and 90% lead. On this basis the amount of tin in radiator solder per car is less than ½ lb.

If tin is curtailed, several alternates are available to replace tin-bearing radiator solder. One radiator engineer favors employment of hydrogen brazing now used to bond unit heaters. Another authority[3] points out that a 95% lead, 5% silver solder already is being used on radiators for glycol-cooled aircraft engines; he also recommends cadmium-silver, and lead-cadmium solders as usable.

A typical body solder contains 24% tin, 1% antimony, and 75% lead. Body solder is used principally for smoothing up bodies and fairing body joints. The amount of body solder used per car has been decreasing steadily and either has been eliminated or practically eliminated on the three low-priced high-volume cars. Gillett[3] estimates that the total consumption of tin for body solder was only 500 tons in 1937; it is unquestionably lower today. As a result, body engineers report that, if tin grows scarce, they simply will make bodies without solder.

Terne plate, containing about 20% tin and 80% lead, is used extensively on automobile mufflers, cylinder-head gaskets, and other parts. In a tin shortage, passenger-car engineers report that straight carbon steel or copper would be used instead of the terne-plated steel for mufflers and steel-and-asbestos or embossed thin-steel gaskets would be substituted.

For some electrical applications, tin foil could be replaced by aluminum foil if available.

[3] See SAE Transactions, June, 1941, pp. 205-212; "Made in America Substitutes," by H. W. Gillett.

Rubber

Whether or not a rubber famine will develop if supplies of crude rubber from the East Indies are cut off is a much-debated point. One rubber technologist contends that the chances of a serious shortage are remote. He points out that both private companies and the Government have built up huge stock piles. If imports of crude rubber should be cut off, he argues, these stock piles are adequate to supply the needs of industry until plants are ready to produce synthetic rubbers in mass production.

Pointing to a crude rubber consumption already 50% above a year ago, other authorities believe that a rubber famine is imminent after any shut-off of crude-rubber supplies unless construction of plants and facilities for *large-scale production* of synthetic rubbers is undertaken immediately. The four synthetic-rubber pilot plants, for which lease agreements recently have been authorized by the RFC's Defense Plant Corp. at the cost of $1,250,000 each, they point out, are designed to produce only 2500 tons each per year, whereas crude rubber is now being consumed at the rate of 800,000 tons per year. They explain that only the Government can make the huge investment necessary for these mass-production plants which, they estimate, will take two or three years to build and that the plants will require large quantities of critical materials such as monel metal and stainless steels. For these reasons, they urge an immediate start.

Anticipating the possibility of a crude-rubber shortage, rubber technicians are checking over specifications for their tire and mechanical goods compounds with the objective of replacing as much of the crude-rubber content as practicable with reclaimed rubber, synthetics, or other material. Rubber engineers report that many such specification changes already have been made voluntarily.

In addition, many recommendations have been made to reduce further the crude-rubber consumed. One rubber engineer recommends mixing 40% of reclaimed rubber into tire tread stock and increasing the amounts of reclaimed rubber used in other parts of the tire. As much as 80% of reclaimed rubber could be used in tire treads, he believes, if operators would hold passenger-car speeds below some predetermined moderate rate. Reclaimed rubber also could be used in rubber engine mountings, he points out. In addition, the following suggestions have been made: reduction in the number of tire sizes and grades, including elimination of white-walled tires as a dispensable luxury; that drivers continue to use tires until they no longer can be used with safety; that the practice of retreading be extended; and even that week-end automobile pleasure trips be curtailed.

Other Materials

Magnesium – Mandatory priorities were invoked some time ago on this material, but it was used in passenger cars only in a few places and there only in very small amounts. For example, there is about 0.04% of magnesium in zinc-alloy die castings, but zinc suppliers contend that it cannot be taken out without impairing the physical properties of the alloy; besides, zinc itself is critical. Some high-grade aluminum alloys contain from 1 to 1½% magnesium which acts as a hardening agent – another reason why these aluminum alloys are being eliminated as automobile piston materials.

Manganese – Some material experts are of the opinion

that critical shortages in manganese, another strategic material, may develop before they do in chromium. Almost all the manganese used in automobiles is in their steels; a certain amount is present in all steel. In case of an acute manganese shortage, it is believed that only the manganese SAE 1300 and X 1000 Series steels, which contain up to 1.90% manganese, would be affected. In addition, the amount of manganese (from 0.20 to 1.00%) present in all other SAE steels probably could be reduced by juggling other alloying elements. Whether or not an acute manganese shortage would bring serious problems, metallurgists believe, depends upon the availability of other alloying elements such as nickel, chromium, vanadium, molybdenum, and silicon. In the case of the free-cutting screw-stock X 1300 steels, Gillett[3] has pointed out that equivalent machinability can be had with less sulfur and manganese through the use of lead.

Antimony – In case of a shortage of this strategic material for antimonial-lead die castings containing 8 to 13% antimony, they can be replaced by parts of plated stamped steel, or plastics in most instances. It is also possible that, by the time such a shortage develops, the present bottleneck in zinc may be relieved so that engineers could go back to the original zinc-alloy die castings. To replace the 10% of antimony in the average battery grid, metallurgists[3] claim that 0.1% of calcium can be substituted.

Tungsten – One of the few uses in automobiles of tungsten – another material under mandatory priority – was for the valve inserts used in one line of cars, containing from 14 to 17% of tungsten. Engineers of this company announce that the material of these inserts already has been changed to a chrome-molybdenum alloy to eliminate the tungsten. The fact that molybdenum high-speed steels can be substituted satisfactorily for tungsten high-speed steels is now well known and has been explained fully.[3] Because tungsten is irreplaceable in sintered-carbide cutting tools, it is unlikely that supplies of the small amount required will be affected by tungsten shortages.

Leather – Anticipating a possible leather shortage, one automobile company has been experimenting with oil seals that have treated paperboard as a base.

Cork – A cardboard alternate for the cork used in bodies of some cars is under consideration.

Conclusions

1. Substitutions of alternate for critical materials made or definitely planned to date will necessitate no lowering of the standards of safety, durability, performance and comfort established in the 1941 passenger cars. So far, passenger-car engineers report, all alternate parts are required to pass the same proving-ground and laboratory tests used on the original critical parts.

2. In virtually all cases, parts of alternate materials cost more than do the former critical parts, the increase being proportional to the amount of new tooling, extra machining, or other additional processing necessary, and to the speed with which the shift must be made, as well as to the comparative cost of the material. These added costs are running into millions of dollars throughout the industry.

3. Although adopted through necessity, there are indications that some of the alternate parts will stay permanently in automobiles after the emergency is over since they will have time to prove their worth, demonstrating hitherto undiscovered qualities. In addition, the cost of many of these parts will have been lowered through design and production development. This is one of the brightest spots in the picture since the exigencies of the emergency will force many developments that would not have been made under normal conditions.

4. In considering alternate materials for certain parts in the automobile, engineers working for the best interests of national defense often find themselves in a dilemma. Substitution of the alternate material will release a relatively small amount of material needed for defense, when compared with that released by replacing other parts, but production of the alternate parts requires extra machining and processing, entailing additional machine tools, machinists, and other skilled labor needed just as urgently in defense industries as is the critical material itself. The example most often cited to illustrate this problem is that of carburetors made of critical zinc alloy versus those of alternate cast iron. The zinc-alloy carburetor die castings weigh little when compared with zinc-alloy radiator grilles; also, they require virtually no machining or processing. Cast-iron carburetors, on the other hand, would require additional batteries of machine tools.

On behalf of the SAE Passenger-Car Activity Committee the author gratefully acknowledges the splendid cooperation given him by the many engineers who contributed material for this paper.

DISCUSSION

Foresees Problems of Service Replacement

– James C. Zeder
Chief Engineer, Chrysler Corp.

MR. BISSELL'S article should be helpful in acquainting those not too closely associated with what is being done by manufacturers in cooperating in the national defense effort and in pointing out that a substitute does not mean cheese.

The subject has been so well prepared that little is left to be said. However, he has reported facts disclosed by his investigation and, no doubt, has purposely omitted any personal comments or suggestions. These would have been interesting.

Mr. Bissell relates that stand-by equipment is in many cases being set up. This doesn't seem efficient when the defense program as a whole is considered. It is costly to manufacture and ties up needed tool and die men.

Mr. Bissell deals principally with substitutes for strategic materials. However, this class of material might not prove to be any greater problem than some other material which is plentiful in the raw but for which there are insufficient facilities for fabrication due to the unusual demand. Also, transportation may be a cause of shortage.

Alcohol for anti-freeze or other uses may be scarce, not because of alcohol-producing materials, but because of the lack of plant facilities or scarcity of boats for transportation of alcohol-producing molasses, and so on. Synthetic rubber and plastics may be in a similar category. It is not unusual to have to substitute a different gage of ordinary cold-rolled steel because the mills have all the business they can handle in gages other than the specifications require, and it is necessary to take what they have because of the lack of the time to make a specified gage which is not in stock.

Mr. Bissell points out very well in the case of roller bearings why factors other than materials must be considered.

Perhaps the statement that antimonial-lead die castings can be made from the same dies used previously for zinc should be qualified. This is true with some parts but, in many cases, it is necessary to redesign the part and remake the die before it is satisfactory in antimonial-lead.

The question of service replacement parts of strategic materials is becoming a more serious problem. If there is no material available for a zinc die-cast radiator grill of a past model, what is the answer? The volume would probably be entirely too small to justify stamping dies. Either some special provision will have to be made to allow material for service parts or else monstrosities are going to be running around on our streets.

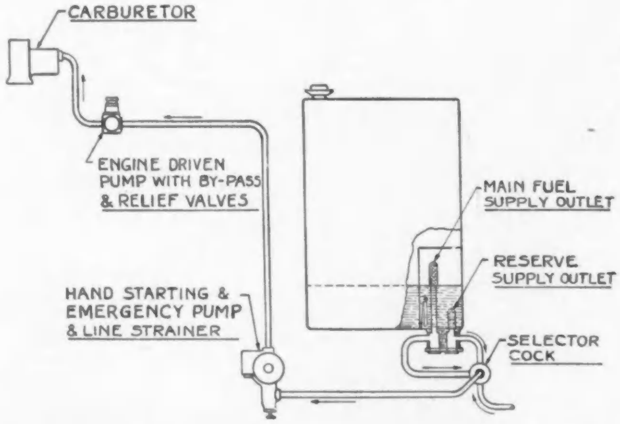
New HIGH-ALTITUDE FUEL S

THIS paper deals in a general way with a brief history of the work that has been done in the past on aircraft fuel systems to adapt them for high-altitude flight, the reasons for failures that developed, some of the physical aspects of aircraft fuel at high altitude, and a brief description of the Thompson Booster system.

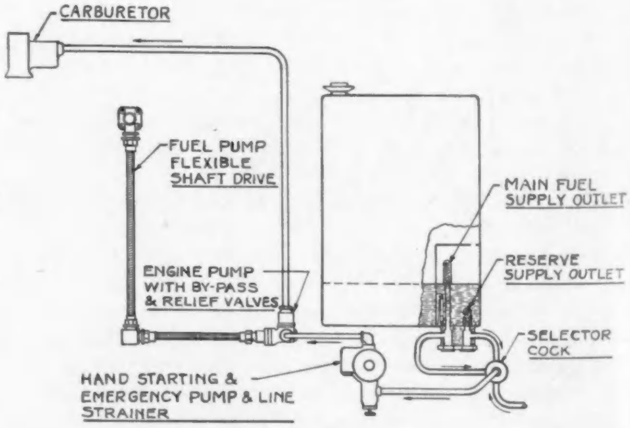
The discussion of the physical behavior of aviation fuel at high altitude is based upon observations made during many laboratory tests. Observed behavior of fuel systems in laboratory altitude simulations is described and differences between this and apparent behavior during actual flight are explained. Due to these differences, it is pointed out, no direct correlation between laboratory simulations and actual flight performance can be made at the present time. Until this handicap can be overcome, prediction of fuel-system performances must be based upon comparison between simulation tests some of which are simulations of satisfactory fuel systems.

The Thompson Booster system was evolved as the result of the application of analytical reasoning to the fuel systems that preceded it. The booster unit is a modified centrifugal pump, attached directly to the fuel tank and driven by an electric motor. Its function is to prevent entrance of released vapor and air to the fuel line leading to the fuel pump on the engine. It also maintains sufficient pressure in this line to prevent additional release of air or vapor.

OWING to the necessity of preparing this paper in a very brief time, it was considered best not to attempt a detailed technical discussion, but rather to limit ourselves to an account of the fuel systems that previously have been tried, together with a brief statement of some of the more important and significant test observations that have been made in our study, and a description of the best system thus far found as the direct result of experimental work that has been in progress for the past two years. Another, and perhaps more compelling, reason for deferring technical discussion is the complexity of the subject and the need for extending the research now in progress to the point where it is hoped that most of the phenomena encountered may be resolved to some rational basis that will make it possible to use the derived data more directly in altitude fuel-system problems. At the present time, we are faced with a mass of test and flight data which, from any other than a restricted practical viewpoint, is indeterminate. They may not, therefore, be applied in any general way to the problem but are limited to the precise arrangement used in the test. As often as not, the behavior of aircraft fuel systems at high altitude has not followed previously applied theoretical analysis, and reasons for departures have not been clear. However, test and flight data have now accumulated to the point where reasonably accurate predictions of fuel-system behavior can be made, notwithstanding the lack of correlating data that would permit rational analysis. Such predictions are based upon data obtained in simulation bench tests by methods which it is anticipated may be discussed in a later paper. It may be pointed out here, however, that the results of these simulated bench tests cannot, with present experimental data, be directly correlated to actual flight performance. Therefore, altitude flight performance predictions must be based upon comparison of many simulation tests, some of which are known to be simulations of fuel systems with a known



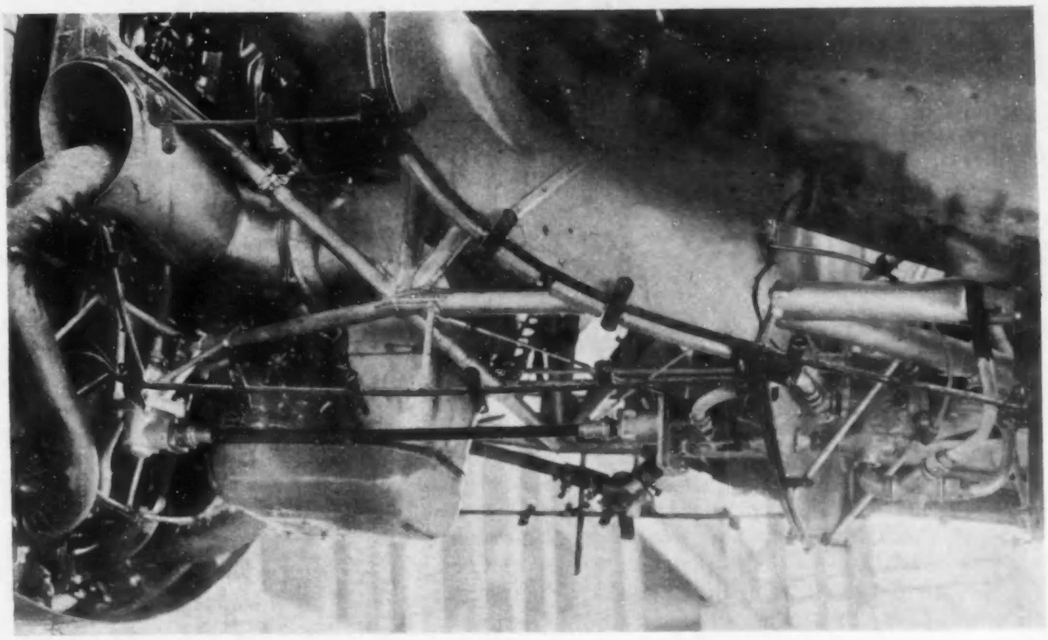
■ Fig. 1 – Simple fuel system with direct engine-driven pump



■ Fig. 2 – Diagrammatic sketch of simple fuel system with lowered pump driven by flexible shaft

SYSTEM for AIRCRAFT

by W. H. CURTIS and R. R. CURTIS
Thompson Products, Inc.



■ Fig. 3 – Actual installation of simple fuel system with lowered pump driven by flexible shaft, shown in Fig. 2

performance in actual flight at the required altitude. It is hoped that this method eventually can be improved.

The altitude problem in aircraft fuel systems is essentially one of vapor lock. Under some conditions trouble may be encountered at relatively low altitudes. Vapor lock occurs when the ratio of vapor to liquid fuel becomes so great that the fuel pump, built to handle liquid only, ceases to deliver the required fuel pressure to the carburetor. This vapor may, in the sense used here, consist of a mixture of air, fixed gases, and fuel vapor. It is formed throughout that portion of the fuel system that lies on the suction side of the fuel pump when pressure is sufficiently depressed, or temperature raised at a sufficient rate.

The history of the work that has been done in the solution of the high-altitude fuel-system problem is a long one. If the performance of airplanes had not gone through such a rapid advancement, it is possible that a better solution might have been found quite early in the period of development. But each new advancement brought on new fuel-system problems which always served in some way to impede progress in the matter of obtaining high-altitude performance. It is true, also, that altitudes above 25,000 to 30,000 ft were not considered essential until quite recently. So, we have the situation wherein performance has not only been on the upgrade but where it was expected to be retained to much higher altitudes. For instance, as an example of higher performance, average rates of climb to 25,000 ft have been tripled during the past ten years. This fact alone is the cause of many new problems.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 7, 1941.]

Commercial and military airplanes that are not to be flown to altitudes above 16,000 to 18,000 ft above sea level are equipped with simple fuel systems in which the fuel supply is below the level of the fuel pump during normal flight altitude of the airplane. Thus the pump is above the hydraulic gradient. A conventional diagram of this system is shown in Fig. 1. This system, for reasons that will become clear later in this discussion, is not satisfactory for high-altitude flight.

To the best of our knowledge, the first device to assist the fuel pump at high altitude was that of applying air pressure to the fuel tank. It is believed that this was first used in 1916, by the British, in their DeHaviland airplanes. A special air pump was mounted on the engine to supply the required pressure. This idea, in various forms, has persisted to this day, notwithstanding the extreme hazard involved in its use. It works perfectly at any altitude, provided nothing goes wrong with the rather delicate pressure controls that must be used to prevent damage to the fuel tanks from over-pressuring or excessive vacuum. It is not permitted on commercial airplanes. At present, to a limited extent only, it is employed in one aviation division of our own Government. We have no knowledge of its use in any of the types of airplanes being used in the present European war.

What may be considered as the next step in the high-altitude fuel system problem was the idea of locating the fuel pump below the hydraulic gradient and driving it by means of a flexible drive shaft extended thence to the engine. A typical system of this type, in diagrammatic form, is shown in Fig. 2. Fig. 3 is reproduced from a

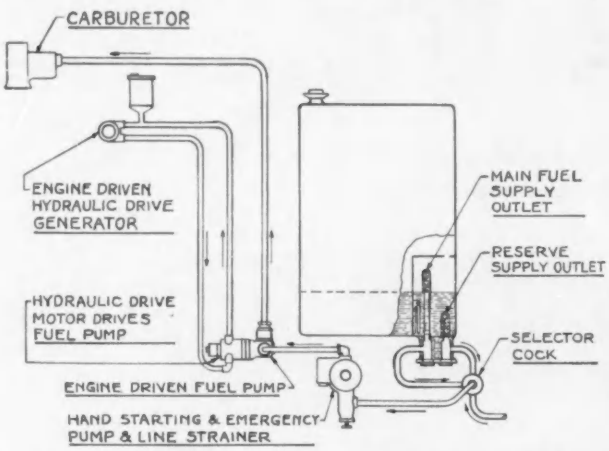


Fig. 4 – Diagrammatic sketch of fuel system with lowered engine pump driven by hydraulic means

photograph of an actual installation. This device, records indicate, was first used in 1918 on a Liberty-engine-equipped LePere airplane built by Packard. Its use was limited to military airplanes and was continued to a considerable extent until about 1937. In general, this system was not very satisfactory, the reasons being both mechanical and functional. Mechanical failures appeared to be induced by excessive torsional whip in the flexible driving member which, it was thought may have been aggravated by the pulsating action of the vane-type fuel pumps that came into use during the period of this development. Doubtless, this whip was also in some measure caused by variations in angular velocity in the drive shaft on the engine. It is also known now that excessive whip occurs when functional failure, due to vapor lock, makes its appearance in the fuel pump. However, this functional failure was not observed until late in the life of the development. Without doubt, it was brought on by the higher performance requirements of the later airplanes for reasons that will be made clear later.

Mechanical failure, therefore, appears to have been the chief reason for extending the search for a satisfactory fuel system into other channels, inasmuch as this step actually was taken before it was known that functional failure of a fuel pump could occur when the pump was below the hydraulic gradient.

This new research was started in 1931, when it was proposed to install the fuel pump close to and below the fuel tank and drive it by an hydraulic motor, the power for which was to be furnished by an hydraulic pump mounted on the engine. Diagrammatically, this arrangement is shown in Fig. 4. Fig. 5 is reproduced from a photograph of an actual installation. The advantage of this system, it was argued, rested chiefly in the fact that the fuel pump could be installed closer to the fuel tank without in any way adding to the difficulty of driving it, inasmuch as distance from the hydraulic pump on the engine involved only length in stationary hydraulic piping. Thus, it was claimed, both the mechanical shortcomings and placement limitations of the flexible drive system would be overcome. Work with this system, now known as the hydraulic fuel-pump drive, continued with some degree of success until 1940, when it was discovered that standard fuel pumps, notwithstanding their position below the hydraulic gradient, could and did fail functionally at high altitude under conditions that could be encountered in service. The most serious of these conditions was fuel at elevated temperature such as might be found in either mobile refueling equipment or the airplane itself, following exposure for any appreciable time to direct summer sunlight; for, in modern high-performance airplanes, it has been found that very little temperature drop in the fuel occurs during the climb to high altitude. This accounts for the extraordinary amount of vapor that is released during the climb and at altitude. Early vapor lock naturally results.

Hydraulic troubles of a functional nature were also encountered in service tests. Moreover, by reason of its complexity, high vulnerability, and excessive weight, it became less and less desirable as the performance of the airplanes was raised. Thus, so many defects and disadvantages developed that it became not only advisable but quite necessary to find some other and simpler means of assuring an adequate supply of fuel to the engine at high altitude. This eventually led to the development of the Thompson Booster system.

Before taking up the details of the Thompson system, it may be well to discuss, in a general way, some of the

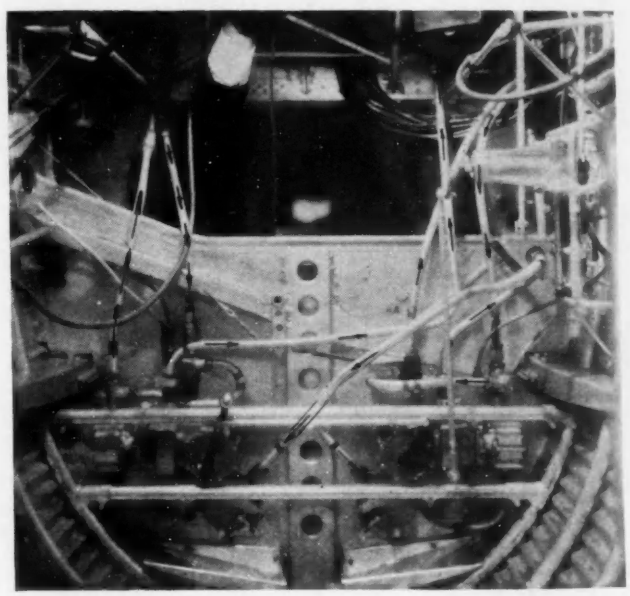


Fig. 5 – Actual installation of fuel system with lowered engine pump driven by hydraulic means, shown in Fig. 4

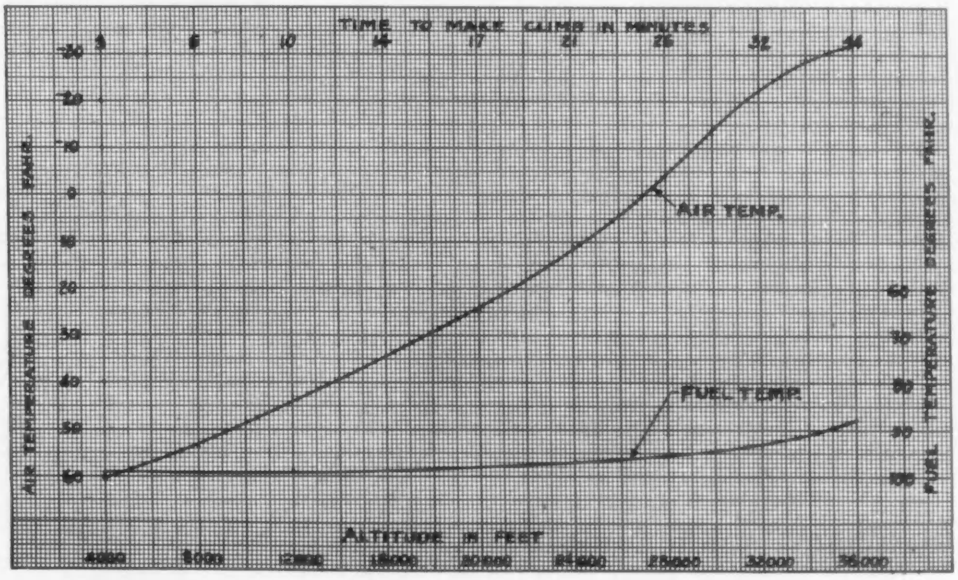
fuel-system problems that are encountered in high-altitude flight. Aside from the few mechanical difficulties that may be expected in any new arrangement of mechanism, the peculiar physical characteristics of the fuel present the real problem. In fact, the primary research is being directed toward finding out what happens to aviation fuel when it is taken to high altitude in an airplane fuel tank, where it happens, and what to do about it. And, even with a fair knowledge of the fuel behavior, what to do about it heretofore has been a real stumbling block.

From a physical viewpoint, aviation fuel, and motor gasoline as well, may be considered to be a stable solution of air and fixed gases in a fluid that is composed of a chain of hydrocarbons some of which may boil, at standard sea-level barometric pressure, at a temperature as low as 100 F. The air is atmospheric air and the fixed gases

are generally combustible hydrocarbons. Both are held by absorption, or true solution. The total volume of gases and air so held may exceed 12% at 60 F temperature and sea-level barometric pressure. Any change of either temperature or pressure renders the solution unstable, and it will either absorb or release air depending upon the direction of the change of temperature or pressure, or both. Thus, if we have a stable fuel to start with, which is in free contact with air, absorption of additional air will occur with increase of pressure or decrease in temperature while release of air and fixed gases will occur with decrease in pressure or increase in temperature. If the rate of change in pressure or temperature is slow, the fuel may remain stable with respect to the air-liquid solution inasmuch as some time is afforded for the absorption or release of the air or gases. But, if the rate of pressure or temperature change is rapid, it appears to bring about an unstable condition wherein the absorption or release of air and gases will continue for some time after the terminal temperature and pressure have been reached. In the modern airplane the temperature of the fuel changes very little during the time between take-off and arrival at optimum altitude in official flight tests. One example of this is shown in the graph in Fig. 6. The effect of the rate of temperature change may, therefore, be neglected. The initial temperature at the time of take-off is, however, quite important for, the higher the temperature, the higher the pressure, and therefore the lower the altitude at which boiling will occur. Thus it will be seen that the initial temperature of the fuel, coupled with the rate of pressure change, appear to be the factors that affect the physical behavior of the fuel during a climb to high altitude. And, inasmuch as the change of pressure is minus, and its rate is related directly to the rate of climb of the airplane, it can be seen readily why recent high-performance airplanes with exceedingly high rates of climb serve to increase the difficulty of the problem. So much air and gas are released that, in ordinary fuel systems, vapor lock occurs at relatively low altitudes.

Further, there is reason to believe that a well-defined critical occurs when the pressure is reduced to the point where boiling begins and that this is not entirely due to vapor produced by the boiling action. If we use the well-known CO_2-H_2O solution as a parallel, it seems safe to assume, where the reduction in pressure has been fairly rapid, as in a modern airplane climbing to altitude, that much of the air remains in solution until boiling actually begins, at which time it is released almost completely in a comparatively short period of time. Some tests, however, have led us to believe that the amount of air that may remain in solution at the time boiling begins depends in some measure upon the vibration to which the fuel may be subjected, and that the effect of vibration is to reduce the amount of air or fixed gases that are discharged during early boiling. If this is true, then vibration during the period of pressure reduction must accelerate the release of air and gases, thus causing the fuel to enter the boiling phase in a more stable condition than would otherwise be the case. And herein appears to lie one of the principal differences between simulated flight tests and actual flight. Vibration is absent in the simulations and is certainly present to some degree in the airplane. A sharper critical has been observed in simulations than appears to exist in flight. Moreover, a marked temperature drop occurs in simulations as the test proceeds through the critical, while this has so far been absent in actual flight. Also, the total temperature drop observed in simulations to a given altitude has always exceeded that experienced in flight tests. This strongly indicates that less boiling occurs at altitude in airplane fuel tanks than in simulated systems. The causes for these differences are not known but it is strongly suspected that vibration and its probable resultant – less instability in the air-liquid solution – may have something to do with it. The research has not progressed to the point where it has been considered advisable to include vibration in the simulations, but it is hoped that this can be done in the near future. If the correct simulation of vibration can be found, there is reason to believe that it may be possible, with proper apparatus, to develop rational data in laboratory altitude simulations that will correlate directly to altitude flight performance.

In addition to the known physical characteristics of the fuel that affect its behavior, it has been found that other variables may serve to prevent proper operation of the fuel system at altitude. It has been pointed out that very early in the history of the airplane the idea of placing the fuel pump below the hydraulic gradient was adopted under the impression that functional failure could not occur with



■ Fig. 6 – Curves of air and fuel temperatures taken in flight

the pump in this position, and that this reasoning was later found to be faulty. However, it has been found that the altitude at which failure occurs depends upon how far the pump is below the hydraulic gradient or, in other words, the head of fuel on the suction side of the pump. The greater the head, the greater the altitude at which failure will occur. Unfortunately, airplane structures are such as to place severe limitations upon the vertical location of the fuel pump, and it therefore has been impossible to take full advantage of this characteristic.

Another variable that we have reason to believe can affect operation of fuel systems at altitude, and the last one that we shall deal with here, is the volume of fuel in the suction side of the system, or more specifically, in the fuel tank. There is reason to believe that less trouble is encountered when fuel is being taken from large tanks than is the case when it is being taken from small tanks, the head being the same in each case. Some differences have been noted in laboratory simulations but it is not known if this will hold true in actual flight. Whatever difference in operation there may be, it is probably less serious than other characteristics that we have discussed.

The reasoning that led to the development of the Thompson fuel system was, in reality, very simple. It was obvious, of course, that functional failure of the fuel pump was being caused by air or vapor, or both, passing through the suction line along with liquid, and thus finding its way into the pump. It seemed obvious also, inasmuch as failure occurred in pumps that were installed below the fuel tank and therefore below the hydraulic gradient, that much of the air and vapor must be released in the fuel tank and that, with high fuel rates of modern airplane engines, some of it was finding its way into the suction line that led to the fuel pump. Moreover, it appeared likely that the action of the fuel pump itself caused the formation of additional vapor at the pump inlet. A prime requisite therefore was the elimination of vapor at the fuel pump as well as in the fuel suction line leading thereto. Other considerations were that the system must be simple, with the units easily replaceable. Reliability was the third and most important requirement. Thus we evolved the following fundamentals:

1. The fuel pump must be restored to its former position on the engine, where it had proved to be an exceedingly reliable unit, in so far as mechanical performance was concerned.

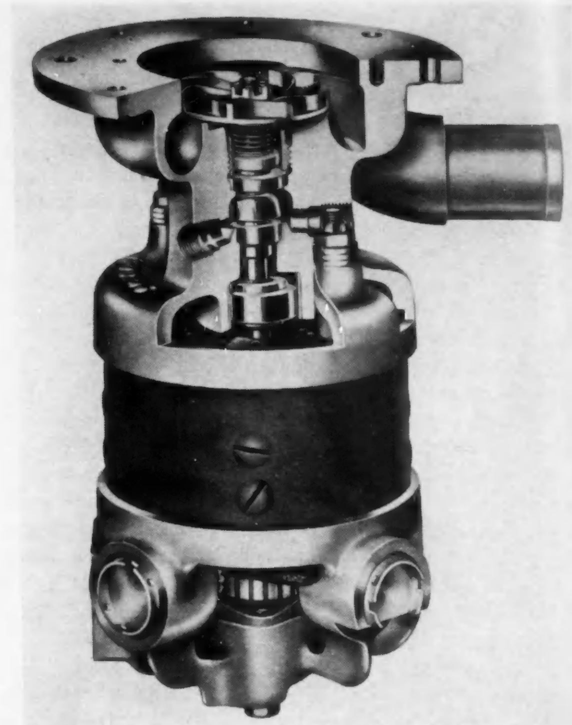
2. A means must be found for supplying vapor- and air-free fuel to the fuel pump under such conditions that its stability of air-liquid solution could not be upset by pump action to the point where release of additional air or vapor would occur.

3. A suitable mechanical unit must be devised for doing the job outlined in Requirement No. 2. It should be independently driven, preferably by an electric motor, and must be completely free of delicate controlling devices.

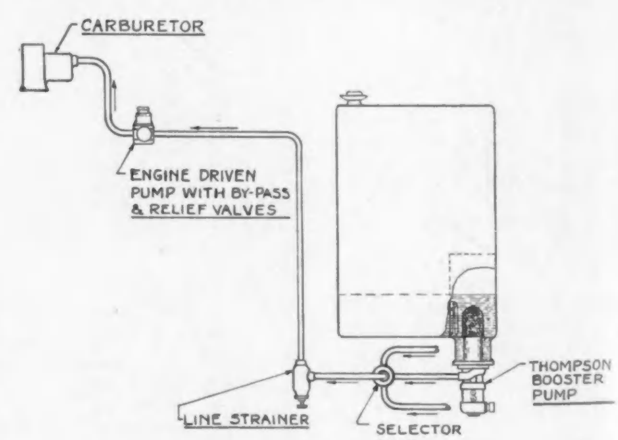
With this statement of fundamentals before us it required no great amount of analysis to point to some form of a centrifugal pump as offering the simplest and, possibly, most reliable means for doing the job. It also seemed to be more adaptable to the modifications that might be required than any other type considered. Once the selection was made, the unit almost immediately became known as a booster and this term is retained in the official name – Thompson Booster Pump.

After considerable experimentation, the form shown in Fig. 7 was found to be the best, and all modified types are now based upon it. One form of the simplified fuel system in which it is used is shown in Fig. 8. Mechanically, this unit is exceedingly simple. The motor is explosion-proof, thus removing practically all fire hazard. It will be noted that it is mounted, with motor end down, directly to the under side of the fuel tank. While the illustration shows it in a vertical position, angular mounting to within 20 deg of the horizontal is equally satisfactory in some instances.

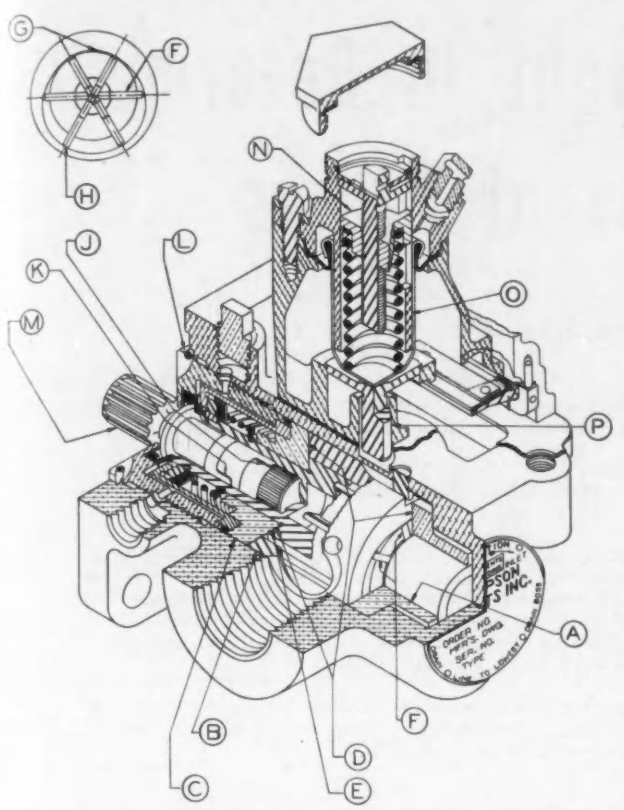
In this simplified fuel system the present function of the booster is three-fold. A fourth function is being added. At present, by centrifuge action, it prevents flow of re-

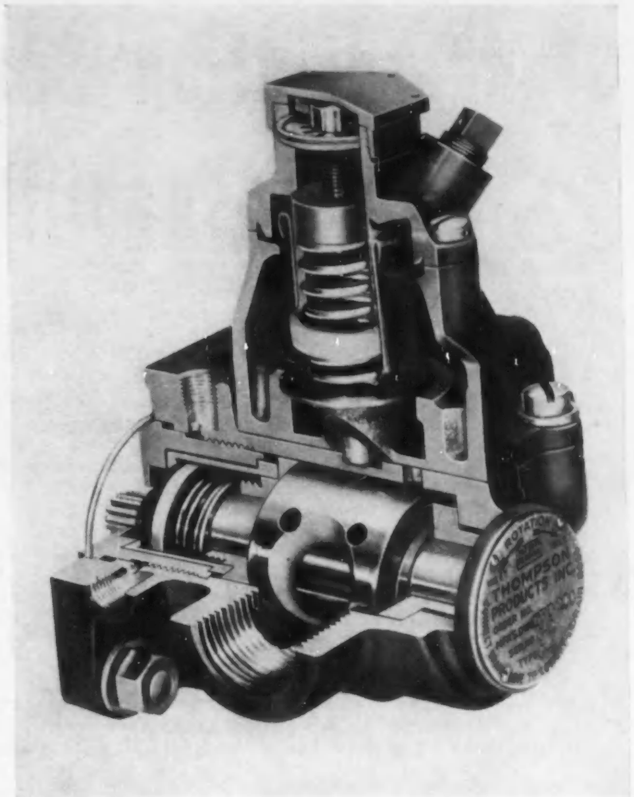


■ Fig. 7 – Thompson booster pump



■ Fig. 8 – Fuel system with Thompson booster and engine pump





■ Fig. 9 (left) – Cut-away sketch and Fig. 10 (right) Cut-away photograph of Thompson engine-driven aircraft fuel pump

leased air or vapor through its impeller at all rates of fuel delivery required in modern airplanes. The air or vapor thus rejected finds its way upward through the fuel in the fuel tank and is discharged through the tank's vents. With full fuel tanks, it is thought that much of it is taken back into solution during its passage through the fuel, inasmuch as the loss of fuel by reason of booster operation appears to be no greater than with it out of operation.

This action of the booster, by rejecting through centrifuge action all air and vapor that may appear at the throat or pump inlet, assures vapor-free, or solid fuel in the line leading from the booster to the fuel pump. Moreover, and this is the second of the functions mentioned, the fuel in this line is held at sufficient pressure, by the pumping action of the booster, to prevent the release of additional air or vapor as the fuel passes through the fuel pump. Close regulation of discharge pressure from the booster is not necessary. This is a function of the fuel pump on the engine, which is equipped with a specially balanced relief valve for this purpose. This relief valve so functions that the fuel-pump discharge pressure is independent of, or not affected by, either inlet depression or inlet pressure within the normal operating ranges of modern carburetors. Thus, an even and accurately controlled pressure at the carburetor is maintained notwithstanding considerable discharge pressure variation from the booster. A cut-away sketch of the Thompson engine-driven fuel pump is shown in Fig. 9, and a cut-away photograph in Fig. 10.

The third function of the booster was not anticipated at the time of its development. It is that of starting the engines. It has been found that by using the booster instead of the hand pump to prime and pressure the system, engines that are equipped with modern high-pressure carburetors start much easier and with less exhaust firing.

The fourth function, now being developed, is that of taking over the duty of the emergency hand fuel pump. When this project is completed, it will be possible to eliminate one piece of equipment. This is to be done by increasing the operating speed of the booster so that its discharge pressure will be sufficient to operate the engine in the event of failure of the engine-driven fuel pump. At present the details of this development are in confidential status and may not be disclosed. However, it is hoped that release may be in order when a more detailed discussion of this development is held.

Basically, this system appears to be sound. As service experience is gained it is anticipated that some mechanical alterations may be indicated. There has, however, been a very large amount of service testing during which the major defects have been spotted and remedied. Further corrections, for this reason, are likely to be minor in character.

Functionally, the system has no failures recorded against it. Numerous test flights, many to altitudes of more than 36,000 ft, have disclosed no indication of failure notwithstanding several take-offs with fuel heated to abnormal temperature. The graph shown on Fig. 6 is an actual temperature record of one of these flights.

■ Acknowledgment

We wish to acknowledge the assistance of F. W. Heckert, who is associated with us, in the preparation of the illustrations.

Standardization Sought in Determining Hardenability of Steels

*A Symposium**

IT IS possible for two steels of the same specification to show entirely different degrees of hardenability. Type of chemistry, grain size, and many factors in the making of steel affect the depth of hardness.

For several years efforts have been made to standardize on this important property in production steels by the establishment of a practical method of checking, which would be acceptable to both the makers and consumers of steel.

The ability of steel to harden and the depth and decrement of hardenability are important engineering requirements. This hardness is correlated with tensile and torsional values. Hence, in any part where strength or resiliency are essential factors, this relation is extremely valuable to the engineer and metallurgist. For example, steels which harden too deeply might become excessively brittle, and others, which show shallow hardening characteristics, might fail due to fatigue or low strength. There are applications where both deep-hardening and shallow-hardening steels are desirable.

The authors of the four succeeding papers comprising this symposium have all been very prominent in the research on this subject, have contributed valuable information on the basic principles involved, and have done much toward standardizing the measurements of hardenability.

■ ■ ■

Use of Hardenability Tests

for Selection and Specification of Automotive Steels

by A. L. BOEGEHOLD

Research Laboratories Division, General Motors Corp.

THE degree of hardness obtained in any steel as a result of heating and quenching is dependent principally upon how fast the steel is cooled during the quenching operation. Many factors will modify the hardness obtained at some one rate of cooling. Among these factors are grain size, carbide condition, and alloy content. But, for any steel of a particular composition, grain size, and carbide condition, the rate of cooling during quenching and no other factors (except possibly the indeterminate one of cooling stresses) determines the hardness produced.

Since this fundamental dependence of hardness exclusively upon cooling rate exists, it should logically be the basis of classifying steels both for specification and selection purposes. When a steel is described in terms of certain cooling rates necessary to produce hardnesses between 10 and 60 Rockwell C, it is defined on a basis that is applicable to all degrees of hardenability, all types of quenches, and all section sizes.

A standard hardenability test procedure is necessary for determining the relationship between hardness and cooling rate for each steel. The requirement that must be met by a standard hardenability test procedure is that it shall furnish the relationship between hardness and cooling rate applicable to a wide variety of steels, section sizes, and cooling rates and do it with a minimum amount of work. This general statement may be divided into a number of individual requirements as follows:

1. The hardenability test bar shall be inexpensive and of a size that can be cut from any size stock that may be purchased for manufacturing parts.

* [This symposium was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1941; the four papers which, with their discussion, comprise the symposium are: "Use of Hardenability Tests for Selection and Specification of Automotive Steels," by A. L. Boegehold, General Motors Research Laboratories; "A Method for Hardenability of Small Sizes," by F. E. McCleary and R. Wuerfel, Chrysler Corp.; "Determination of Specific Hardenability of Shallow-Hardening Steels," by O. V. Greene and C. B. Post, Carpenter Steel Co.; and "Correlation between Jominy Test and Quenched Round Bars," by M. Asimow, W. F. Craig, and M. A. Grossmann.]

2. Preparation for making hardness tests after hardening shall be inexpensive and as few as possible hardness determinations should be made.

3. One test bar shall yield hardness results representing a wide range of cooling rates.

4. The different speeds of cooling in the test bar must be so located as to permit ready and accurate measurement of those cooling speeds.

With requirements Nos. 1, 2, and 3 as a guide, the "end-quench" or Jominy hardenability test was developed. This test bar fortunately met requirement No. 4, a necessity which became apparent as the work progressed.

Three types of test bars all of the end-quench type are necessary to cover the desired range of bar sizes and cooling rates. These are shown in Fig. 1, and will be discussed later in this paper.

A number of methods for measuring hardenability have been proposed, each one embracing a limited field of cooling rates and hardenabilities and each one proposing to define the steel in certain empirical or arbitrary terms. One of the earliest tests, the Shepherd test, rated the steel in terms of the hardness penetration in a ¾-in. round quenched in brine solution. Burns, Moore and Archer[1] devised an arbitrary formula for rating steel in terms of the shape of the hardness penetration curve in a 1-in. round.

Asimow, Grossmann and Urban[2] proposed to describe hardenability in terms of the size of round bar that would just harden to the center when quenched with the ideal quench.

Queneau and Mayo[3] have proposed that hardenability be expressed in terms of a line showing the depth to which a steel will harden in various sized round bars.

All these methods of expressing hardenability are concerned with hardness penetration in bars of one size or another, but do not afford a means of interpreting hardenability test-bar results in terms of results to be expected in complicated machine parts made from the same steel.

The "end-quench" or Jominy hardenability test is the only one so far proposed that allows us to measure accurately a wide range of cooling rates and thus provides us with the fundamental relationship between cooling rate and hardness. With the aid of this relationship it is possible by substitution of cooling rates in place of corresponding hardness to determine easily the cooling rates at any place in any shaped article, be it round, square, slab, cone, or complicated machine part.

Knowing the cooling rate in an object enables us to find what hardness to expect in that object when made from any steel, by referring to the hardness-cooling rate curve for that steel.

The steps in the procedure of predicting hardnesses obtainable in various automobile or machine parts by using hardness determinations on the Jominy hardenability test bar will be made clear by illustrating the application of the method to a specific object. For the purpose of making such predictions the most useful arrangement of information obtained from the Jominy hardenability test bar is in the form of a curve showing the relation between cooling rate and hardness. *This curve hereafter will be known as the H-CR curve.* It is produced by merely substituting cooling-rate values in place of values for distance from the quenched end of the test bar. Fig. 2 shows cooling rates at different distances from the quenched end for the standard test bar and for the Type L test bar.

THE fundamental relationship between cooling rate during quenching and hardness produced in steel is pointed out in this paper. The requirements for a hardenability test bar for determining this relationship for a wide variety of section sizes and steels are given. Some of the methods that have been suggested for testing hardenability are discussed briefly, pointing out that the end-quench specimen is best suited for obtaining this fundamental relationship between hardness and cooling rate.

The use of the hardness-cooling rate curve for determining cooling rates in objects to be hardened and also for predicting hardnesses that will be obtained in such a part depending upon the steel used is described. A method of specifying hardenability of steels in terms of hardness cooling rate curves is described and the procedure outlined for determining what the hardenability limits should be. Application of this general procedure for selecting and specifying steels is illustrated in connection with a specific automobile part.

A method of interpreting hardenability information obtained from various hardenability tests in terms of hardness cooling rate curves is explained. This translation of hardenability information from any test bar into terms of a hardness-cooling rate curve permits the use of these various test bars for the purpose of predicting hardness in complicated shaped articles. The H-CR curve is the abbreviated name taken for convenience in referring to the hardness-cooling rate curve.

Let us consider the spool-shaped piece shown on Fig. 3 as an object with which to illustrate the proposed method.

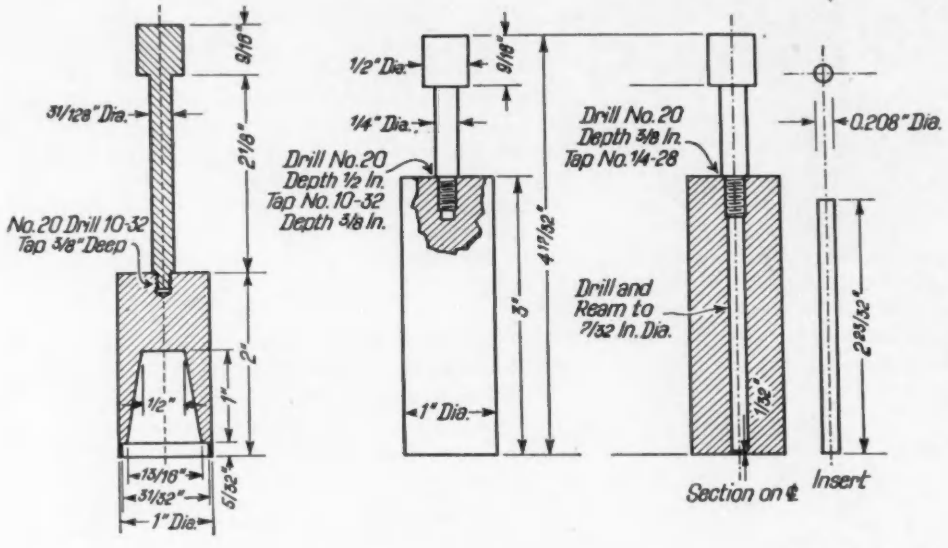
Step No. 1 in procedure for predicting hardness is to determine the cooling speed at various points in the object under consideration. This may be accomplished by making the object out of a low-hardenability steel, quenching the part in oil or water, depending on whichever is to be used in production, determining the hardness at various points in the cross-section, and finding the cooling rates corresponding to these hardness values by referring to an H-CR curve for the same steel as used for the object.

In Fig. 3, marked on the upper half of the cross-section of the spool are shown the hardnesses resulting when the spool is made from 5130 steel and quenched in oil from 1675 F. The H-CR curve obtained from a Jominy test bar made from the same bar of 5130 steel as used for the spool and quenched from 1675 F, is shown at the bottom of the figure. From this curve are picked the *cooling rates* corresponding to the hardness values shown in the upper half of the cross-section of the spool. These cooling rates are

[1] See Transactions of the American Society for Metals, Vol. XXVI, 1938, pp. 1-22: "Quantitative Hardenability; Proposed Standardization Test," by J. L. Burns, T. L. Moore, and R. S. Archer.

[2] See "Hardenability, Its Relation to Quenching and Some Quantitative Data," by M. A. Grossmann, M. Asimow, and S. F. Urban, Symposium on Hardenability of Alloy Steels, American Society for Metals, 1939.

[3] See "Hardenability and Its Designation, the Hardenability Line," by B. R. Queneau and W. H. Mayo, Symposium on Hardenability of Alloy Steels, American Society for Metals, 1939.



■ Fig. 1 – Test specimens for end-quench method of determining hardenability – Usual specimen at center; "L" bar for steels of low hardenability at left; drilled bar at right for steels available only in small sizes

shown in the corresponding positions in the lower half of the cross-section of the spool with iso-cooling rate curves showing the location of equal cooling rates.

This completes the first step consisting of determining cooling rates in the object to be heat-treated. In this step 1675 F was the temperature selected because the intention was to carburize the object sufficiently to provide a hard surface, relying on the hardness developed in the 0.30% carbon core to back up the light case to resist compressive loads. The hardness over a range of cooling speeds obtained from a 1675 F quench with 5130 steel is not the same as would be obtained by quenching from 1550 F, the usual hardening temperature for 5130 steel. Therefore, if the intention were to use 1550 F for the quench temperature, it would be necessary to use that temperature for the Jominy hardenability test bar as well as for the object.

Step No. 2 is to decide what hardness is required in the object, assuming it were to be used as a machine part subject to certain loads and stresses in service. There are several possibilities depending upon the usage of the part. Let us consider three possibilities, illustrating what might be prescribed to withstand service under differing sets of conditions:

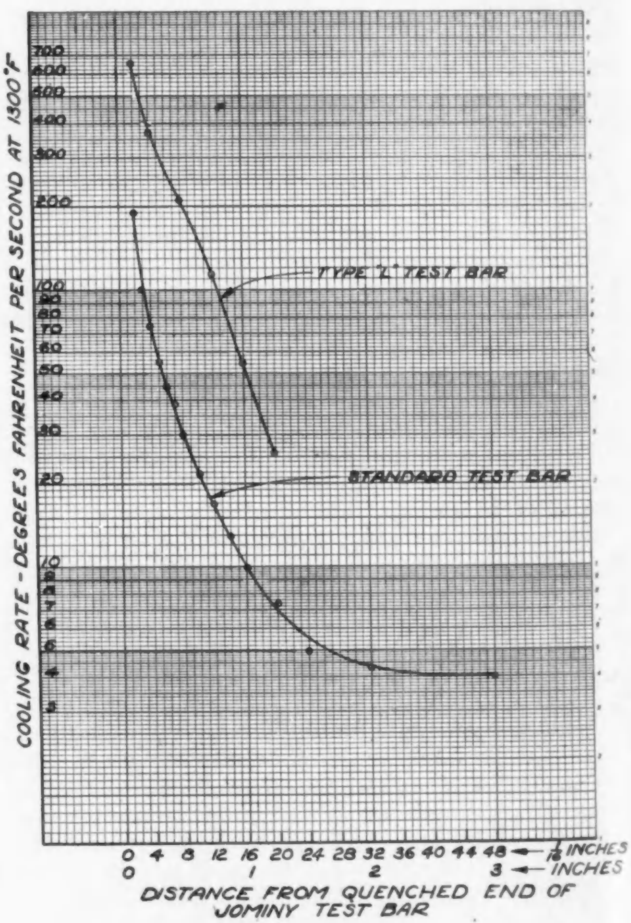
1. The as-quenched hardness must be 50 Rockwell C or more throughout.

2. The as-quenched hardness must be 50 Rockwell C or more to a depth of at least 1/4 in. below cylindrical surfaces of 1 3/4 in. diameter.

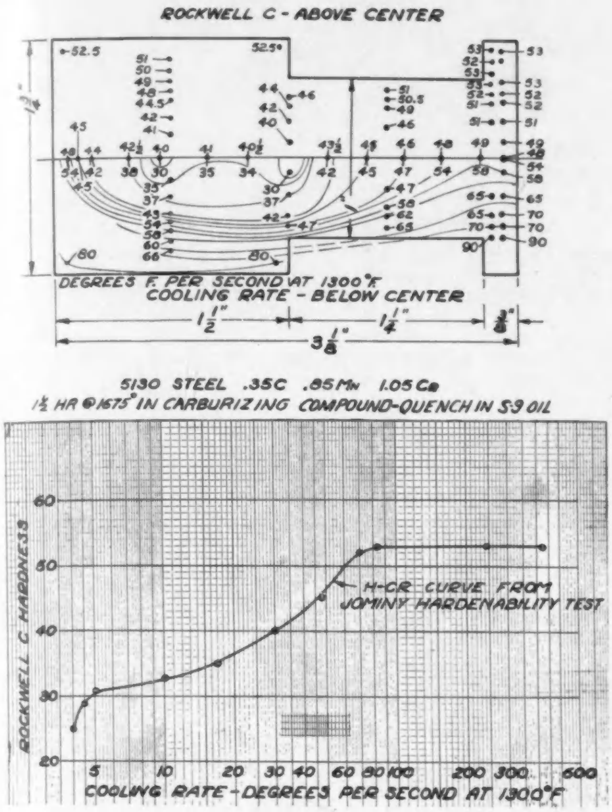
3. The hardness shall be 60 Rockwell C to a depth of 0.025 in. and the core hardness shall be 30 Rockwell C minimum.

Step No. 3 is to examine H-CR curves for a number of different steels to find the ones that meet the foregoing requirements.

As a means of facilitating selection of the steel for each of the preceding three specifications, it would be helpful to draw the H-CR curve dictated by those three specifications. This has been done as depicted in Fig. 4. Curve 1 has been drawn so that, for all cooling rates above 30 F per sec, a hardness of 50 Rockwell C is obtained. This is dictated by the fact that the slowest cooling rate in the spool is 30 F/sec. Any steel having an H-CR curve like Curve 1, or one lying to the left of Curve 1, will meet requirement No. 1 preceding.



■ Fig. 2 – Cooling rates at different distances from the quenched end for the standard test bar and for the Type L test bar



■ Fig. 3 – (Above) Hardnesses resulting in spool-shaped piece of 5130 steel when quenched in oil from 1675 F – (below) H-CR curve obtained from a Jominy test bar made from the same bar of 5130 used for the spool and quenched from 1675 F

Curve 2 has been drawn so that, for all cooling rates above 59 F/sec, the hardness will be 50 Rockwell C or higher. This is dictated by the fact that the slowest cooling rate to a depth of ¼ in. below the 1¾-in. diameter cylindrical surfaces of the spool is about 50 F/sec. This curve is practically the same as that for the 5130 steel used for determining the cooling rates in the spool. This steel then, or any steel having an H-CR curve lying to the left of Curve 2, would meet specification No. 2.

Curves 3, one for the case and one for the core, are the H-CR curves dictated by specification No. 3. With the aid of equal cooling rate curves drawn in Fig. 3 on the cross-section of the spool, it is seen that the line for 60 F/sec comes very close to the surface at the intersection of the 1-in. cylinder and the 1¾-in. cylinder. This then is the slowest cooling rate at the surface, and the steel to meet the specification must attain 60 Rockwell C at any speed above 60 F/sec.

Similarly, the slowest cooling rate in the core is 30 F/sec so the curve for the core steel will pass through Rockwell C 30 at 30 F/sec cooling rate. Any steel having an H-CR curve lying to the left of Curves 3 will meet the requirements of Specification No. 3.

Step No. 4 is to examine the H-CR curves available to see which steels will meet the requirements.

None of the 0.30% carbon steels in Fig. 5 will meet the requirements of H-CR Curves 1 or 2. 4340 and 3240 in Fig. 6; 6145, X3045, 3145 in Fig. 7 all meet requirements of H-CR Curve 1.

Naturally, the cheapest and easiest to machine of these several steels will be the one selected – unless fatigue tests on the finished part indicate otherwise.

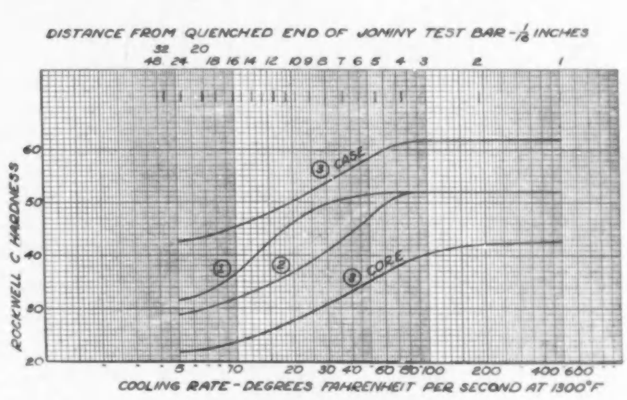
4340, 3240, 3140 and X1340 in Fig. 6, and all but X1045 and 1045 in Fig. 7, meet the requirements of H-CR curve 2.

The case requirements of Specification No. 3 illustrated by H-CR curves No. 3 are met by all but steel 1120 in Fig. 8, but the core requirements are met completely by only 2315 in Fig. 9, and probably sufficiently by 4615 and 4815 on the same chart.

It will be seen that a slight change in the hardness requirements specified for the piece to be hardened makes a great difference in the variety of steels that will meet those requirements. Also, it is obvious that a small increase in carbon in the steels represented in Fig. 9 would change their H-CR curves so that any of them would meet specification 3.

Figs. 10 and 11 show H-CR curves for the carburized case 0.015 in. below the surface and at the surface for the steels listed in Fig. 8.

Any change in any of the factors that influence the rate of heat extraction from the object to be quenched would make necessary a recalibration to determine the extent of change in cooling rates. Change in the quenching medium, either its quality, temperature or rate of circulation, change in the quality or quantity of scale produced by the furnace atmosphere in which the object was heated, change in the time temperature cycle, would require a redetermination of either the cooling rates of the object quenched or the H-CR curve. Whether it is necessary merely to conduct another test on a hardenability test bar to obtain a new H-CR curve or whether a new object must be quenched and explored for hardness, or whether both must be done depends upon what conditions are changed. Any change that affects hardenability of the steel requires a new H-CR curve to be determined. Any change that affects the cooling rate of the object to be quenched requires a redetermination of cooling rates under the new condition. The cooling rate at various points in the hardenability test bar is not affected for practical purposes by change in heating temperature or quality of steel. These changes affect only the hardenability of the steel.



■ Fig. 4 – H-CR curves for steels which will meet sample specifications Nos. 1, 2, and 3

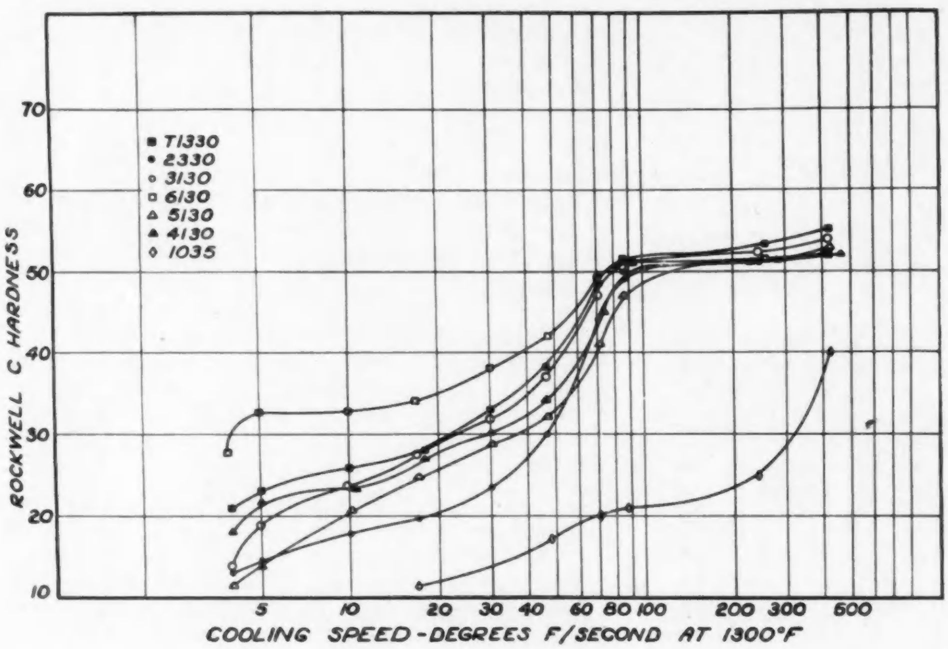
Table 1 contains factors involved in heat treating, classified as to their influence on hardenability of the steel or on cooling rate of the object to be quenched.

It is imperative that the heat-treatment and steel used for the object to be studied for cooling rates be the same as for the Jominy test bar used to obtain the H-CR curve. Any violation of this rule will result in erroneous values determined for cooling rates of the object heat-treated.

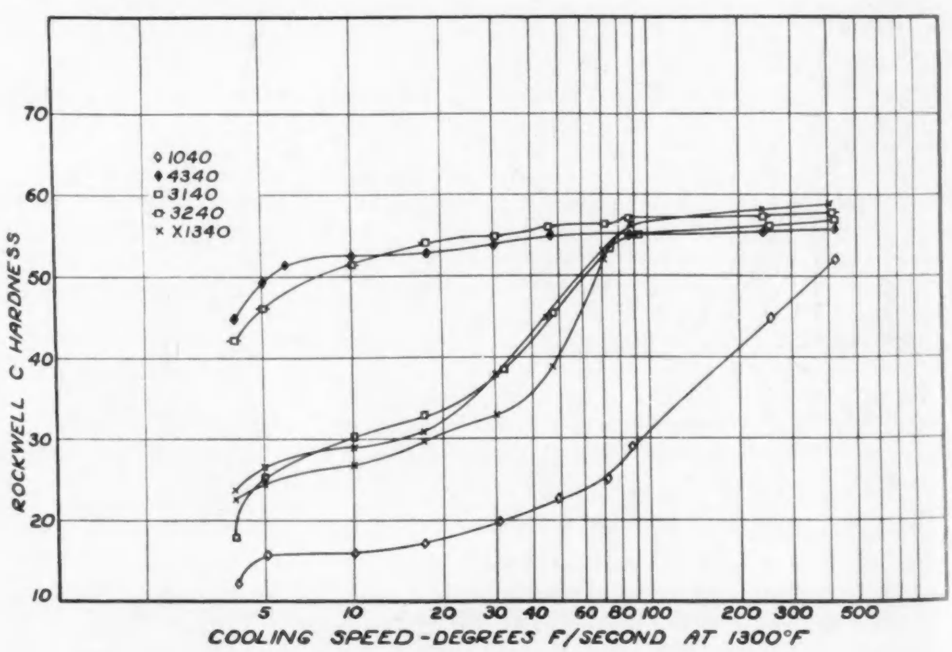
In comparing actual hardness values with hardness values predicted by means of H-CR curves, it must be always kept in mind that all the factors under *A* in Table 1 must be the same for the determination of actual hardness

Table 1

A	B
Factors Affecting Cooling Rate of Object Quenched	**Factors Affecting Hardenability of Steel**
1. Quality of quenching medium.	1. Temperature to which steel is heated (effect on grain size).
2. Temperature of quenching medium.	2. Prior treatment affecting condition of carbides.
3. Circulating speed of quenching medium.	3. Rate of heating below critical temperature (effect on carbide condition).
4. Size and shape of object.	4. Composition.
5. Quantity and quality of scale on surface of object.	5. Grain size.
6. Temperature of object quenched.	6. Carbide condition and little understood factors inherited from steel-making process.



■ Fig. 5 – H-CR curves for 0.30 carbon steels



■ Fig. 6 – H-CR curves for 0.40 carbon steels

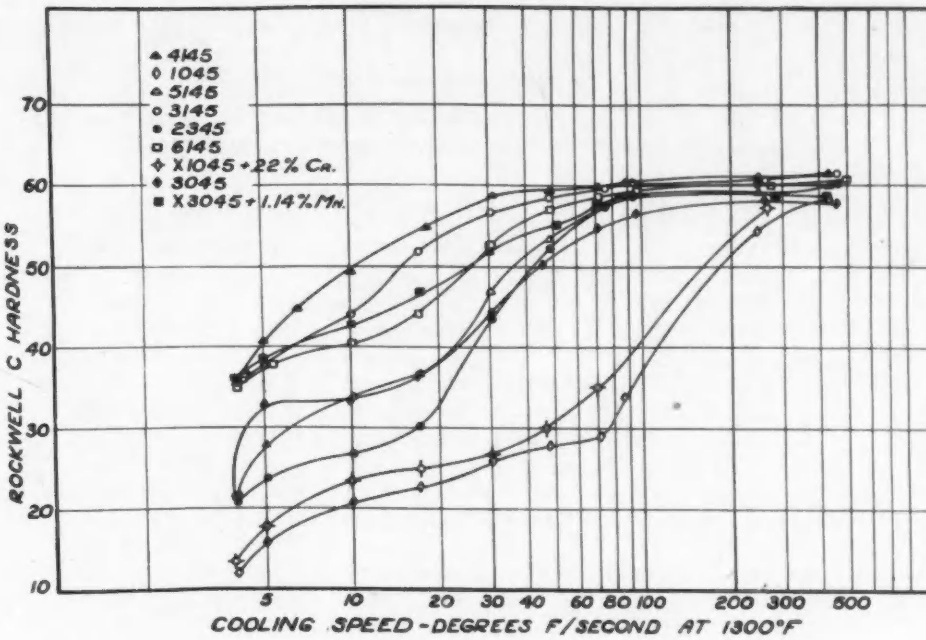
values as they were in the object used for determining the cooling rates in the first place.

Another condition that must be observed is that, in checking predicted hardnesses with actual hardnesses obtained, the steel and heat-treatment used to make the objects must be the same as the steel and heat-treatment used to obtain the H-CR curve used for predicting hardness values. Otherwise the hardnesses predicted will not agree with hardnesses actually obtained when the object is made from another heat of the type of steel represented by the H-CR curve.

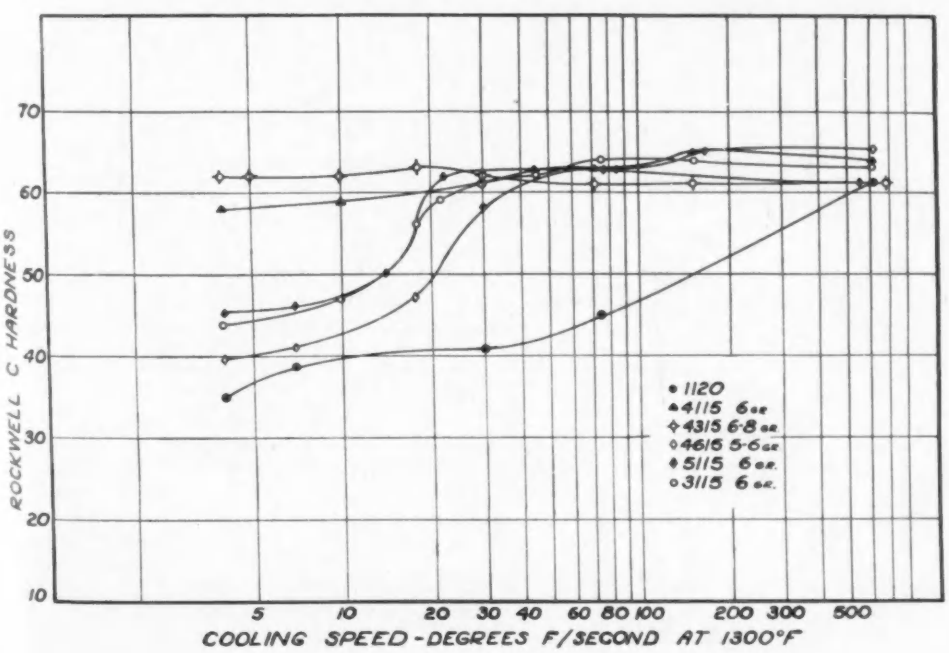
In using the foregoing principles for the selection of materials for automobile parts, if we will substitute the name of the part concerned, be it steering knuckle, crankshaft, or drive pinion, in the place of "object" or "spool" wherever it appears in the preceding text, the procedure will apply directly.

In addition to the use of H-CR curves for predicting hardness obtainable in heat-treated automobile parts, specifications for purchasing steel may be expressed in terms of H-CR values.

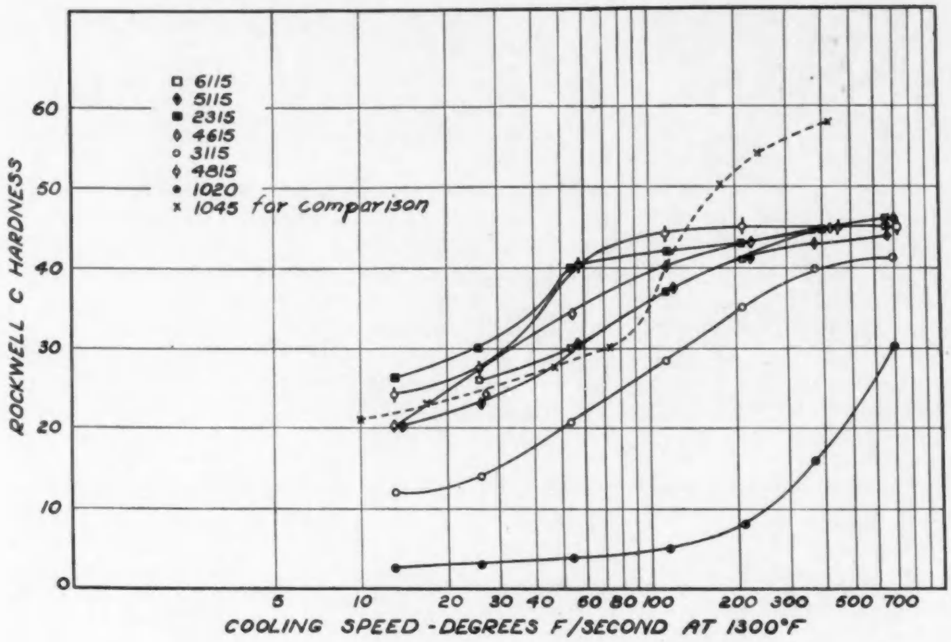
For purposes of illustration let us suppose that the spool-shaped object used in this discussion must have a minimum core hardness between 28 and 32 Rockwell C and a hardness between 40 and 45 Rockwell C to ⅛ in. below the surface of the 1¾-in. diameter cylinders. This means that the steel used must be such that, at 30 F/sec cooling rate, the lowest in the spool-shaped object, the hardness will be 28 to 33 and that at 66 to over 90 F/sec cooling rate (the cooling rates in the ⅛-in. surface layer of the



■ Fig. 7 – H-CR curves for 0.45 carbon steels



■ Fig. 8 – Hardenability of carburized case at 0.025 in. below surface



■ Fig. 9 – Core hardness of carburizing steels

1¾-in. diameter cylinders), the steel will have a hardness between 40 and 45 Rockwell C. The reasons for such a specification need not be considered here as this is a hypothetical case to illustrate how a steel may be specified, given certain hardness requirements in some object to be hardened.

The H-CR limits stated in the foregoing case are shown graphically in Fig. 12, the placing of maximum and minimum allowable limits of hardness at different regions of the spool-shaped object makes the specification much more restricting than sample specifications 1, 2 and 3 represented by Fig. 4 wherein only minimum hardnesses were specified.

The only steel for which H-CR curves are shown in this report that will satisfy the requirements of the limits shown on Fig. 12 are 4815 and 2315 in Fig. 9.

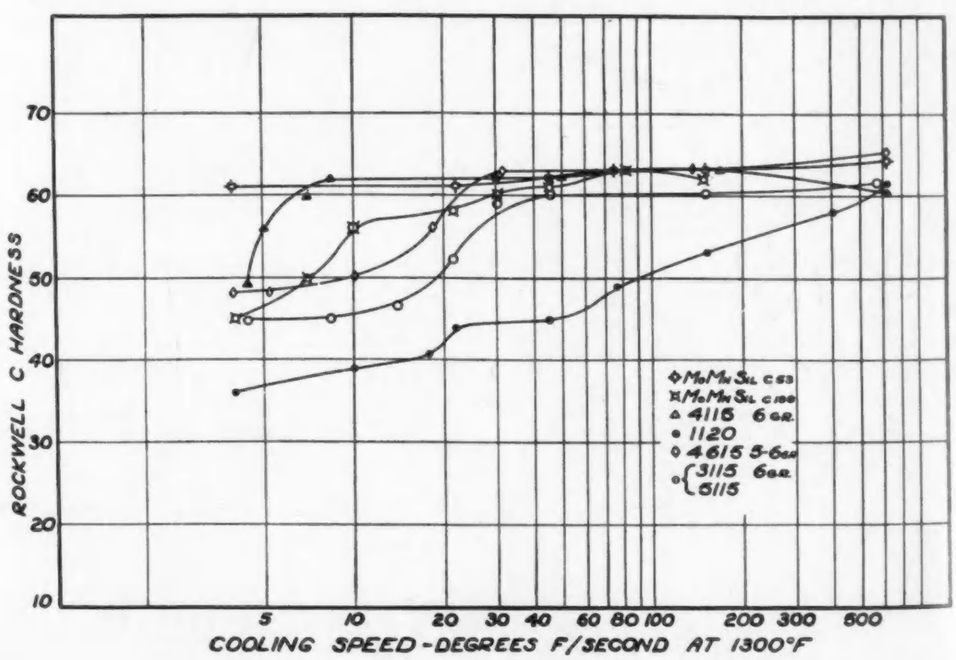
The diagonal part of the H-CR curves in Fig. 12 limit the hardenability, and the horizontal part of these curves limit the maximum and minimum hardness and, therefore, the maximum and minimum carbon content allowable in the steel.

The specification for this steel may be expressed in two ways – either in terms of the Jominy bar distance or in terms of cooling rates.

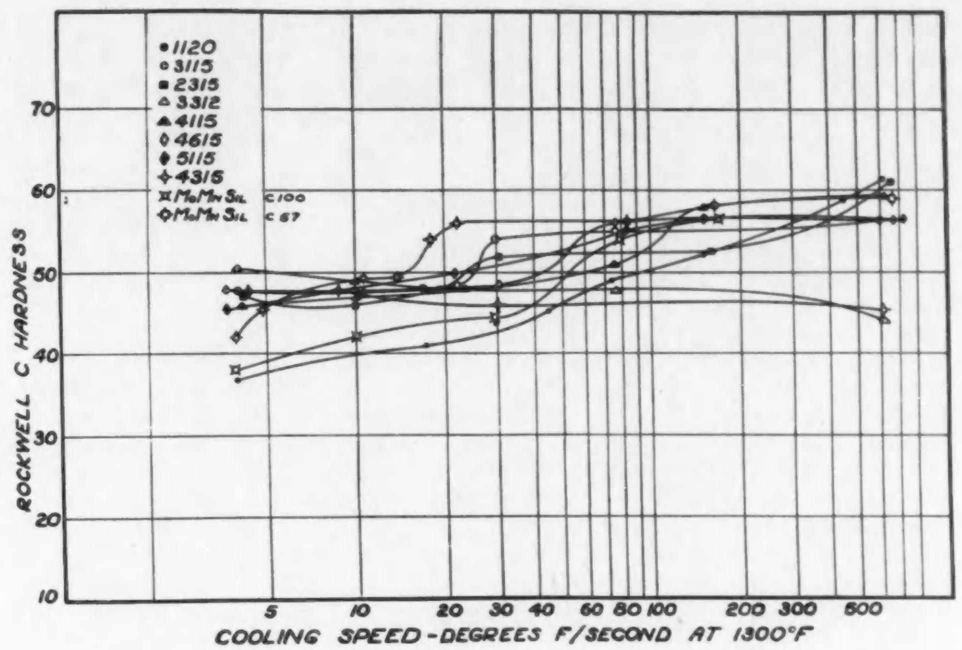
The first method will provide a specification in the following terms – Rockwell C 28 to 33 at J8 – Rockwell C 40 to 45 at J4½ to J2. (The numbers prefixed by J denote sixteenths of an inch from the quenched end.)

The second method of specification will be – Rockwell C 28 to 33 at 30 F/sec, Rockwell C 40 to 45 at 66 to 100 F/sec.

The second specification for steel is expressed in terms

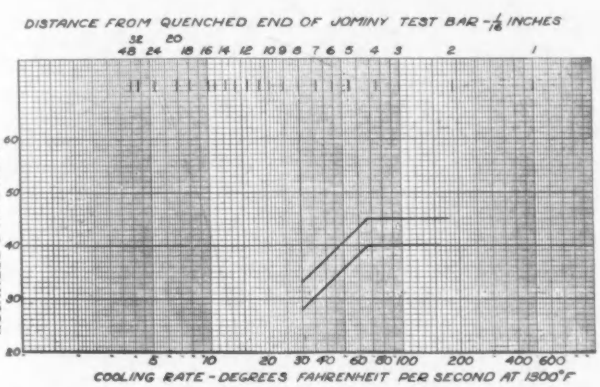


■ Fig. 10 – Hardenability of carburized case at 0.015 in. below the surface



■ Fig. 11 – Hardenability at surface of carburized case

relating not only to the test bar but also to the object to be made from the steel because the object will be referred to in terms of degrees per second cooling rate imposed during quenching.



■ Fig. 12 – H-CR limits for steels which will meet sample specification No. 4

H-CR Curve Method for Selecting Steel Applied to Rear Unit Center Gear for Hydramatic Transmission – Fig. 13 is a working drawing of the rear unit center gear. To determine cooling rates, this gear was made from 5040 steel. A Jominy test bar was made from the same bar of steel and the H-CR curve determined. H-CR curves were made also for X5145 steel and 5030 steel. These H-CR curves are shown in Fig. 14.

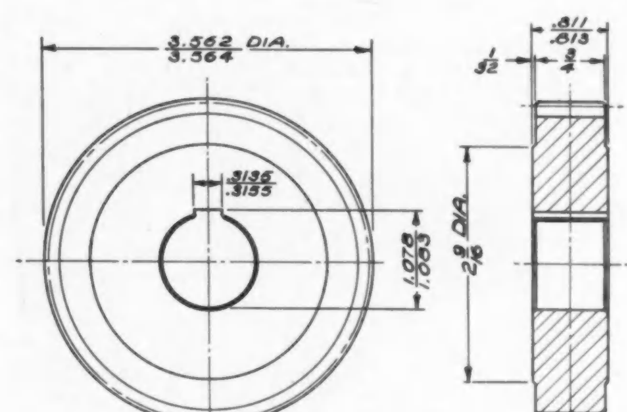
The gear made from 5040 steel was heated in activated salt at 1525 F and quenched in S-9 oil. The hardness traverse from outside to center of the gear is shown in Fig. 15. Also shown are the corresponding cooling rates picked from H-CR curve in Fig. 14 for 5040 steel. It so happened that the hardness values obtained with the 5040 steel are satisfactory for the gear in question. The H-CR curves for 5030 and X5145 steels can be used to tell us whether a gear made from these steels would be satisfactory.

The rate of cooling in the gear varies from 47 to 100 F/sec. The hardness obtained in each of the three steels for each cooling rate shown in Fig. 15 is shown in Table 2 following:

Table 2

Cooling Rate, F/sec at 1300 F	Hardness – Rockwell C X5145	5040	5030
100	57.5	54	46
82	57.0	52	42
74	56.0	49	39
70	56.0	47.5	38
64	55.0	45	36
58	54	43	34
56	53.5	42.5	33.5
54	53	42	33
52	52	41	32
49	51.5	40	31.5
47	50.5	39	30.5

From this information we can predict that X5145 may be unsatisfactory because of excessive hardness in the core



■ Fig. 13 – Rear unit center gear for Hydra-Matic transmission

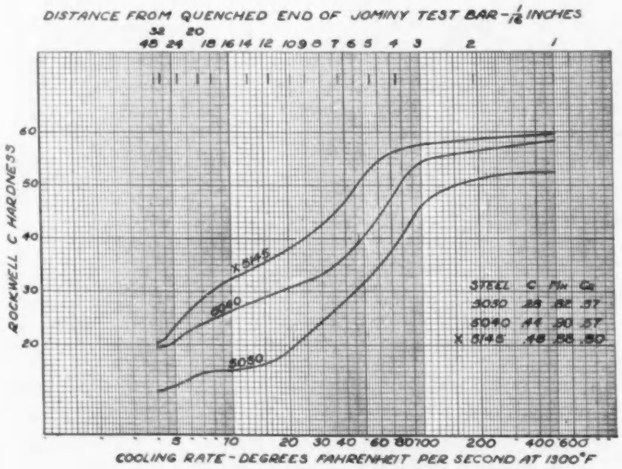
or hub of the gear. We also can predict that 5030 would be unsatisfactory because of too low hardness in the gear teeth – unless provided with a cyanide case and even then may not be good enough for heavy duty.

By plotting a curve of hardness versus carbon content in the three preceding steels, we can predict how much variation in hardness to expect at any particular cooling rate as a result of normal variation in carbon content in any one of the foregoing three steels.

Many companies have adopted methods for testing hardenability using some one of the several tests in existence. In the case of steel mills making a product which is usually a simple geometric shape such as a round, square, hexagonal or other symmetrical cross-section, a simple measurement of hardness penetration serves the purpose for classification of the product. Such information cannot be interpreted accurately, however, in terms of how the steel will behave in the complicated shapes forged from those simple bar shapes.

It is not necessary for those companies that already have standardized on some hardenability test procedure to abandon their present methods in order to correlate their information with that needed by consumers of steel. By the application of the principles previously described in this paper, it is possible to ascertain cooling rates throughout their standard hardenability test specimen be it cylindrical, flat, or conical. Once this is done, there will have been established a direct relationship between the test bar and the part for which the steel is intended. It may be repeated here that the means of obtaining this relationship between the object for which the steel is intended and the test bar is the H-CR curve. The H-CR curve is the master gage for the steel. Whether the H-CR curve is obtained from an end quench test bar or from a totally immersed quenched test bar is immaterial. The important point is to obtain an H-CR curve for every steel and this can be used to predict hardnesses in any object to be heat treated.

By way of illustration let us consider shallow-hardening steels in which class water-hardening tool steels come. O. V. Greene, of the Carpenter Steel Co., prefers to use a tapered test specimen heated to 1450 F and quenched in a brine flush. With this test always conducted the same way, the cooling rates in this test bar will always be the same in the same locations. When the cooling rates for this test bar have been once determined, they can be considered as fixed as the dimensions of the test bar so that hardness data obtained from that test bar can be plotted in terms of cooling rates as well as in terms of hardness penetration. In this way the hardenability information in the more useful form of an H-CR curve may be obtained.



■ Fig. 14 – H-CR curves for 5030, 5040, and 5145 steels

For similar shallow-hardening steels we prefer to use the Type L end-quench test bar shown in Fig. 1 and for which Fig. 2 gives the cooling rates at points along the side of the bar. The range of cooling rates that have been measured with this type bar extends up to 660 F/sec at 1300 F. It would be an extremely shallow-hardening steel that would not harden at that speed. Such a steel would not be commercially useful for hardening purposes.

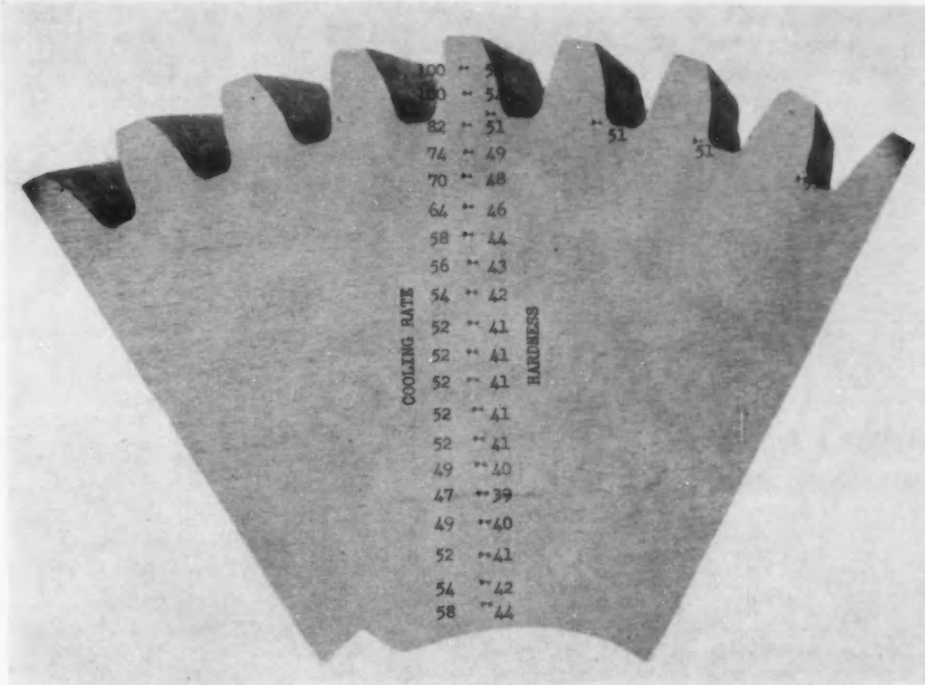
Whether one prefers the cone specimen or the Type L Jominy bar, or some other shape for obtaining hardenability of shallow-hardening steels, is not of great consequence. The important point is that the test bar affords a means of determining hardness over a certain range of cooling rates, thus providing data for plotting an H-CR curve. The cooling rates occurring in the tapered test bar used by Mr. Greene may be arrived at conveniently by the method described at some length in this paper.

Also shown in Fig. 1 is a means of obtaining an H-CR curve for steel purchased in sizes less than ½-in. diameter. Down to ½-in. diameter the standard end-quench test bar method may be used. Below that size there is danger of water traveling up the side of the test bar, thus spoiling the test. For sizes less than ½-in. diameter, the method of inserting the test piece in a sheath which is the same shape as the standard test bar is used. This is an adaptation of the Wuerfel hardenability bomb proposed by F. E. McCleary differing in the shape of the sheath to provide end quenching instead of total immersion. The cooling rates along this small insert bar are difficult actually to measure so they have been determined by substitution for hardness values on an H-CR curve of the same material tested in a 1-in. round end-quench test bar. It has been determined in this way that the cooling rates plotted versus distance from the quenched end are the same as for the standard 1-in. round end-quench bar. Cooling rates in the Wuerfel test piece may be determined in the same way. To do this a bar of the same steel is needed big enough to make the 1-in. round test bar. After the test bar is once calibrated as to cooling rates, H-CR curves may be obtained by using cooling rate values instead of penetration values or diameter of round values.

■ Severity of Quench Determination

The method of hardenability interpretation and translation proposed by Mr. Grossmann involves the use of a factor called "severity of quench." This factor does not have to be known to predict performance of steel by the H-CR method. As a means of classifying various quenching media as to severity of quench, however, the H-CR curve method is ideally suited because it affords a means of finding how cooling rates are changed in any size or shape bar by changing the quenching medium. The efficacy of various quenching media then may be expressed in terms of their effect upon cooling rates in quenched articles, thus giving the rating a special significance which is not possessed by any arbitrary rating.

The H-CR curve method also will be valuable in determining the effect of scale on the hardening of steel. The



■ Fig. 15 – Hardness traverse on gear section

effect of different kinds of scale expressed in terms of its retarding influence on cooling rate is fundamental information that will improve greatly our working knowledge of the heat-treatment of steel.

With such a simple means available for determining cooling rates in quenched objects, each user of steel can explore readily every heat-treated article that he makes. Such information collected and catalogued will make the selection of steel for machine parts a precise scientific procedure in place of the present cut-and-try method of selection.

■ Selection of Steel

In the selection of steel to be used for determining cooling rates in articles to be heat-treated, the object to be explored for cooling rates should be made from a steel of such hardenability that a wide range of hardnesses will result from quenching.

To illustrate, take the object shown in Fig. 3. When made of 5130 steel and treated as shown, the hardness varied from 53 to 40 Rockwell C. According to the H-CR curve for that steel shown in Fig. 3, 53 Rockwell C results from any cooling rate faster than 90 F/sec. We know, therefore, that, wherever the hardness is 53, the cooling rate was at least as fast as 90 F/sec. It is obvious from observing the shape of the piece that the 53 Rockwell C regions near the periphery of the flanged end cooled faster than the regions nearer the axis but how much faster we cannot tell because the H-CR curve is flat above 90 F/sec.

If this same object had been made from 4340 steel which has an H-CR curve shown in Fig. 6, the hardness would have been about 53 Rockwell C throughout and no idea would be gained as to cooling rates in the spool other than that they were all higher than 20 F/sec, that is, where the hardness of 4340 falls below 53 Rockwell C. If the object had been made from 3115 steel which has the H-CR curve shown in Fig. 9, we would have had hardnesses steadily increasing at cooling rates up to and higher than 90 F/sec, so the cooling rates in the flanged part of the object could have been determined as well as in all other parts of the object.

This teaches then that the steel selected for use in determining cooling rates in objects should be one having a continual gradation in hardness over a wide range of cooling rates such as is illustrated by H-CR curve for 3115 steel in Fig. 9. The steel selected for very light objects would be one having the slanting part of the H-CR curve in the high range of cooling speeds say from 100 to 600 F/sec. For heavy sections, a steel should be used having an H-CR curve with the slanting part of the curve in the low range of cooling speeds, say up to 40 F/sec.

■ In Terms of Hardness and Cooling Rates

As soon as cooling rates throughout an automobile component have been determined, and it has been decided what hardness is desired in each part of that automobile component, the hardenability specification for the steel to be used for that component is thereby defined. By plotting the desired hardness versus the cooling rates in the various parts of the component, we obtain an H-CR curve that defines the hardenability of the steel required to provide the desired hardness. It remains only to decide how much variation in hardness can be tolerated at each location in the component, that is, how much variation can be tolerated at each of the cooling rates in the component. The decision of what this tolerance may be determines two H-CR curves, one with a maximum hardness at each cooling rate and the other with a minimum hardness at each cooling rate. The H-CR curve for the steel that meets the specification will fall between these two specified H-CR curves.

This fundamentally is the complete hardenability specification for the steel. Limited portions of the two complete curves may be used for specification purposes. How complete a description of the two limiting curves is given in the specification for the steel depends on how wide the

hardness limits are and whether only minimum limits are required instead of both maximum and minimum.

The simplest form of specification would be the first one following. Other more restricting specifications would appear like those following No. 1:

1. 50 Rockwell C minimum at 100 F/sec cooling rate.
2. 50 Rockwell C at 60 to 100 F/sec cooling rate.
3. 50 to 55 Rockwell C at 100 F/sec cooling rate.
4. 30 to 40 Rockwell C at 30 F/sec cooling rate. 50 to 55 Rockwell C at 100 to 200 F/sec cooling rate.

DISCUSSION

Favorable Experience with Jominy Hardenability Test

– J. H. Jones
Republic Steel Corp.

MR. BOEGEHOLD apparently feels that there may be some reluctance on the part of steel producers to interrupt their present procedure for checking hardenability. This is not true in our case. In fact, we will welcome a simple standard test that will permit hardenability to be stated in a common term. Such a test would immediately indicate not only whether steel meets the hardenability requirements for the order for which it was made, but would classify it for possible application on any other order specifying hardenability, and without further testing.

Steel users now specify that their hardenability requirements be determined by running quite a variety of test pieces, such as key sections, discs, rounds, cubes, flats, and square or rectangular sections. There has been little or no definite coordination of these tests, therefore, it is obvious that a standard test that would at once classify a heat as to its possible application, without further testing in the various sections just mentioned, would be desirable and would save much time for the mill.

We believe that the standardization of hardenability testing and the elimination of unnecessary testing is at this time just as important as the standardization of chemistries.

The hardness cooling rate curves are obviously important in developing certain laboratory data but, when specifying Jominy hardenability to the mill, it is suggested that for the sake of simplicity and to avoid misinterpretation, it be stated in terms of "Jominy" Rockwell C so far from quenched end or (RC 35 min. at 9/16 in.).

Good judgment must be used when both top and bottom hardness values are specified. It is suggested that such a set-up not be made until experience with a number of heats has indicated the hardness range necessary.

It is known that the Jominy test does not give a very true picture of segregation. This is probably the major criticism that can be raised against its adoption. In our experience, this shortcoming by no means outweighs its advantages.

We have run several thousand standard Jominy test pieces and have released a considerable amount of steel by this method over the past 18 months. Our experience indicates that it is the most satisfactory hardenability test now available for a large group of the automotive steels. We will gladly cooperate with anyone wishing to substitute it for any of the individual test pieces now in use.

Determining Hardenability on Small Sizes

by F. E. McCLEARY and R. WUERFEL
Chrysler Corp.

LIMITATIONS of the two general methods available for determining hardenability in steel, the authors point out, are that the test piece may not have a sufficient cross-section in which to develop the desired series of cooling rates, and that a special test piece (known as the L-type) must be machined for steels of low hardenability. The method using the Wuerfel bomb described in their paper, they explain, is directed primarily toward removal of these two limitations.

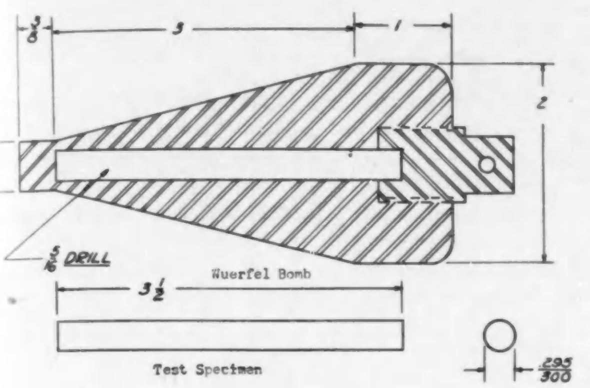
Stated in terms of the critical diameter, they report that the results of the method are reproducible within 1/8 in.

■ ■ ■

THERE are available two general methods for determining hardenability in steel: (1) a cylinder quenched by immersion; and (2) the Jominy method in which a cylinder is quenched on one end only. In Method (1) the cross-section is examined by visual means, or a series of hardness readings are made. In Method (2) a longitudinal series of hardness readings is taken on the outside of the bar. Both methods are alike in that a series of varying cooling rates is imposed on the test piece. The relation between cooling rate and resultant hardness is fundamental in all such tests.

The application of hardenability measurements as an inspection procedure directs attention to the size of stock available, or the dimensions of a test piece which may be machined from a finished part. That is, the test piece may not have a sufficient cross-section in which to develop the desired series of cooling rates by Method (1). Prior to a recently published modification the Jominy method has not been recommended for sizes below 5/8 in. diameter, and still requires the machining of a special test piece (known as the "L" type) for steels of low hardenability. The method to be described is primarily directed towards removal of these two limitations.

Fig. 1 shows the construction of the bomb used for this determination. Briefly, it is a conical-shaped piece of steel provided with a hole concentric with the axis of the cone. This hole is enlarged at the top and threaded to take a plug, which also provides the handle of the assembled bomb. The hole is slightly larger than the test piece. This provides an annular space to be filled with a low-



■ Fig. 1 – Wuerfel bomb and test specimen

melting alloy such as Wood's metal which provides thermal contact between bomb and test piece.

In use, the bomb is warmed above the melting point of the alloy, the required amount poured in, the test piece put in place, and the plug screwed in. The assembly is then heated to the temperature suited to the steel under test and quenched in water. The assembly is warmed to melt the alloy and the test piece removed. Hardness readings are taken along the length of the test piece and data are at hand for the calculation of "hardenability." This term is used as defined by Grossmann and Asimow[1], and their procedure for calculating critical diameter gives a rating which may be made common to all other procedures.

Composition of Low-Melting Alloy	Steel Used in Making Bomb
50% Bismuth	0.12% Carbon
25% Lead	
25% Tin	0.22% Manganese
Melting Point, 200 F	

Five parts of bismuth may be replaced by mercury, with a consequent lowering of the melting point to 185 F. This alloy is also more brittle when cold, allowing easier removal from the threads of the bomb.

■ Standardization

By ascertaining on the test piece, the position of the 50% martensite structure[1] and relating it to the critical diameter as determined directly by the Grossmann procedure, the plot in Fig. 2 may be constructed. Thus the longitudinal and cross-sectional positions of equal cooling rate, as determined by the two procedures, are correlated.

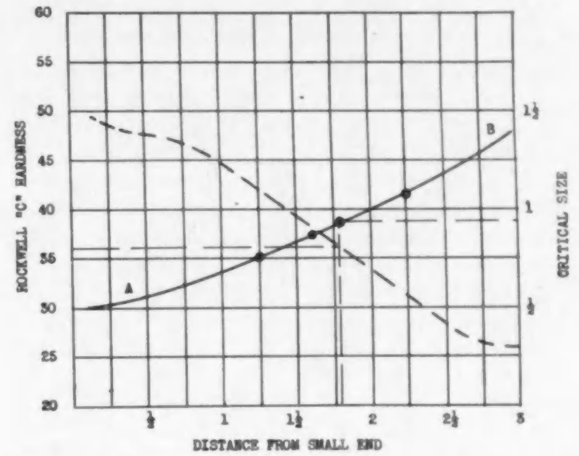
In Fig. 2, the broken line is a plot of the Rockwell hardness as obtained in the bomb against the position along the length of the piece. The solid line *AB* then relates the longitudinal position of critical hardness to the critical diameter.

The four points on *AB* have been determined by quenching four series of cylinders with varying diameters, one series from each of four different steels. Hardness data as given by the Wuerfel bomb also were determined on the same four steels. To avoid complication only one of these is shown.

Standardization might have been made by direct cooling-rate determination. However, the bomb was designed originally to take care of a small part of the hardenability field, so that comparative methods seemed sufficient.

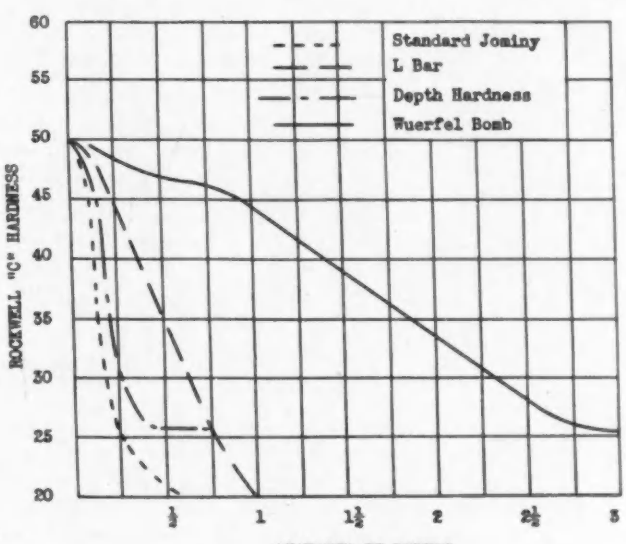
■ Accuracy

Stated in terms of the critical diameter, the results are reproducible within ⅛ in. This limit probably can be narrowed somewhat by careful work. The relative dimensions of the test piece and the hole in the bomb are important. The minimum annular spacing is approximately 0.006 in. A spacing of 0.024 in. will vary the critical diameter 3/32 in.



■ Fig. 2 – Critical diameter and hardness along the length of the piece

Best results in protecting the surface of the bomb have been obtained by using a nickel-plated surface and heating in a nearly neutral atmosphere. Quenching has been carried out in a tower, the water being forced in at the bottom at a fixed rate. The details of the quenching system are



■ Fig. 3 – Comparison of Wuerfel method with Standard Jominy, L-Bar, and "Depth-Hardness" procedures

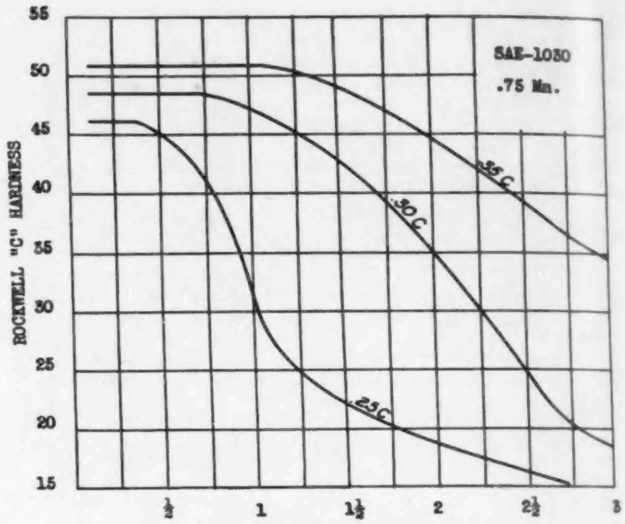
[1] See ASM Monograph: "Hardenability of Alloy Steels," pp. 154-171, Cleveland, 1939.

relatively unimportant. The principal requirements are that the quench should be constant throughout all related tests and that the maximum hardness characteristic of the steel should be approximated at the small end of the bomb.

■ Comparison With Other Methods

A comparison of the Wuerful method with the two standard procedures is shown in Fig. 3. The steel used had the following analysis: C, 0.29, Mn, 0.82, Si, 0.24, Mo, 0.22. The cooling-rate scale is seen to open to a marked degree. The curve labeled "Depth Hardness" is the cross-section hardness of a quenched cylinder. The Standard Jominy and "L" Bar are end-quenched specimens, as recommended by Jominy. The curve for the "L" bar was constructed by using cooling-rate data published by Jominy.

Hardenability ratings on SAE 1030 in as-received sizes ranging from ¼ to ½ in. round have been determined on 250 samples representing 40 heats. The indications are at present that quench cracks found in production will occur in steels having the larger critical diameters. However, the spread in the whole series is not great and some factor not yet determined may have caused this grouping. The expansion of the cooling-rate scale can be controlled, within limits, by the taper of the bomb. The particular taper used produces a broad band when the data just mentioned are plotted, Fig 4. Obviously, shrinking the cooling rate scale to the limits of the other methods noted will mask the hardness variation.



■ Fig. 4 – Hardenability ratings on SAE 1030 steels

At present the sensitivity of the bomb cannot be stated definitely. The result of further study on this point will be available at a later date.

Determination of Specific Hardenability of Shallow-Hardening Steels

by O. V. GREENE and C. B. POST
Metallurgical Department, The Carpenter Steel Co.

AT the present time users and producers of shallow-hardening steels are confronted with a large number of different types of tests which purport to give various measures of the so-called hardenability of these types of steel. Sometimes the correlation among these various tests is a matter of some difficulty, and consequently the desirability of having a fundamental, simple and easily interpreted test is obvious. In order to measure the property which will be hereafter designated as "specific hardenability," it is necessary that the test embody several fundamental concepts which have been established in recent years.

The first of these concepts is concerned with the assumption that the hardness attained for any particular grade of steel is a function of the rate of cooling through some particular temperature, such as 1100 F or 1300 F. Upon such an assumption the Jominy-Boegehold tests[1] are developed. Using experimentally determined cooling rates along a bar of fixed dimensions, Jominy and Boegehold determine the Rockwell hardness as a function of cooling velocity at 1100 F or 1300 F. In this manner their tests classify hardenability of any particular steel in terms of a so-called "critical cooling velocity," which, according to these authors, is the cooling velocity needed to obtain a given predetermined Rockwell hardness.

■ Fig. 1 – Dimensions of taper test specimen

[1] See Transactions of the American Society for Metals, Vol. 26, pp. 574-606: "A Hardenability Test for Carburizing Steel," by W. E. Jominy and A. L. Boegehold.

A TAPER test specimen having a taper of 1 in. in 5 in. of length, 1/4 in. in diameter at one end and 1 1/4 in. in diameter at the other end, is proposed as a test for measuring the specific hardenability of shallow-hardening steels. The hardened taper test specimens are split and Rockwell hardnesses obtained along the central axis of test specimen, or the central section may be etched lightly to bring out the relative colors of case and core. This taper test specimen has been correlated with the Jominy-Boegehold "L" bar. This enables the rate of cooling at any point along the central axis of the taper test specimen to be expressed in terms of deg F per sec at 1300 F.

This taper test specimen has also been correlated with the work of Grossmann and his associates concerning critical bar diameters and severity of quench. It is shown experimentally that this degree of taper is small enough so that the taper test specimen behaves within experimental error like a series of round bars whose diameters are given by twice the perpendicular distance from the surface of the taper test specimen to any point along the center axis.

A correlation is shown between the Shepherd disc hardenability Nos. 10 to 16 inclusive and the critical cooling velocity in deg. F per sec at 1300 F (as determined from the cooling rates published for the Jominy-Boegehold Type "L" bar), and critical bar diameters at a severity of quench of $H = 4.5$, or the "ideal critical bar diameters" for $H = \infty$.

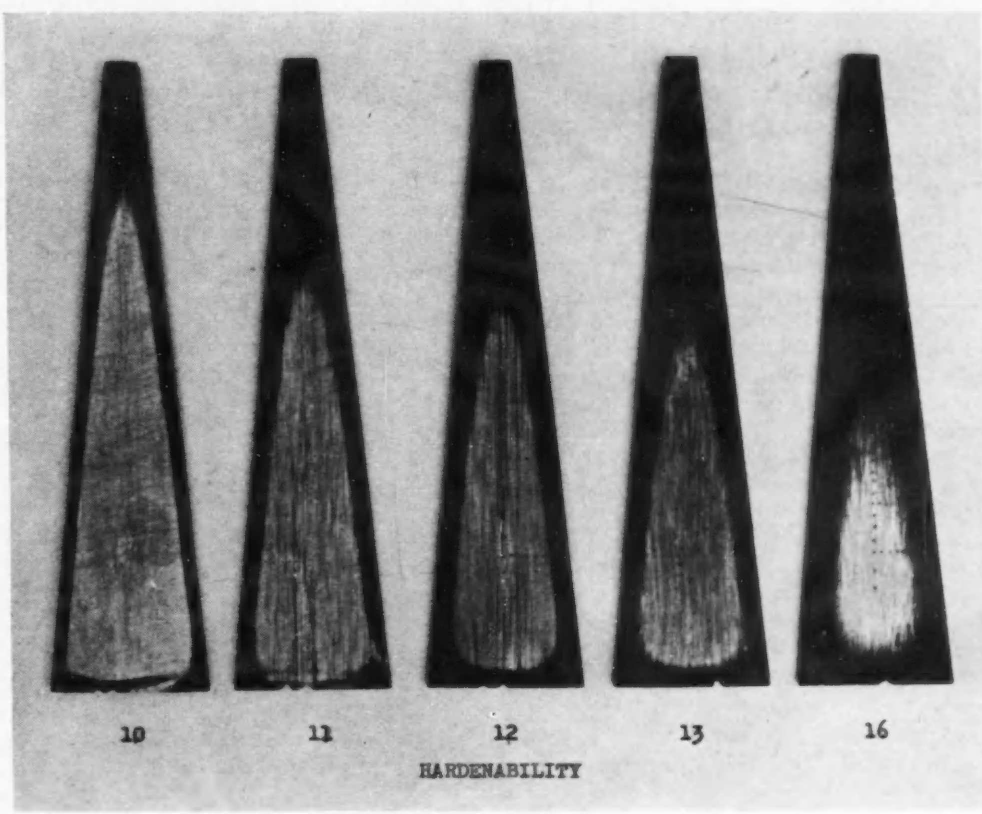
Considering the experimental application of the Jominy-Boegehold "L" bar[2] to the evaluation of hardenability of shallow-hardening steels, it should be pointed out that a greater differentiation between steels of various hardenabilities is desired than is shown on this type of test bar.

Another fundamental concept concerning the specific hardenability of steels has been exhaustively studied by Grossmann and his associates[3]. Grossmann considers the effect of severity of quench on the penetration obtained in various sized round bars, and at the same time considers the so-called critical bar diameter at constant severity of quench. This critical bar diameter is believed to have more physical significance in hardenability than the arbitrary selection of any given Rockwell hardness, such as might be done in the Jominy-Boegehold test. For this reason it would be well to incorporate in a fundamental test for shallow-hardening steels this concept of critical bar diameter at constant severity of quench. Grossmann and his associates have made available methods by which these critical bar diameters may be transferred to an ideal critical bar diameter, which represents the penetration obtained in a perfect quenching medium.

Thus, for the purpose of developing a fundamental test

[2] See Transactions of the American Society for Metals, Vol. 27, 1939, pp. 1072-1089: "A Hardenability Test for Shallow-Hardening Steels," by W. E. Jominy.

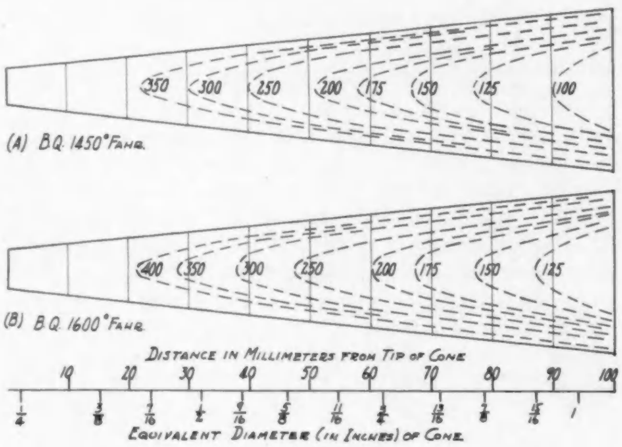
[3] See "Hardenability of Alloy Steels," published by American Society for Metals, Cleveland, 1939, pp. 124-190: "Hardenability, Its Relation to Quenching," by M. A. Grossmann, M. Asimow, and S. F. Urban.



■ Fig. 2 – Penetration pattern for Shepherd Hardenabilities Nos. 10, 11, 12, 13, and 16

for shallow-hardening steels, it would be desirable to have a test which would incorporate the concept of cooling velocities through a given temperature, say 1300 F and, at the same time, give due weight to the equally fundamental concept of critical bar diameter at a given severity of quench. Such a test would then be able to evaluate specific hardenability in terms of either: (1) cooling velocity necessary to give a certain minimum Rockwell hardness or, (2), critical bar size at a given severity of quench.

It is also to be understood that this test will not be introduced merely to supplement other tests on hardenability but will furnish a medium by which other tests may be correlated with one another in terms of this more fundamental test. A test specimen which will answer the foregoing requirements consists of a tapered round bar, the degree of taper being 1 in. of diameter for each 5 in. in length. The actual specimens used here were ¼ in. in diameter at one end, and 1¼ in. in diameter at the other, by 5 in. long. The test specimen is shown in Fig. 1.



■ Fig. 3 – Iso-rate lines (deg F/ sec at 1300 F) for taper cone specimen

In applying this test to a determination of specific hardenability, these tapered test pieces were heated in a gas-fired semi-muffle furnace, using an atmosphere inside the furnace which contained approximately 4% O_2 and 8.2% CO_2. The test pieces were soaked 10 min at temperature, generally 1450 F, and quenched in a 10% brine solution at room temperature in 3-in. vertical pipe flush, using an overflow 1 in. in height. After quenching, the specimens were ground down to the center section, and Rockwell hardness readings taken down the center line of the taper test, and also on lines normal to the surface. If it is so desired, this center section may be etched lightly to show the penetration, and several samples are shown in Fig. 2 of split test pieces for Nos. 10, 11, 12, 13 and 16 Shepherd hardenability[4] steels. As may be seen in Fig. 2, the taper test specimen offers a large range of differentiation between shallow-hardening steels of various commercial hardenabilities.

The reproducibility of the taper test when determined by means of Rockwell readings down the center section of the test pieces from the same heat, using the same treating procedure, has been found to be ±2 mm along the central axis at the critical hardness point. This critical hardness is understood to be the inflection point on the Rockwell hardness profile taken down the center line of the cone, that is, the hardness where the Rockwell hardness is changing most rapidly with change in distance.

It might be well to point out several minor disadvantages of this taper test piece for shallow-hardening steels. In the first place, the test requires a bar at least 1 in. in diameter. To overcome this, we have tried cones made out of austenitic stainless steels in which plugs made of shallow-hardening tool steels have been inserted, somewhat in the manner of the Wuerfel hardenability bomb, as recommended by McCleary[5]. This bomb has not been found suitable for shallow-hardening steels because of the occurrence of soft spots. It must be realized that these shallow-hardening steels require drastic, uniform quenches, and therefore slight variations in the conductivity of Wood's metal, and so on, lead invariably to dangers of soft-spot formation. Another disadvantage is that the taper test specimens must be split or ground down and kept cool at all times. In case these taper specimens are split by means of radial cuts, a coolant must flow on the specimen at all times in order to prevent overheating of the test piece during the cutting operation. In those laboratories where grinding equipment with automatic flush is not available, this could be a time-consuming operation.

The rates of cooling of various round sections, as determined experimentally by H. J. French[6], enables the rate of cooling at any point on the longitudinal section of the taper test to be specified to a first approximation. These cooling velocities for brine quenching from 1450 F (in F/sec at 1300 F) are given in Fig. 3 in the form of an "Iso-rate" chart for the taper test piece. The severity of quench for which this Iso-rate chart is valid is characteristic of the brine flush just mentioned, that is, a 3-in. vertical pipe with about a 1-in. overflow.

At the present time experiments are in progress in our laboratories to determine experimentally these rates of cooling on the longitudinal section of the taper test piece. The work to date indicates that the rates of cooling shown in Fig. 3 are correct within experimental error.

A consideration of the heat flow conditions in the taper test specimens during the quench leads to the conclusion that, at every point on the surface of the taper test specimen, heat is being conducted from the test specimen approximately normal to its surface. The rate of cooling at any point along the center axis of the taper test specimen will approximate the rate of cooling of a round bar whose radius is equal to the perpendicular distance of this point from the surface of the taper test specimen. This perpendicular distance will become equal to the radius of the round bar as the degree of taper decreases. Experimentally it has been determined that the taper test specimen having a taper of 1 in. in 5 in. of length behaves within experimental error like a series of round bars whose diameters are given by twice the perpendicular distance from the surface of the taper test specimen to any point along the center axis of the specimen.

This statement was determined experimentally by a

[4] See Transactions of the American Society for Steel Treating, Vol. 17, 1930, pp. 90-101: "Inherent Hardenability Characteristics of Tool Steel," by B. F. Shepherd.

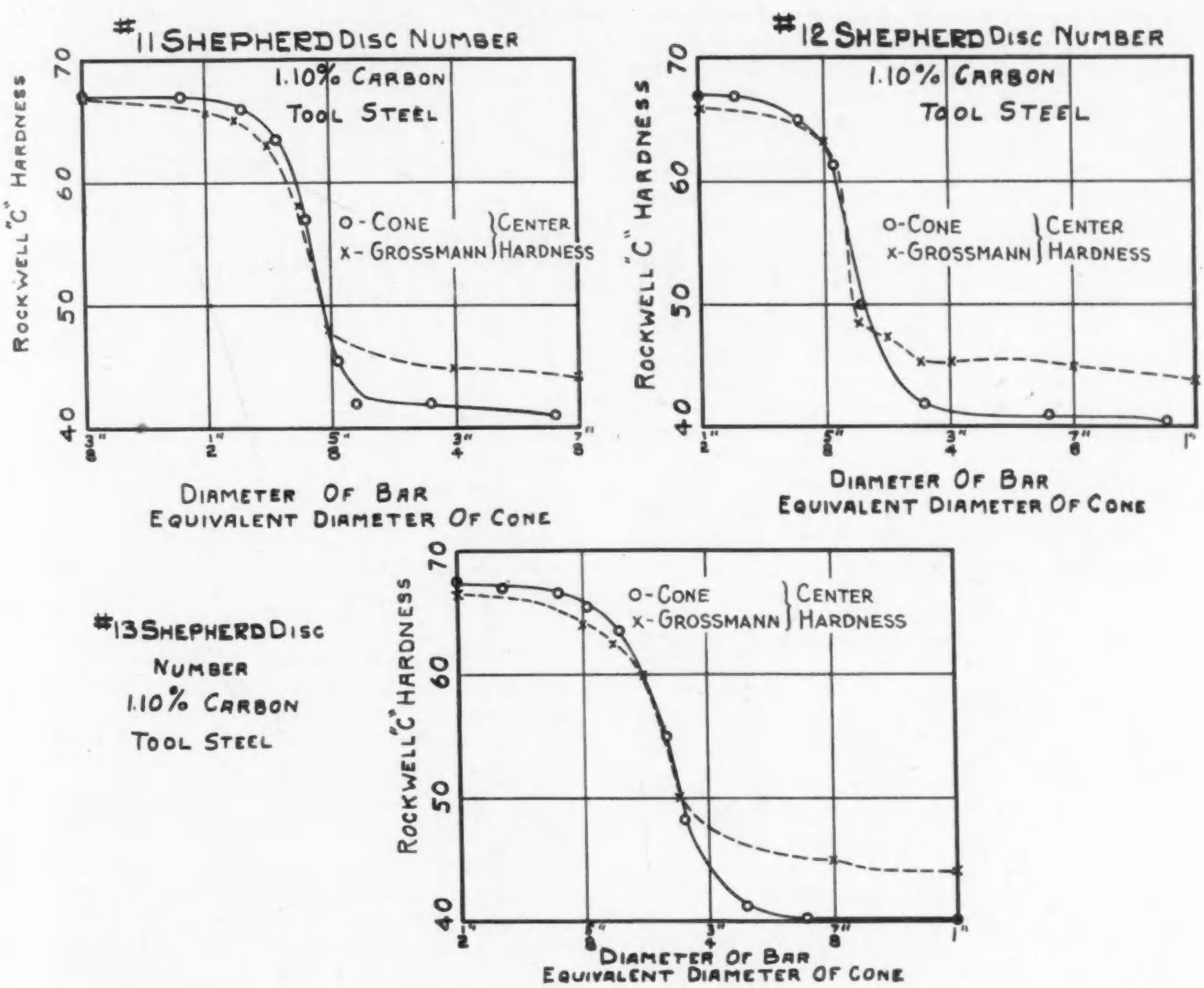
[5] See "Wuerfel Hardenability Bomb," by F. E. McCleary, Report to the SAE Committee on Hardenability, Aug. 5, 1940.

[6] See Transactions of the American Society for Steel Treating, Vol. 17, 1930, pp. 646-727; 798-888: "A Study of the Quenching of Steels," by H. J. French.

study of a series of rounds ranging from 1¼ to ⅜ in. in diameter, which were machined from three heats of steel, representing Shepherd hardenability numbers of 11, 12, and 13. The lengths of these rounds in all cases were at least five times their diameter. These rounds were soaked 10 min at 1450 F in a gas furnace with an atmosphere containing about 4% O_2 and 8.2% CO_2. The specimens were brine-quenched in the 3-in. vertical pipe flush, using an overflow 1 in. in height. Thus the heat-treatment of these rounds corresponds exactly with the heat-treatment used in hardening the taper test specimens. These pieces were then cut in half and Rockwell hardnesses determined over the cross-section. In this manner the center hardnesses of this series of rounds were obtained.

Taper specimens made from the same three heats were hardened and split, in the same heat-treating manner as just described. After these taper specimens were split and ground down to the cross-section, Rockwell hardnesses were taken down the center axis of the taper test specimen as described previously. These center hardnesses were then compared directly with the center hardnesses obtained from the series of round bars, ranging from 1¼ to ⅜ in. round, made from the same heats. These results are shown in Fig. 4. Hereafter it is to be understood that the equivalent diameter to be associated with any given point along the center axis of the taper test specimen is twice the perpendicular distance from the surface of the specimen to the point in question. The results of Fig. 4 show clearly that, within the significant portion of the test piece wherein heats of No. 10 to No. 16 hardenability will just harden through, the heat-flow phenomenon associated with any given point along the central axis of the taper test specimen is the same as that associated with the center point of a round bar whose diameter is equal to the equivalent diameter.

This enables the work of Grossmann and his associates concerning critical bar diameter at a given severity of quench to be applied to the taper test specimen for obtaining approximate penetrations of other sized rounds and shapes. Grossmann defines the critical bar diameter to be the diameter of round which just hardens through for any given heat of steel. In the case of 1.10% carbon tool steels



■ Fig. 4 – Center hardnesses of taper test specimens compared with center hardnesses obtained from a series of round bars

it has been found that a Rockwell of C-55 is generally associated with the inflection point of any Rockwell contour curve, that is, C-55 is the hardness of a structure composed of 50% martensite and 50% troostite. This Rockwell hardness of C-55 also can be shown to be the boundary line between etched case and core. The critical bar diameter of Grossmann may be read directly from either the Rockwell contour down the center axis, or by inspection of the etched case-core pattern obtained on the longitudinal cross-section of the taper test. This follows because the tip of the case-core pattern of the taper test represents the point where the Rockwell hardness is falling most rapidly with change in distance along the axis and, as previously pointed out, the structure at this tip is 50% martensite and 50% troostite. The critical bar diameter may thus be found very conveniently when the taper test specimens are used as a test for specific hardenability of shallow-hardening steels.

Another important consideration of Grossmann and his associates has been concerned with evaluating the severity of quench employed during any specific experimental measurement of hardenability. It would take us too far afield to consider in detail Grossmann's work along this point, but it is pertinent to point out that the etched case and core pattern of the split taper test specimens may be used easily in the same manner that Grossmann uses the penetration obtained on various size round bars to evaluate the severity of quench. The severity of quench appropriate to the 3-in. vertical pipe flush with about a 1-in. overflow has been found to yield H-values ranging between 4 and 5.

The work of Grossmann and his associates on critical bar diameters leads to the concept of critical bar diameter obtained for an ideal quench, that is, when the H-value associated with the severity of quench is equal to infinity. This "ideal critical" bar diameter for $H = \infty$ may be determined from the critical bar diameter obtained experimentally when the severity of quench is measured quantitatively. There is a distinct possibility in the future that, if Grossmann's method of measuring hardenability is employed to any great extent among users of shallow-hardening steels, misunderstandings concerning the severity of quench employed in various laboratories may arise. Thus the problem of the interpretation of hardenability data would remain the same, except for the change in terminology from "hardenability" to "severity of quench."

We believe that Grossmann's greatest contribution to hardenability will eventually be associated with the methods he has developed for predicting, to a first approximation, the Rockwell contours for various sized rounds when either the center hardness of a group of round bars is known at a given severity of quench, or when the Rockwell contour of a given round bar is known in addition to the critical bar diameter and severity of quench. On the other hand, Jominy and Boegehold cancel out the effect of severity of quench because this variable is taken care of by carefully standardized experimental procedures. Whether specific hardenability should ultimately be expressed in "rate of cooling through 1300 F" or in terms of "ideal critical bar diameter" cannot be judged at the present time. Both concepts have important bearing on the subject of hardenability. It is to be noted, however, that the taper test specimen as developed here, embodies the essentials of both concepts when applied to specific hardenability of shallow-hardening steels.

The first evaluation that we have attempted between various hardenability tests in use at the present time was between the Shepherd disc number and the specific hardenability obtained from the taper test specimens. This specific hardenability is expressed in terms of (1) critical cooling rates and (2) critical bar diameters as defined by Grossmann. The critical cooling rate, as previously pointed out, is defined to be the cooling rate necessary to give the Rockwell hardness of the inflection point of the Rockwell contour curve. In the case of 1.10% carbon tool steels this inflection point has a Rockwell hardness of C-55. Thus the critical cooling rate is the cooling rate through 1300 F which will yield a structure composed of 50% martensite and 50% troostite (and whose hardness is equal to C-55). This correlation is shown in Table 1. Table 1 shows this correlation for heats ranging from No. 10 to No. 16

Table 1 – Correlation between Shepherd Disc Number and Specific Hardenability Obtained from Taper Test Specimens

Shepherd Disc Number	Critical Cooling Velocity[a] F/sec at 1300 F	Critical Bar Diameter[b] $H = 4.5$	Mean "Ideal Critical Bar Diameter"[b] Bar
10	340/370	$^{29}/_{64} - ^{31}/_{64}$ in.	0.62 in.
11	270/300	$^{17}/_{32} - ^{19}/_{32}$ in.	0.72 in.
12	230/255	$^{39}/_{64} - ^{41}/_{64}$ in.	0.78 in.
13	195/230	$^{43}/_{64} - ^{23}/_{32}$ in.	0.84 in.
14	155/185	$^{47}/_{64} - ^{49}/_{64}$ in.	0.92 in.
15	140/155	$^{25}/_{32} - ^{53}/_{64}$ in.	0.98 in.
16	110/140	$^{27}/_{32} - ^{29}/_{32}$ in.	1.04 in.

[a] Cooling velocity necessary to give Rockwell hardness at inflection point of Rockwell contour curves; $R_c = 55$ for 1.10% carbon tool steels.
[b] After M. Grossmann and Associates.

Shepherd hardenability or for steels in the range of critical cooling velocity from 370 F per sec to 110 F per sec. Also included in Table 1 are shown the critical bar diameters (severity of quench specified by $H = 4.5$) and the extrapolated "ideal critical bar diameters" which are to be associated with an ideal severity of quench, $H = \infty$. This latter value was determined by the published methods of Grossmann and his associates.

■ Summary

1. The specific hardenability of shallow-hardening steels can be measured by use of a taper test specimen. This test specimen consists of a tapered round bar, the degree of taper being 1 in. of diameter for each 5 in. in length. The actual specimens used in this investigation were ¼ in. in diameter at one end, and 1¼ in. in diameter at the other end, by 5 in. long.

2. Rates of cooling along the longitudinal section of the taper test specimen are shown in Fig. 3. This enables the taper test to give specific hardenability in terms of a "critical cooling rate" (in deg F per sec) at 1300 F.

3. The taper test specimen has been correlated with the work of Grossmann and his associates on "critical bar diameters" and severity of quench. It has been shown experimentally that the taper test specimen behaves within experimental error like a series of round bars whose

diameters are given by twice the perpendicular distance from the surface of the taper test specimen to any point along the central axis. This enables the severity of quench to be evaluated by the methods proposed by Grossmann and his associates, and also enables approximate calculations to be made concerning the penetration to be expected in any given sized round bar.

4. By way of illustrating the use of the taper test specimen as an instrument for measuring specific hardenability, a correlation is shown between the Shepherd disc number and the "critical cooling rate" obtained from taper test specimens. In this case, the critical cooling rate is defined to be the cooling rate necessary to give the Rockwell hardness of the inflection point of the Rockwell contour curve. This correlation covers the range of hardenabilities designated by Shepherd disc numbers of Nos. 10 to 16 inclusive. Critical bar diameters at a severity of quench of $H = 4.5$ and the extrapolated "ideal critical bar diameter" of Grossmann also have been correlated with the Shepherd disc hardenability numbers.

DISCUSSION

Effect of Type of Steel On Wuerfel Bomb Results

-R. K. Wuerfel
Chrysler Corp.

I APPRECIATE the authors' interest and comment on their experience with my test. The conductivity of rustless and stainless steels is from ½ to 1/3 that of plain carbon. This limits the small end of a stainless bomb to a critical size of about ⅞. Water-hardening tool steels run as low as ½ in. in critical size. This low hardenability limits the use of stainless.

I have had very good luck with water-hardening tool steels down to an analysis of 100- C and 0.15 Mn in a bomb made of 0.15 Carbon and 0.22 Mn. The higher conductivity of plain carbon gives a critical size of ½ in. at the small end. Special design of the bomb or higher conductivity material will allow smaller critical sizes should there be this need.

Correlation Between Jominy Test and Quenched Round Bars

by M. ASIMOW, W. F. CRAIG, and M. A. GROSSMANN
Carnegie-Illinois Steel Corp.

A NUMBER of different tests have been developed for ascertaining the hardenability of steel, that is, its susceptibility to hardening by quenching. Although each of these different tests may be suited particularly to a specific problem, it would be useful to know how to interpret one test in terms of another. The present paper suggests a manner of correlating the extent of hardening in the Jominy-Boegehold end-quench test with the extent of hardening in quenched round bars.

For any particular steel, the extent to which it hardens when quenched varies with the cooling rate (cooling time) in the quench. That is, if cooled rapidly enough, it will become hard and, if cooled slowly, it will be soft, so that for each steel a series of hardnesses may be found experimentally corresponding to a series of cooling times. Different cooling times occur along the length of a Jominy bar, and various cooling times are also found at various positions in different sizes of quenched bars, quenched with various severities of quench.

It therefore becomes possible to predict from the results of a Jominy test what the hardness distribution will be on the cross-section of a quenched round bar when quenched with a known severity of quench.

■ ■ ■

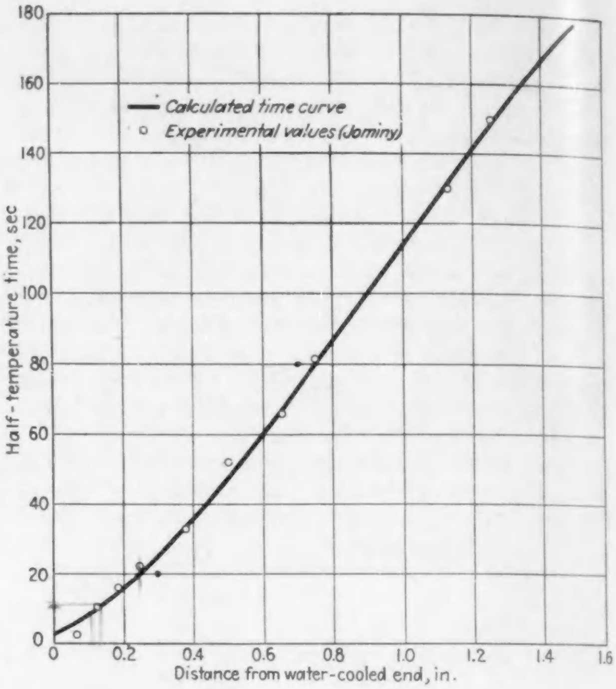
IN the Jominy end-quench test, first reported for carburized steels by Jominy and Boegehold[1] and then for direct-quenched steels by Jominy[2], the piece is quenched on its end with a stream of water of precisely stipulated velocity. This provides a constant quench, and the resulting hardening is stated in terms of the distance from the quenched end. In the case of quenched round bars, the severity of quench can be ascertained[3, 4], so that the hardness distribution in such bars can be stated in terms of a relationship to the time occupied in cooling in the quench.

(Footnotes 1-4 on following page)

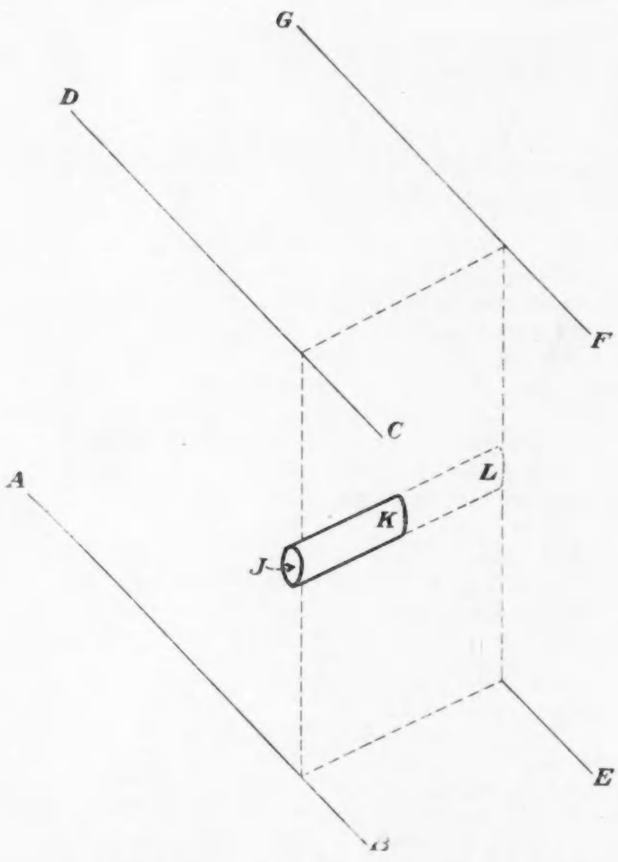
If, therefore, the cooling conditions in the Jominy quench can be ascertained, and thereupon related to the cooling conditions in quenched round bars, it should be a simple matter to predict, from the results in a Jominy test, what the hardness distributions would be in round bars when using a known severity of quench.

In studying the relationship between hardening and severity of quench[3], it was found that the extent of hardening for any particular steel correlated well with the "half-temperature time" in cooling. (This is the time occupied in the quench, in cooling from the quenching temperature to a temperature half-way down to that of the quenching medium. It is used in the same manner that "cooling rate" has been used by some investigators.) Therefore, the half-temperature times in a Jominy test, when correlated with the half-temperature times in quenched bars, should provide a basis for correlation of the hardening in these two forms of test. In the case of the Jominy bar, the quenching conditions involve two factors: (1) the cooling effect on the quenched end due to the stream of water and, at the same time, (2) the cooling effect of still air on the rest of the bar. The half-temperature times resulting from these simultaneous effects are shown in Fig. 1, where the curve shows both experimental and calculated values.

The experimental data were very generously contributed by W. E. Jominy[6], who had determined the values at a series of positions as shown in Table 1 and plotted in Fig. 1.



■ Fig. 1 – Half-temperature times – Calculated and experimental values



■ Fig. 2 – Basis for calculating water-quench effect

The calculated values were derived by a scheme described previously[3], adapted to the method suggested by T. F. Russell[7] for simultaneous cooling effects on different faces. The proportionate temperature level after a particular time is considered to be the product of the proportionate levels due to the two cooling effects (water on the end, air cooling on the balance). That is to say if, at a particular position after a particular time, the temperature would have been at 0.6 of its original value due to the cooling effect of the water alone, and at 0.9 of its original value due to the cooling by air alone, then the resulting value would be $0.6 \times 0.9 = 0.54$ of its original temperature level. The *water-quench* effect on the end can be calculated on the basis of Fig. 2. The Jominy bar, marked *JK* in Fig. 2 can be considered to behave as though it were part of a large plate, whose opposite faces are *ABCD* and *EFG* respectively. The thickness of this plate would be twice the length of the Jominy bar, as indicated at *JKL* in Fig. 2, and the quenching behavior is considered to be that of a plate of this thickness (but of infinite length and width) quenched on the opposite faces, namely on faces *ABCD* and *EFG*. For the *air-cooling* of the rest of the bar, the Jominy bar is considered as being

[1] See Transactions of the American Society for Metals, Vol. 26, 1938, pp. 574-606: "A Hardenability Test for Carburizing Steel," by W. E. Jominy and A. L. Boegehold.

[2] See "Hardenability of Alloy Steels," published by American Society for Metals, Cleveland, 1939, p. 66: "Hardenability Tests," by W. E. Jominy.

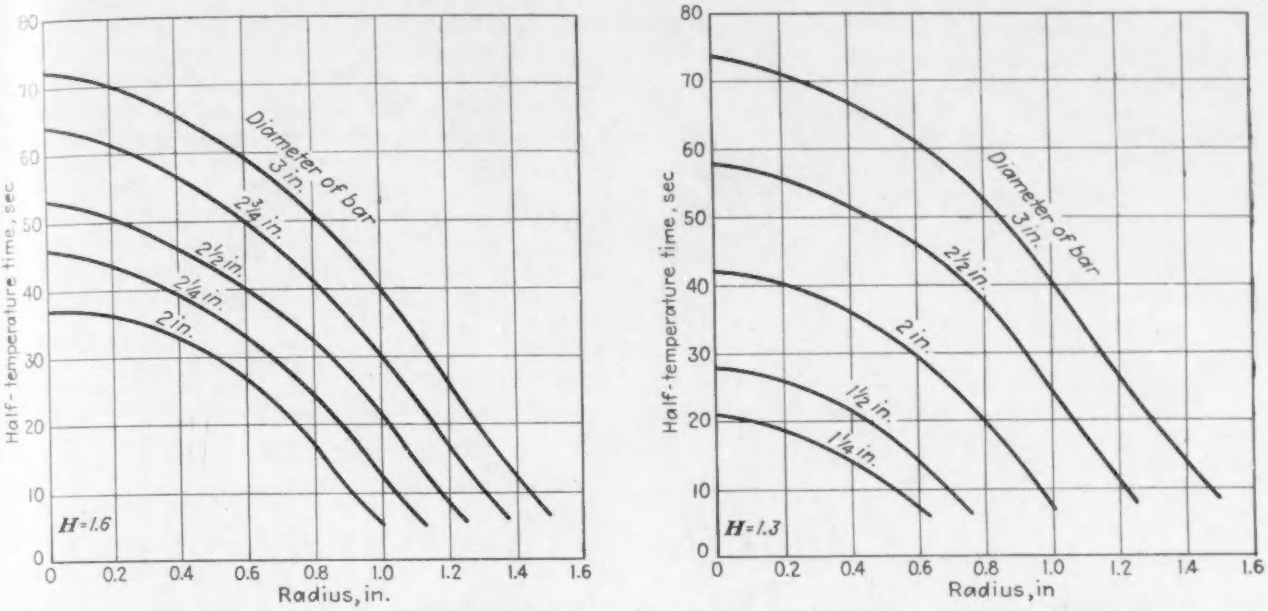
[3] See "Hardenability of Alloy Steels," published by American Society for Metals, Cleveland, 1939, p. 124-190: "Hardenability, Its Relation to Quenching," by M. A. Grossmann, M. Asimow, and S. F. Urban.

[4] See *Iron Age*, April 25, 1940, pp. 25-29 and May 2, 1940, pp. 39-45. "Hardenability and Quenching," by M. A. Grossmann and M. Asimow.

[5] See preprint of American Society for Metals, 1940: "Hardening Characteristics of Various Shapes," by M. Asimow and M. A. Grossmann,

[6] Personal communication, W. E. Jominy, General Motors Research Laboratories.

[7] See First Report of Alloy Steels Research Committee, Iron and Steel Institute (British), 1936, pp. 149-187, by T. F. Russell.



■ Fig. 3 – Half-temperature times at various positions from the surface to the center of various diameter Jominy bars at two values of H

simply a round bar cooling in air (removed, of course, from its situation of Fig. 2). The calculation is therefore carried out on the basis of cooling times in a quenched plate, for a plate 6 in. thick, the thickness being thus equal to twice the length of a 3-in. Jominy bar, combined with the air-cooling effect which is simply that of a 1-in. diameter bar cooling in air. (The air-cooling effect on the end (opposite the water-quenched end) presumably can be neglected because the area there is very small compared with the cylindrical surface of the rest of the bar, and the subsequent data will show that this assumption is permissible.) On the basis of the foregoing premises it would be possible to estimate the half-temperature times if the two severities of quench (water and air) were known. Previous experience[4] had shown that the water-quench, of the type applied on the end of the Jominy bar, might be expected to show a severity of quench in the neighborhood of $H = 2$ to $H = 5$. For the air-cooling, values around 0.02 had been found. As described in the Appendix, several different values of H in the neighborhood of the just-stated figures were employed, and those values adopted which fitted most closely the experimental points determined by Jominy. It turned out that using a value $H = 2.33$ for the water quench and $H = 0.022$ for the air cooling, a curve (Fig. 1) was obtained for the half-temperature times which accorded with Jominy's values very satisfactorily indeed.

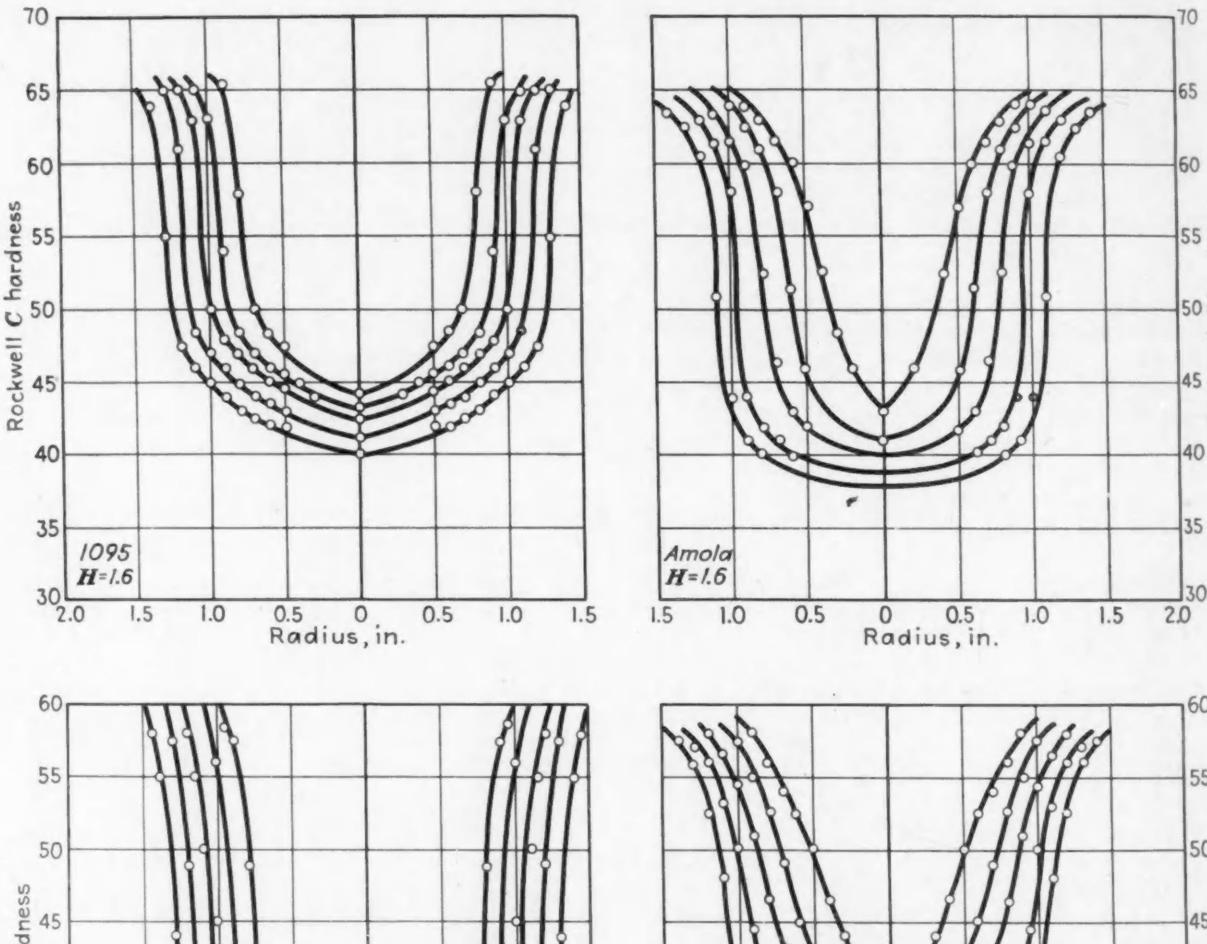
Table 1 – Half-Temperature Times at Various Distances from Water-Cooled End

Distance from Water-Cooled Face, in.		Time – Sec. To Cool to One-Half Temperature
1/16	0.0625	2.5
1/8	0.125	10.5
3/16	0.1875	16
1/4	0.250	22
3/8	0.375	33
1/2	0.500	52
5/8	0.625	66
3/4	0.750	81
1 1/8	1.125	130
1 1/4	1.250	150
2	2.00	224

■ Fig. 4 – Half-temperature time versus diameter of round in ideal quench

It may therefore be concluded with some confidence that the half-temperature times on the Jominy bar are substantially as shown in Fig. 1. To compare these values with those in quenched bars, it is necessary to know the half-temperature times at different positions (surface to center) within a bar when using a known severity of quench. The method for the bars is based on previous calculations as follows:

Suppose the bar under consideration is a 2-in. diameter round, quenched with a severity $H = 1.6$. It is desired to ascertain the half-temperature times at various positions from the surface to the center of such a bar when so quenched, so that a curve could be drawn, such as the lowest curve at the left in Fig. 3, which shows the half-temperature times from surface to center, for this size in this quench. As an example of one of the points on this curve, take the position at mid-radius, which in this case would be at 0.5 in. from the center. By the use of curves and data from previous publications[3, 4, 5], the cooling-time can be ascertained and, in this case, the half-temperature time is found to be 30 sec. Thus, at the midway position on a 2-in. bar, namely at 0.5 in. along the radius when the



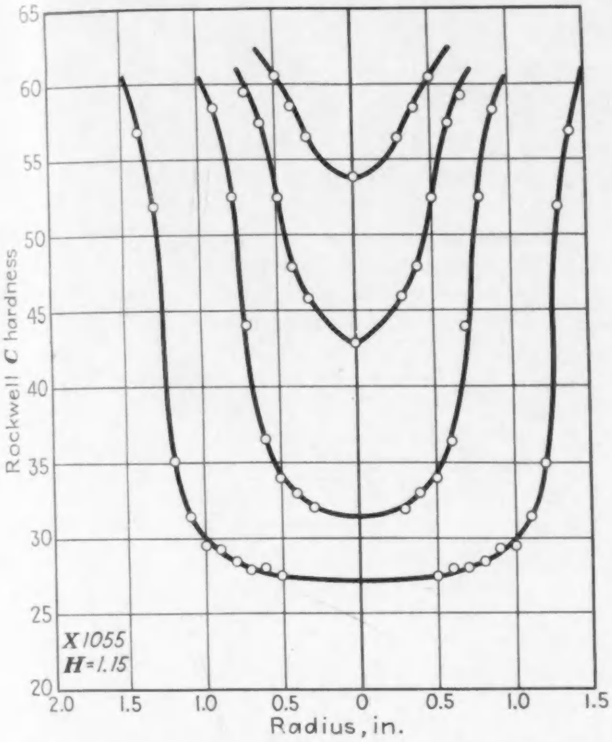
■ Fig. 5 – Rockwell C hardnesses at various distances from the center for different bar diameters and steels

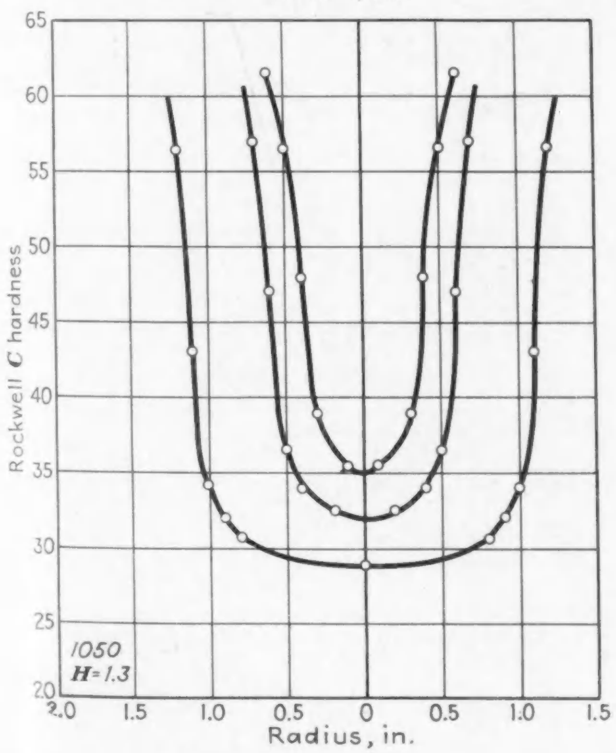
severity of quench is $H = 1.6$, the half-temperature cooling time is 30 sec.

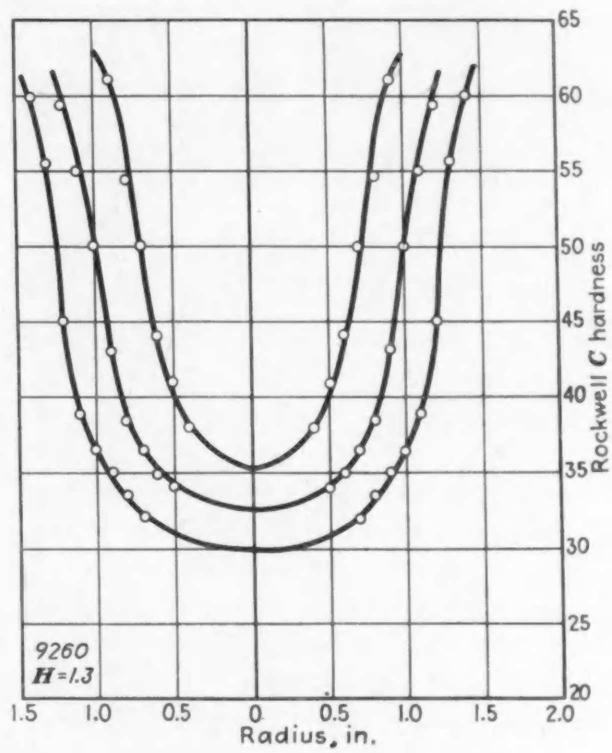
In the same way, the procedure is repeated for various positions in the 2-in. round, as well as for the different size rounds of Fig. 3, making it possible to draw the curves shown there.

As previously mentioned, it was found that the "half-temperature time" correlated well with the extent of hardening for any particular steel; hence the position in the Jominy bar which has a certain cooling time should exhibit the same hardness as the position in a round bar which has the same cooling-time (when using the same particular steel, for example, the 1045 steel of Fig. 5). The curves of Fig. 5, when appropriately examined,[3, 4] show that the severity of quench in those cases was $H = 1.6$. In Fig. 3, it was seen that the cooling time of 30 sec occurred at the midway position, namely 0.50 in. from the center, in the 2-in. round at $H = 1.6$. In Fig. 5 it is seen that, in the 2-in. round quenched $H = 1.6$, the hardness at 0.50 in. from the center is just about 30 Rockwell C. For the Jominy test, it is seen in Fig. 1 that a half-temperature time of 30 sec is encountered at the position 0.34 in. from the water-cooled end. Turning now to Fig. 7 showing the Jominy hardness values, at a position 0.34 in. from the water-cooled end on the 1045 steel, it is seen that the hardness is 31 Rockwell C, which is a very satisfactory check with the 30 Rockwell C obtained for the same estimated half-temperature time in the quenched round. In the same way, by the use of Figs. 1, 3, 5, 6 and 7, it is possible to check the whole series of Rockwell hardnesses versus half-temperature times, as was just done for one point, resulting in the comparisons shown not only for the 1045 steel in Fig. 8 but also for the other steels of Figs. 8 and 9.

It will be seen that the hardness values versus half-temperature times check very closely for all of the seven steels which were investigated, as shown in Figs. 8 and 9.







■ Fig. 6 – Rockwell C hardnesses at various distances from the center for different bar diameters and steels

Table 2–Compositions and Grain Size of Steels

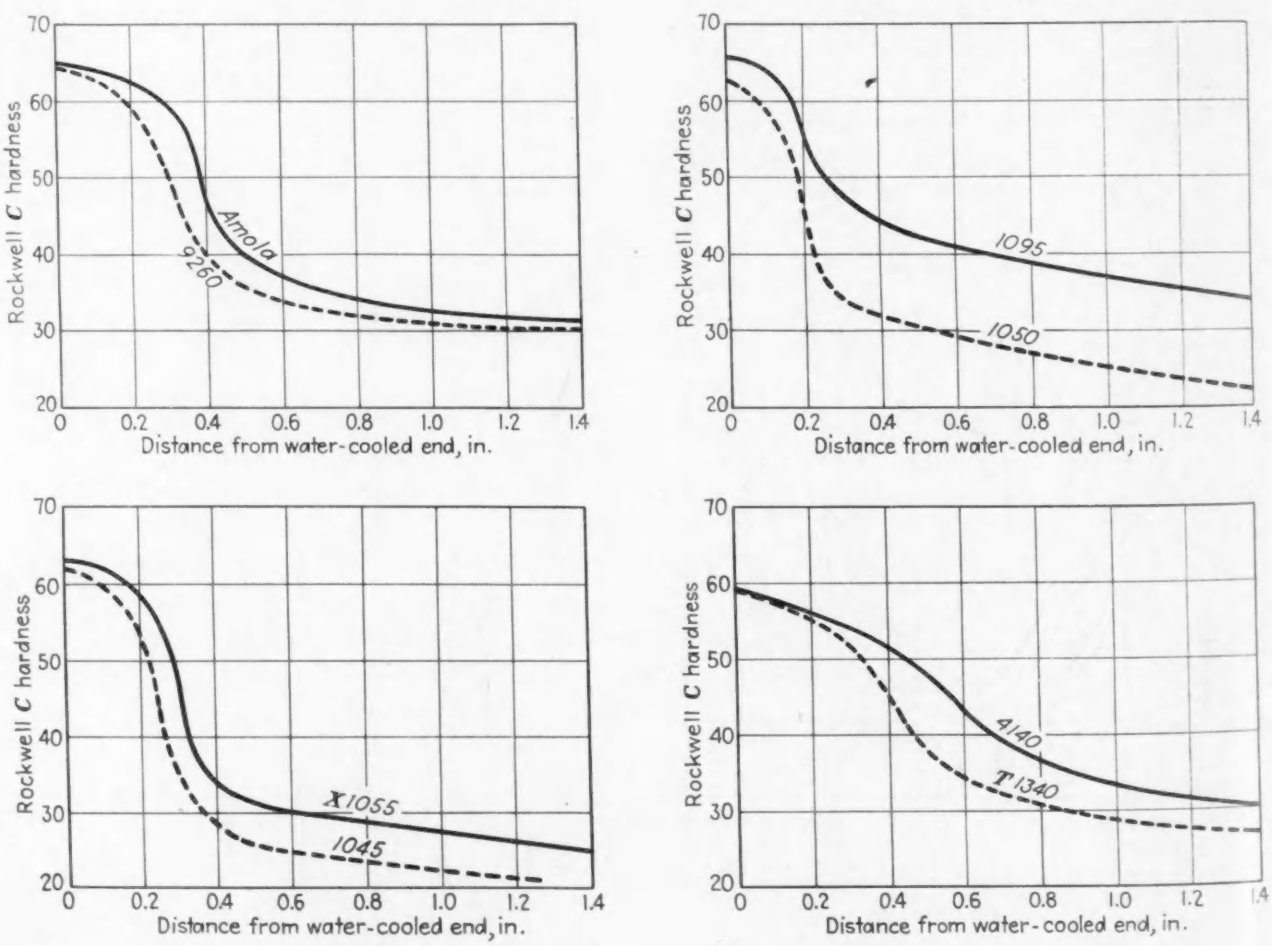
Steel Type	C	Mn	P	S	Si	Cu	Ni	Cr	Mo	McQuaid-Ehn Grain Size
1045	0.47	0.86	0.012	0.050	0.23	....	....	....	....	4
1050	0.50	0.80	0.022	0.022	0.24	0.03	0.03	....	....	7
X1055	0.51	1.05	0.014	0.014	0.29	....	....	0.21	....	6
1095	1.00	0.35	0.033	0.020	0.26	....	....	....	....	5/8
T1340	0.39	1.74	0.023	0.023	0.26	....	0.01	0.13	....	4/6
9260	0.57	0.68	0.019	0.019	2.00	....	....	0.17	....	6/8*
Amola	0.69	0.81	0.014	0.022	0.24	....	0.01	0.00	0.22	5/8

* Determined by oxidation method.

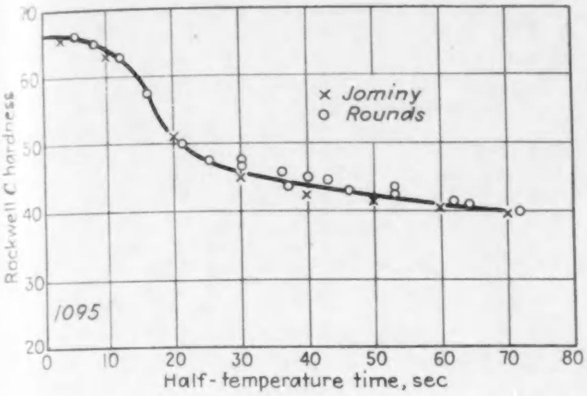
The compositions of these steels were as given in Table 2.

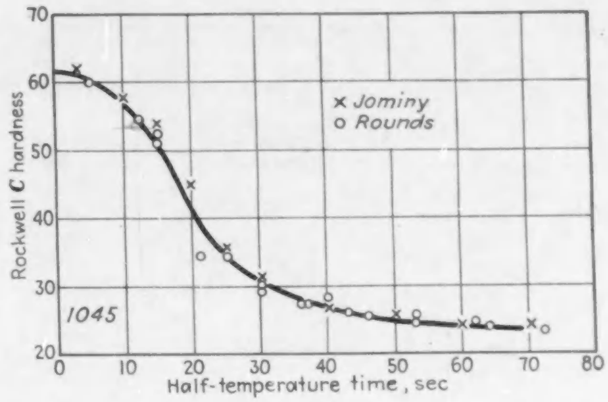
■ Hardness Distribution in Quenched Bars

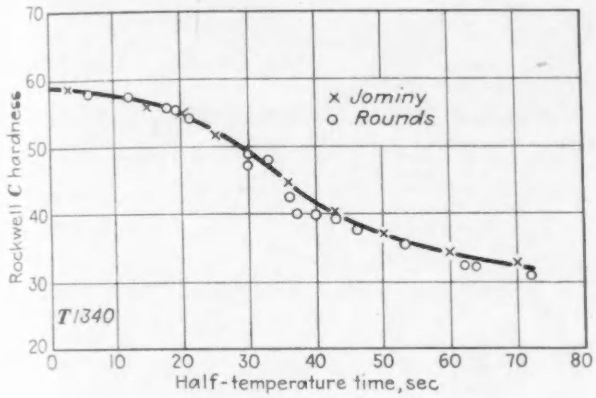
It will perhaps be obvious, once the hardness distribution has been ascertained in a Jominy bar, that the foregoing scheme can then be used to predict the hardness distribution in any size of bar quenched with any known severity of quench. For example, consider the 9260 steel, of which the Jominy hardness distribution is shown in Fig. 7. Suppose the problem is to predict the hardness distribution in a 2-in. round of this steel quenched with a severity of $H = 1.3$. Turn therefore to Fig. 3, in which the right-hand set of curves has been prepared to show the half-temperature time in seconds when the quench is $H = 1.3$. Choosing the 2-in. round (the middle curve), we may select, as one of the points to be determined, the position 0.7 in. from the center, and we find that at this position the half-temperature time is 25.5 sec. We turn now to Fig. 1, and find that, in the Jominy test, the cooling-time of 25.5 sec occurs at 0.30 in. from the quenched end. We may now use Fig. 7, where it is seen that, in the Jominy test of this 9260 steel, at 0.30 in. from the end the Rockwell C hardness was 50. In drawing the

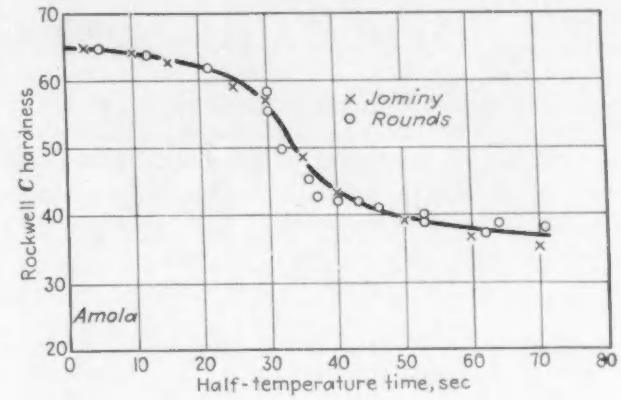


■ Fig. 7 – Jominy hardness values versus distance from water-cooled end for eight steels

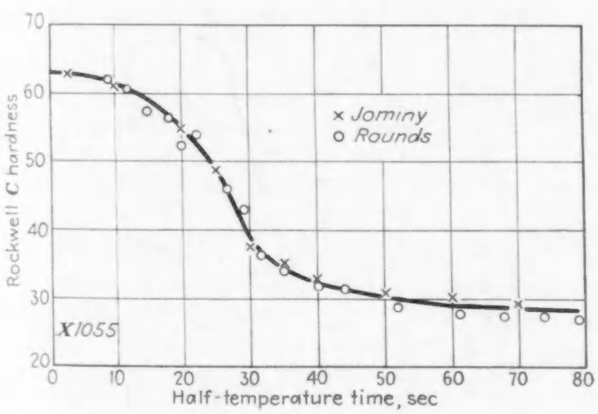






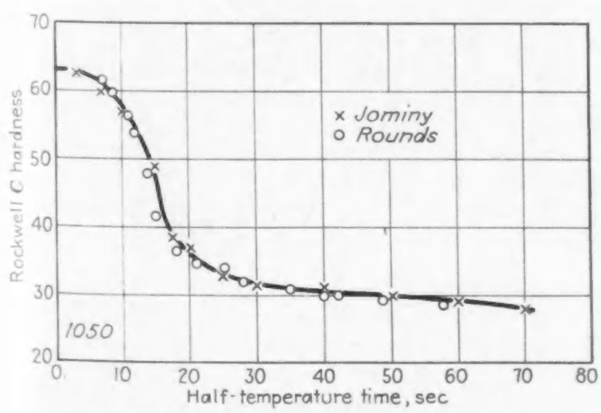


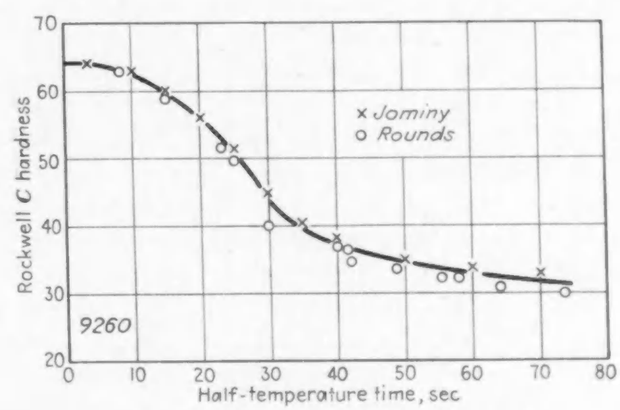
■ Fig. 8 (above) and Fig. 9 (below) – Rockwell C hardness versus half-time temperature – Jominy and rounds for seven steels



predicted hardness distribution curve for the 2-in. round quenched $H = 1.3$, we therefore plot this point at 50 Rockwell C, at 0.7 in. from the center, as indicated by the circle in Fig. 10. Repeating this process for a series of points across the section of the 2-in. round, one obtains the predicted hardness distribution curve shown as the dashed line marked 9260 steel in Fig. 10. For comparison, the actual hardnesses obtained by direct testing of the quenched round are shown in the accompanying full line in Fig. 10, and it is seen that the predicted and the actual values check very satisfactorily.

The curves for the other steels in Fig. 10 were obtained in the same manner. It may be noted that several different





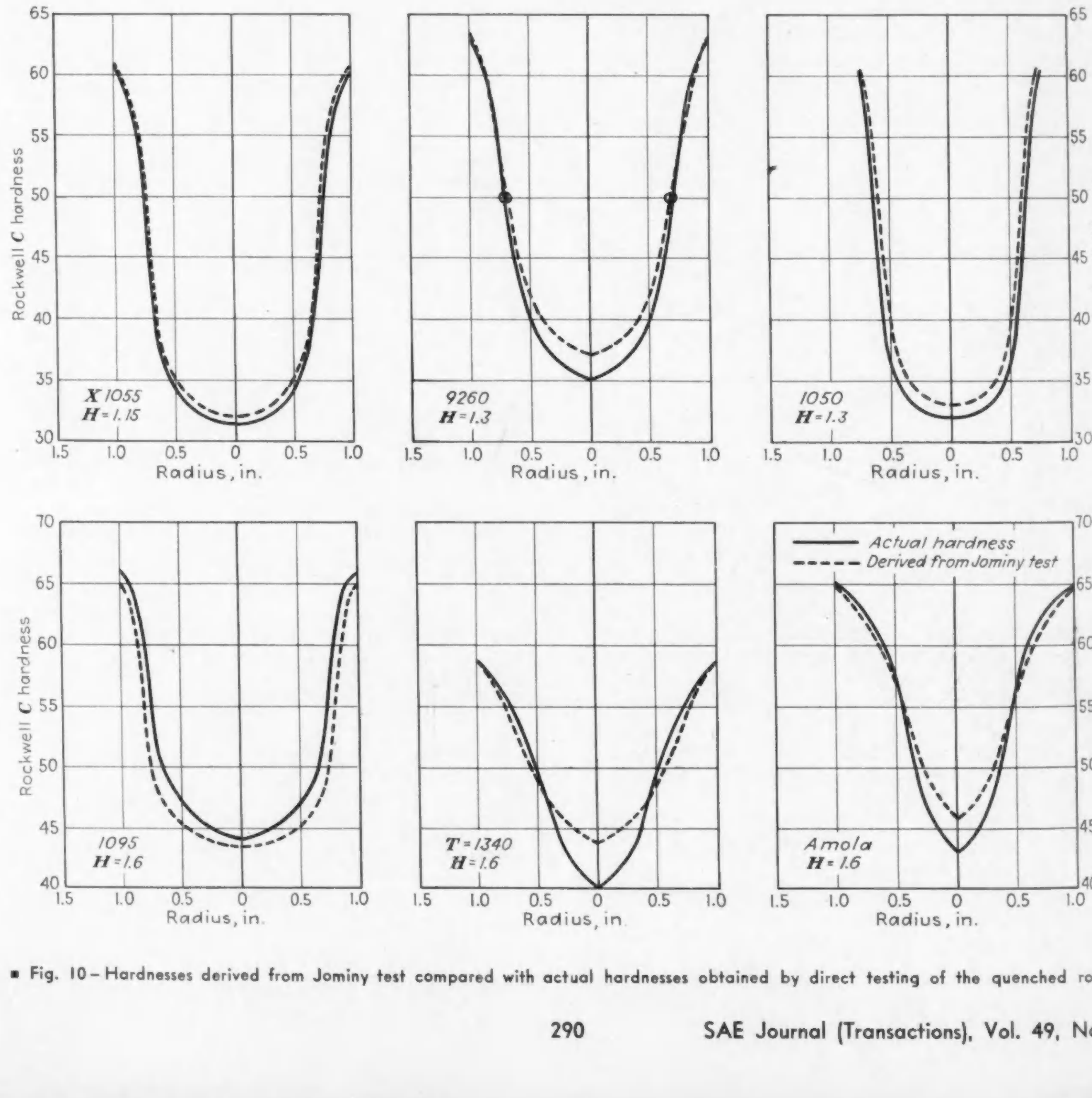
Steel Type	Critical Hardness 50% Martensite, Rockwell-C	Distance on Jominy Bar, In.	Ideal Critical Size From Jominy	Ideal Critical Size From Rounds	Critical Half-Temperature Time, Sec.
1045	43	0.255	1.95	1.90	21.
1050	44	0.202	1.72	1.65	16.5
X1055	44½	0.31	2.20	2.22	26.5
1095	54½	0.205	1.73	1.80	16.5
T1340	39½	0.47	2.83	2.77	44.
9260	46½	0.33	2.28	2.20	29.
Amola	51	0.38	2.48	2.40	34.

Table 3 – Ideal Critical Sizes for Seven Steels

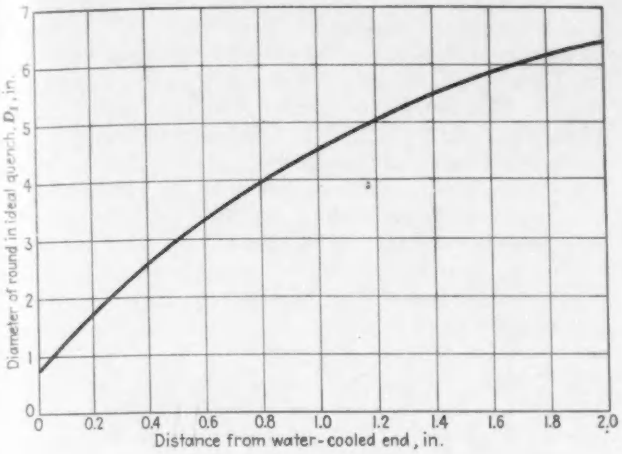
severities of quench are represented, calling for a particular set of "interpolation curves" like those of Fig. 3 for each particular quench. The interpolation curves of Fig. 3 could have been set up for direct reading of Jominy distance instead of time. That is to say, had the ordinates been Jominy distance instead of half-temperature time in seconds, it would then be possible to read directly from Fig. 7 to Fig. 3, instead of interpolating through Fig. 1. The results would of course have been the same.

"Ideal Critical Size" from Jominy Bar

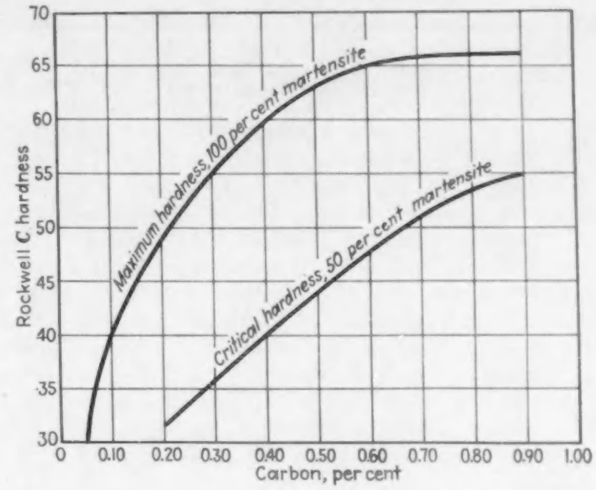
A convenient index which has been proposed[3, 4] as a measure of hardenability is the ideal critical size, which is defined as the size of bar in which the unhardened core



■ Fig. 10 – Hardnesses derived from Jominy test compared with actual hardnesses obtained by direct testing of the quenched rounds



■ **Fig. 11 – Correlation between hardenability in terms of ideal critical size and distance from water-cooled end**



■ **Fig. 12 – Maximum hardness and critical hardness at different carbon contents**

is just absent, when the steel is subjected to the fastest possible or ideal quench. Such a bar would be just half-hardened at the center, that is to say its structure at the center would be 50% martensite (balance fine pearlite, "nodular troostite").

Since, for some purposes, it is useful to know the hardenability in terms of ideal critical size, it would be convenient to be able to read this directly from the Jominy test. Such a correlation, Fig. 11, can be derived very simply from the cooling time.

The cooling time at the center of such an ideally quenched bar is related to the diameter of the bar according to the following relationship[3]:

$$\frac{D_I}{2\sqrt{a\,t}} = 2.229$$

where D is bar diameter, a is diffusivity and t is the half-temperature time. Assigning a value 0.009 sq in. per sec for diffusivity,[5] the formula becomes

$$D_I^2 = 0.179\,t$$

the resulting values being shown in Fig. 4. On this basis, the distance from the water-cooled end in the Jominy test, instead of being interpreted in terms of the half-temperature times as shown in Fig. 1, can be related directly to the diameters of rounds whose centers have these half-temperature cooling times. This results in the curve of Fig. 11.

Once the hardness distribution curves on the Jominy bars have been ascertained as in Fig. 7, the ideal critical size can then be read directly from Fig. 11 as follows: It has been found that the hardness of the 50% martensite structure is related quite regularly to the carbon content of the steel, somewhat as shown in Fig. 12. It is true that the presence of alloys may result in somewhat higher hardnesses for the 50% martensite structures in such steels, but Fig. 12 may be used as a fairly reliable guide. Therefore, for each steel as shown in Table 3, find in Fig. 12 the hardness of the 50% martensite structure, as recorded also in Table 3. Then read from Fig. 7 the distance for this hardness on the Jominy bar, finally reading from Fig. 11 the corresponding ideal critical size, as in Table 3.

For comparison, the ideal critical size was calculated also from the round bars, whose hardness distributions are shown in Figs. 5 and 6, and it is seen that the values accord very reasonably with those read directly from the Jominy bar. As an additional item, the half-temperature times in seconds are also recorded in Table 3, as read for these 50% martensite hardnesses from Fig. 7 and then Fig. 1.

APPENDIX

By M. Asimow and W. F. Craig

In the analytical theory of heat the differential equation for the flow of heat in a body of thermal diffusivity a is

$$\frac{1}{a}\,\frac{\delta T}{\delta t} = \frac{\delta^2 T}{\delta x^2} + \frac{\delta^2 T}{\delta y^2} + \frac{\delta^2 T}{\delta z^2} \qquad (1)$$

in which T is the temperature at time t of any point in the body whose coordinates are x, y, and z. In the case of an infinite plate in which the flow occurs in one direction only, (1) reduces to

$$\frac{1}{a}\,\frac{\delta T}{\delta t} = \frac{\delta^2 T}{\delta x^2} \qquad (2)$$

In cylindrical coordinates for the flow of heat in an infinitely long cylinder equation (1) becomes

$$\frac{1}{a}\,\frac{\delta T}{\delta t} = \frac{\delta^2 T}{\delta r^2} + \frac{1}{r}\,\frac{\delta T}{\delta r} \qquad (3)$$

in which T is the temperature at a distance r from the axis.

The following conditions are assumed in obtaining solutions of Equations (2) and (3) which apply to the problem at hand, that is, quenching both sides of an infinite plate and air cooling an infinitely long cylinder, the simultaneous effects of these two modes of cooling being analogous to the cooling of the Jominy bar:

(1) The body is initially at a uniform temperature T_0 and is then placed in surroundings which have the same temperature as that which the body is ultimately to attain, T and T_0 being calculated as the excess over this temperature.

Table 4 – Half-Temperature Times at Various Positions Along Bar

Distance From Water-Cooled Face, in.	$w = \left(\frac{T}{T_0}\right)_W$	$a = \left(\frac{T}{T_0}\right)_A$	$wxa = \frac{T}{T_0}$	Time, sec.
0.0	0.505	0.992	0.5	3
0.15	0.514	0.973	0.5	12
0.3	0.525	0.947	0.5	26
0.6	0.555	0.901	0.5	60½
0.9	0.585	0.855	0.5	101
1.2	0.620	0.810	0.5	141
1.5	0.650	0.770	0.5	178

(2) The numerical value of thermal diffusivity does not vary with temperature.

(3) Newton's law of cooling holds, that is, the rate of heat abstraction from the body is proportional to the difference in temperature between the surface of the body and its surroundings.

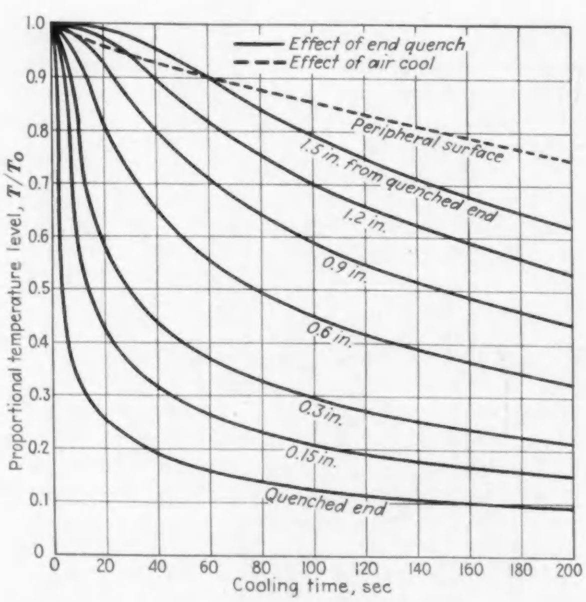
The solution of Equation (2) which satisfies the conditions stated is

$$\frac{T}{T_o} = \sum_{k=1}^{k=\infty} \left[\frac{2 \sin n_k}{n_k + \sin n_k \cos n_k} \right] \left[e^{-\frac{n_k^2}{L^2} a t} \right] \left[\cos \left(n_k \frac{x}{L} \right) \right] \tag{4}$$

in which n_k assumes all of the values which satisfy the transcendental boundary condition equation

$$n \tan n = 2HL \tag{5}$$

where H is the severity of quench. Equations (4) and (5) applied to the Jominy bar state that the end quench produces a proportional temperature drop to T/T_o in time t at a given location x (measured from the slow-cooled end) which depends upon the severity of quench H, upon the length of the bar L, and upon the diffusity of the steel a.



■ Fig. 13 – Calculated cooling curves – Proportional temperature level versus cooling time for various positions along bar

The numerical calculations may be made provided that the proper values of thermal diffusivity and severities of quench are known. Previous experiments indicate that a satisfactory value of diffusivity is 0.009 in.²/sec which is an average of the exact values for the different structures encountered through the range of temperatures involved. The best fit of the calculated and experimental time curves was obtained with a value of $H = 2.33$ for the end-quench and $H = 0.022$ for air-cooling.

The calculated cooling curves from which half-temperature time values may be determined readily as indicated above just shown in Fig. 13. The proportional temperature resulting from an end quench of severity $H = 2.33$ is shown as a function of time for various positions along the bar. Superimposed is the proportional temperature on the periphery resulting from air cooling with a severity of $H = 0.022$ also plotted as a function of time. Much of the laborious numerical work associated with the use of Equations (4) and (5) was eliminated by the generous use of the published tables of T. F. Russell[7].

In Table 4 is shown the final calculation of half-temperature time values at a number of positions along the bar as determined from Fig. 13.

Summarizes and Interprets All Four Papers

–W. E. Jominy

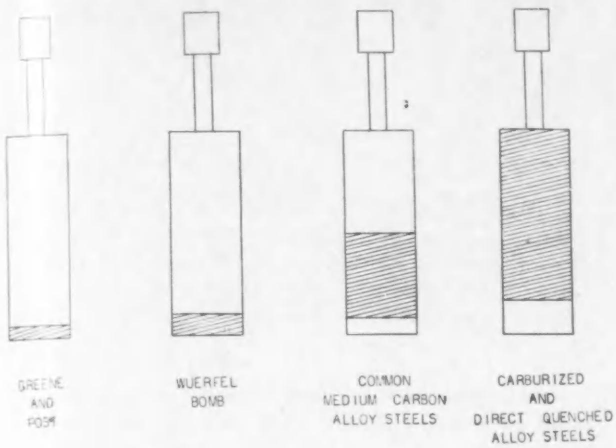
Metallurgical Department, Chrysler Corp.

THE question as to what hardness shall be chosen as the limiting value in making hardenability tests has come up several times. For some steels the point containing 50% martensite corresponding to Asimow and Grossmann's critical bar size is defined quite sharply whereas, for other steels, mainly the deeper hardening types, the point of 50% martensite is difficult to determine accurately. Although this limiting value can be used on the end-cooled bar, we have found it more convenient to use some minimum hardness value which would indicate a largely martensitic structure, and thus be assured of good physical properties to the limiting point. There are many automotive parts whose minimum hardness is required to be far in excess of that obtained with 50% martensite and certainly in these cases it is imperative to know the limiting cooling rates which will give us the required hardness for the steel being tested.

Since the test piece of Greene and Post is cooled on the end faces as well as the sides, it is probable that the slowest cooling obtainable is equivalent to the center of a 1⅛-in. round brine-quenched bar. According to Scott, the cooling rate at the center of a water-cooled 1-in. round bar is 108 F sec at 1328 F. This is about twice the cooling rate given by Greene and post for this location, and it appears that the cooling rates given by them at other locations are also about one-half of what we should expect.

The test bar of McCleary and Wuerfel has as its lowest cooling rate the equivalent of a 1⅜-in. round bar quenched in water as indicated from their Fig. 2. To give an idea of the range in cooling rate covered by the test bar of Greene and Post and that of McCleary and Wuerfel, I have prepared Fig. A. It will be plainly seen, as suggested by the authors, that these test bars are for relatively shallow-hardening steels and most of the medium-carbon alloy steels are too deep-hardening to be measured by them. It is apparent also that they cover the same range of cooling rates, with the McCleary-Wuerfel bar including a little more territory.

For shallow-hardening steels which fall into the range of 0 to ¼ in. on our standard bar we have used our so-called L bar with good success. We have used it to evaluate the hardenability of



■ **Fig. A (Jominy discussion) – Ranges covered by hardenability bar**

water-hardening tool steels for the past two years and have had no difficulty whatever in correlating the tests with expected results.

We are indebted to Asimow, Craig, and Grossmann for their Fig. 12 giving us an approximate hardness range for steels of known carbon content containing 50% martensite. We are glad also to have the cooling time at locations from surface to center of bars from $1\frac{1}{4}$ to 3-in. round as shown in Fig. 3 and, in addition, to have defined in terms of cooling time the center cooling rate of bars "in ideal quench" as shown in Fig. 4. By means of the end-cooled bar these may be converted to cooling rates by using Fig. 11 and previously published data on the end-cooled bar. On the basis of work done in our laboratory and that done by French and by Scott, these cooling times appear rather fast and we wonder whether a curve parallel to that on Fig. 11 but a little lower might not give a more accurate relationship. We realize that this change would require a change also in Fig. 1 or Fig. 4 or both, but we believe it would be worth while to scrutinize these data carefully to avoid future errors.

New Automobile Aids to Aircraft Stressed

Pointing out that the aircraft industry's production methods are just as up to date and efficient as those of the automobile industry in view of the quantities that it has had to handle, John H. Van Deventer, president and editor, "Iron Age," stressed the important contributions that the automobile industry already has made to aircraft production, in a paper: "Tolerance versus Tolerances," presented before the Detroit Section of the Society, Detroit, Mich., May 19, 1941. Excerpts follow:

EVEN at this stage of the game, the automobile industry has made important contributions to its ally, the aircraft industry. Chrysler, for example, has licked the problem of additional aluminum forging capacity, supposed to have been unsolvable; Ford and Dow Chemical, the problem of magnesium castings.

Briggs is showing the way to the use of mechanical dies in the production of airframe parts, thus getting away from the drop-hammer and the rubber-die technique which requires enormous press capacity and also producing a product requiring less afterwork.

Members of the automobile industry have succeeded in gaining the acceptance of rolled threads as superior in strength to cut threads and vast improvements are being worked out through the abolition of rivets in both stressed and non-stressed structures through spotwelding.

Packard has had the heroic job, now nearing completion, of Americanizing the production of the Rolls-Royce engine. So effectively has the progressive assembly schedule been worked out that, so Col. Vincent informs me, there will be but 35 min available for final mechanical inspection of the motor as it goes down the line.

One of the large contributions in technique that has already been made is the elimination of the "two-legged" shop drawing in airframe part construction.

The two-legged shop drawing is the liaison man from the engineering office of the aircraft designer who runs down to the shop to explain what a drawing means when the production man cannot interpret it. The automobile industry used to have these people when it was small.

Today, in the automobile industry such men do not exist, having been replaced by complete sets of detailed working drawings, on which all limits and all changes are recorded. Such drawings, when sent to a sub-contractor or supplier, enable him to know exactly what is wanted and to produce it to fit.

Up to now, the aircraft industry, dealing in small lots and not interested in suppliers or sub-contractors, has largely depended upon the "loft" system, similar to that employed in shipbuilding. A huge master layout has been prepared on sheet metal showing the principal outlines. Templates were laid out from this layout when necessary and were the only pieces of concrete information transmitted to the shops. In addition, there were some sketches but no detail or assembly drawings.

To bridge the gap, the aircraft makers developed large crews of liaison men to explain what the engineers really had in mind.

Murray and Briggs both have contributed largely to the modernization of this system. The former is expending $30,000 alone in the construction of working drawings of the inner wing panel that it will make for Douglas. The latter has sent a crew of its draftsmen to the Vought-Sikorsky plant to make similar drawings of the parts it has engaged to make.

Another vitally important help to the aircraft industry will come from utilization of the automobile industry's knowledge of sub-contracting.

Through its association with the automobile industry, the aircraft makers will learn how to turn out planes faster and the automobile industry will learn how to make better cars. That, in itself, ought to make the joint effort self-liquidating and worth while aside from its necessity. But that is only a part of the economic benefit.

What we are really going to do is to learn how to extend the benefits of mass production, hitherto largely confined to huge production lots, to much smaller quantities. And, as a result of what we learn, innumerable products will be made in the future at substantially lower cost to the benefit of all American consumers and to the reinforcement of our competitive position in the world's future markets.

Standardization of AIRCRAFT- E

STANDARDIZATION of engine components should start in the drafting room with use of a system of sample drawings, Mr. Carvelli contends, and dimensioning of parts should be simplified through use of a two-place decimal system.

He emphasizes the importance of standardization of notes, clearances, tolerances, and other data listed on drawings. In addition, threaded parts, gear tooth form, and many such items can and should be standardized, and serious consideration should be given to adoption of the metric system.

Since the Army-Navy (AN) Standards were developed primarily for airplanes and often do not apply to aircraft engines, he points out that a new set of standards must be developed for parts used on engines only.

■ ■ ■

STANDARDIZATION of engine components should start in the drafting room. To insure uniformity in regard to method of dimensioning and standardization of data and notes, a system of sample drawings may be used. For example, if a drawing is to be made of a connecting rod, crankshaft, piston, piston ring, and so on, the draftsman should refer to a list showing the sample drawing number for each particular part. The drawing may be obtained from file and used as a guide in producing the drawing. Fig. 1 shows one of these lists. To guide the draftsman in selecting the proper sample drawing, a remarks column is used with a description of the part.

■ Dimensioning Standardized

To simplify dimensioning of parts, a two-place decimal system as shown in Fig. 2 will facilitate the work in the engineering and production departments as well as in the manufacturing department. Considering a large casting, dimensioned to the fractional system, the fractional dimension may have to be converted into decimals as many as 11 times, thus the decimal system facilitates the work for: (1) layout men, (2) detailers, (3) checkers, (4) tool designers, (5) tool-design checkers, (6) pattern layout men, (7) pattern maker, (8) layout men of castings for checking, (9) layout men of first casting for machining, (10) layout men for tools and (11) checkers of tools. To appreciate the value of the two-decimal system it is only necessary to note that, when any of the common fractions in use today must be changed into decimals thousands of times a day in any large organization, the simplicity of the two-decimal system will be appreciated. For example, a fraction such as 11/64 to be expressed clearly would have to be carried out to six decimal places, 0.171875; therefore, when it is necessary to add long columns of common fractions as must be done repeatedly in the engineering, production, and manufacturing departments to obtain accuracy of parts made from them, the chances for error are obviously multiplied in addition to the combination involved through requiring the use of converting tables. A two-decimal system will make the drawings easier to read due to the smaller number of figures used for each dimension. The simplicity of the two-decimal system becomes even more valuable when it is required to determine dimensions through the use of trigonometrical fractions.

Standardization of notes on drawings is important. A simple item like breaking edges may prove costly unless treated properly. A drilled hole in a cast or forged flange may not have to have the edges broken or broken only 0.003 to 0.015 in., but the hole in a highly stressed crankshaft or connecting rod should have the edges broken 0.06 to 0.10 in. approximate radius. Specifying approximate radius is sometimes important as a chamfer with sharp edges may be as detrimental as not breaking the edges at

[This paper was presented at the National Aircraft Production Meeting of the Society, Los Angeles, Calif., November 2, 1940.]

NAME	PART NO.	(*)	REMARKS
BEARING - Roller	155D40	C	Max. Type Two lips on outer race. One lip & plate on inner race. One piece Cage One side of inner race to opposite side of outer race controlled by tolerances.
BEARING - Roller	155D25	C	Conrad Type Two lips on outer and inner race. Riveted Cage
BUSHING - Split	2119D	-	Split type. Hard rolled Bronze. These bushings are generally purchased, and all dimensions for each new application must be procured from vendor as the thicknesses etc. vary slightly with diameter and length.
CONNECTING ROD - Master	65265	H	
COUNTERWEIGHT - Front	65664	D	
COUNTERWEIGHT - Rear	64307	D	
CRANKSHAFT - Front End	66460	E	
CRANKSHAFT - Machining & Balancing Assy.	411666	E	
CRANKSHAFT - Rear End	65980	D	
END - Push Rod Ball	66039	C	
GUIDE - Exhaust Valve	29982	C	
PIN - Knuckle	65269	C	Angular oil holes are undesirable but appear unavoidable on this design.
PLATE - Engine Name	66686	-	Single Row Radial Engines.

■ Fig. 1 – Sample drawing list

ENGINE COMPONENTS

by GUSTAF CARVELLI
Wright Aeronautical Corp.

all. The following three notes are typical samples of notes referring to break edges:

NOTE

BREAK SHARP EDGES .01 – .03 APPROX. R UNLESS OTHERWISE SPECIFIED.

BREAK SHARP EDGES .003 – .015 UNLESS OTHERWISE SPECIFIED.

REMARKS

For general use, but sizes may be changed where larger APPROX. R is required.

For lightly stressed parts requiring slightly more than a burring operation.

BREAK SHARP EDGES .003 – .015 ON ALL (*) UNLESS OTHERWISE SPECIFIED.

(*) Insert one or more of the following as required:

DRILLED HOLES
CHAMFERS
COUNTERSINKS

This note is only for use with Note No. 2 and should not be used until the location of the holes, chamfers, or countersinks has been carefully considered, as the application of this note to such points when located in a highly stressed part or section is dangerous and may prove very costly.

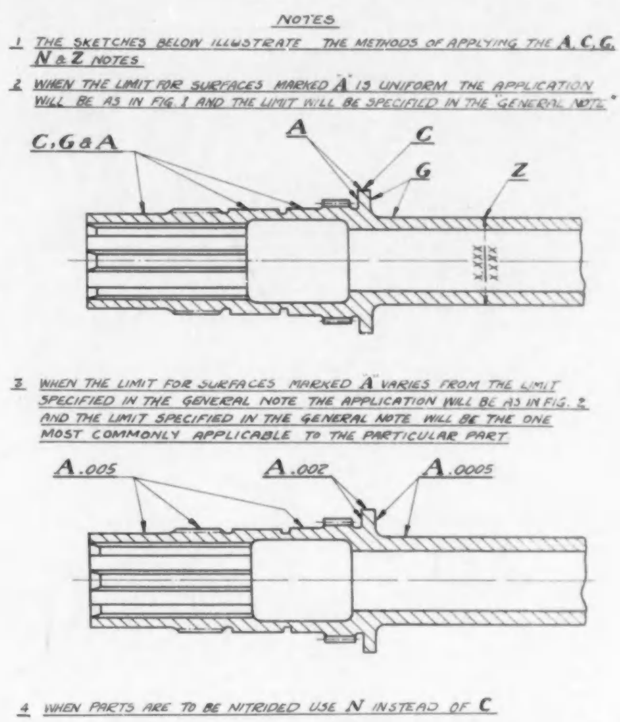
A note quite commonly used on drawings is "remove burrs." This note should be used on sheet-metal parts and tubings only.

Many parts in an aircraft engine are held together with studs and, in order to control the accuracy of the stud

64 THS	32 NDS	16 THS		DECIMAL EQUIVALENT	TWO-PLACE DECIMAL EQUIVALENT	64 THS	32 NDS	16 THS		DECIMAL EQUIVALENT	TWO-PLACE DECIMAL EQUIVALENT
					.01						
1				.0156	.02	33				.5156	.52
	1			.0312	.03		17			.5312	.54
3				.0468	.04	35				.5468	.54
		1		.0625	.06			9		.5625	.56
5				.0781	.08	37				.5781	.58
	3			.0937	.10		19			.5937	.60
7				.1093	.10	39				.6093	.60
			1/8	.125	.12				5/8	.625	.62
9				.1406	.14	41				.6406	.64
	5			.1562	.16		21			.6562	.66
11				.1718	.18	43				.6718	.68
		3		.1875	.18			11		.6875	.68
13				.2031	.20	45				.7031	.70
	7			.2187	.22		23			.7187	.72
15				.2343	.24	47				.7343	.74
			1/4	.25	.25				3/4	.75	.75
17				.2656	.26	49				.7656	.76
	9			.2812	.28		25			.7812	.78
19				.2968	.30	51				.7968	.80
		5		.3125	.32			13		.8125	.82
21				.3281	.32	53				.8281	.82
	11			.3437	.34		27			.8437	.84
23				.3593	.36	55				.8593	.86
			3/8	.375	.38				7/8	.875	.88
25				.3906	.40	57				.8906	.90
	13			.4062	.40		29			.9062	.90
27				.4218	.42	59				.9218	.92
		7		.4375	.44			15		.9375	.94
29				.4531	.46	61				.9531	.96
	15			.4687	.46		31			.9687	.96
31				.4843	.48	63				.9843	.98
			1/2	.5	.50				1	1.	1.

Even decimals are preferred where fractional figures would normally apply and should be used except in special cases

■ Fig. 2 – Conversion table – Fractions to two-place decimals



■ Fig. 3 – Control of accuracy of surfaces in relation to each other and in relation to a spline or gear

itself as well as the stud when assembled, the following note is applicable to studs and should be indicated on the detail drawings:

NOTE

PD OF THREADS MUST BE CONCENTRIC, SQUARE & TRUE TO EACH OTHER WITHIN .XXX IN 1 INCH, AND BODY WITHIN .XXX FULL INDICATOR READING WHEN MOUNTED ON CENTERS OR EQUIVALENT.

REMARKS

Ground thd. tolerance	.003
Die-cut thd. tolerance	.005
Rolled thd. tolerance	.003
Body tolerance	.006

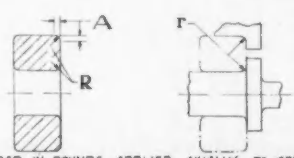
The accuracy of the tapped hole in relation to the surface should be controlled by a note specifying that the threads must be square with the top surface within .XXX in 1 in. This will insure accuracy of the assembled stud in relation to the bolting face as straightening of studs to clear the hole of the mating part should not be permitted.

Fig. 3 – The accuracy of surfaces in relation to each

other, and also in relation to a spline or a gear can be controlled by an "A" note:

NOTE

A SURFACES AND PD OF (*) MUST BE CONCENTRIC, PARALLEL, FLAT, SQUARE AND TRUE (AS APPLICABLE) TO EACH OTHER WITHIN .XXX FULL INDICATOR READING UNLESS OTHERWISE SPECIFIED.



P- THRUST LOAD IN POUNDS APPLIED AXIALLY TO OBTAIN TOTAL END PLAY OF INNER BALL RACE IN RELATION TO OUTER BALL RACE AND SHOULD BE EXPRESSED ON DRAWINGS AS FOLLOWS:-

"END MOVEMENT (TOTAL) UNDER (P) LBS THRUST LOAD."

NOMINAL BORE		LIGHT SERIES				MEDIUM SERIES				HEAVY SERIES			
MM.	DECIMAL	A		r	P	A		r	P	A		r	P
5	.197	03 02		.016 .005	5.5								
6	.236	•		•	•								
7	.276	•		•	•								
8	.315	•		•	•								
9	.354	05 03		.024 .010	•								
10	.394	•		•	•	.05 .03		.024 .010	5.5				
12	.472	•		•	•	.08 .05		.039 .020	•				
15	.591	•		•	•	•		•	•				
17	.669	08 .05		.039 .020	•	•		•	•	.08 .05		.039 .020	11
20	.787	•		•	•	•		•	11	•		•	•
25	.984	•		•	11	•		•	•	.10 .07		.059 .040	•
30	1.181	•		•	•	•		•	•	•		•	•
35	1.378	•		•	•	.10 .07		.059 .040	•	•		•	•
40	1.575	•		•	•	•		•	•	.12 .09		.079 .050	•
45	1.772	•		•	•	•		•	•	•		•	•
50	1.969	•		•	•	.12 .09		.079 .050	•	•		•	22
55	2.165	.10 .07		.059 .040	•	•		•	•	•		•	•
60	2.362	•		•	•	•		•	22	•		•	•
65	2.559	•		•	•	•		•	•	•		•	•
70	2.756	•		•	22	•		•	•	.14 .11		.098 .070	•
75	2.953	•		•	•	•		•	•	•		•	•
80	3.150	12 09		.079 050	22	.12 .09		.079 .050	22	.14 .11		.098 .070	22
85	3.346	•		•	•	.14 .11		.098 .070	•	.18 .14		.118 .090	44
90	3.543	•		•	•	•		•	•	•		•	•
95	3.740	•		•	•	•		•	•	•		•	•
100	3.937	•		•	•	•		•	44			•	•
105	4.134	•		•	•	•		•	•	•		•	•
110	4.331	•		•	•	•		•	•	•		•	•
120	4.724	•		•	44	•		•	•	.22 .18		.157 .120	•
130	5.118	.14 .11		.098 .070	•	.18 .14		.118 .090	•	•		•	•
140	5.512	•		•	•	•		•	•	•		•	•
150	5.906	•		•	•	•		•	•	•		•	•
160	6.299	•		•	•	•		•	•	•		•	•
170	6.693	18 14		.118 .090	•	•		•	•	•		•	•
180	7.087	•		•	•	•		•	•	.26 .22		.197 .160	•
190	7.480	•		•	•	.22 .18		.157 .120	•	•		•	•
200	7.874	•		•	•	•		•	•	•		•	•
220	8.661	•		•	•	•		•	•	•		•	•
240	9.449	•		•		•		•	•	•		•	•
260	10.236	.22 .18		.157 .120	•	.26 .22		.197 .160	•	•		•	•
280	11.024	•		•	•	•		•	•	•		•	•
300	11.811	•		•	•	•		•	•	•		•	•

■ Fig. 4 – Data for ball and roller bearings having a bore of from 5 to 300 mm

REMARKS

A general control for the points of accuracy indicated. When the indicator reading shown in the note does not apply to all points, the amount "OTHERWISE SPECIFIED" can be shown on the body of the drawing.

(*) Insert one or more of the following as required:

GEAR – GEARS
SPLINE – SPLINES
THREAD – THREADS

The accuracy of a crankpin or a highly loaded journal should be controlled by a *Z* note specifying the circularity of the pin as well as the taper.

NOTE

Z DIAMETERS MUST BE CIRCULAR WITHIN .XXXX PARALLEL WITHIN .XXXX AND STRAIGHT WITHIN .XXXX.

REMARKS

For diameters, the circularity, taper, and straightness of which must be held closer than which the diameter tolerance or *A* note would insure.

The heat-treatment of small parts is controlled by specifying that parts should be heat-treated to a Rockwell reading of C19-26, 26-32, 32-36, 36-40, 40-45 or 46-51. Six degrees of hardnesses have been adopted in order to simplify heat-treatment of smaller parts as this will enable the production department to heat-treat many different small parts at the same time. The heat-treatment of larger parts such as crankcase, crankshaft, propeller shaft, connecting rod, and so on, is controlled by a Brinell reading note. The depth of the case on carburized parts is controlled as follows:

■ Depth of Case

The depth of finished case as specified on detail drawings shall be governed by the design. The points of control for establishing this depth are listed below in the order of their importance:

(a) Cams
(b) Gear Teeth
(c) Splines

When the detail is, or incorporates, a cam, the depth of finished case shall be as shown below for "Cams."

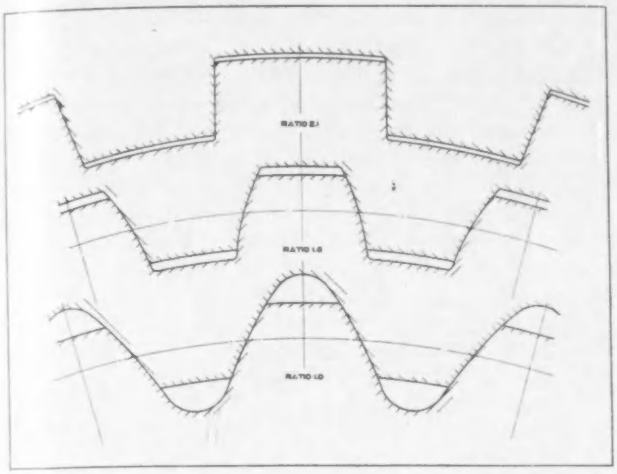
Details of parts having gear or spline teeth shall have a depth of finished case on each surface equivalent to 15% of the chordal thickness of the gear or spline tooth, with a maximum of 0.045. When detail includes both gear and spline teeth, the calculations are to be based on the gear teeth.

For the purpose of uniformity the figures specified on drawings should be the combination listed below having the mean figure nearest to the calculated amount.

Deviations from this standard will sometimes be necessary, but should only be made with the permission of the Chief Metallurgist.

0.005-0.010
0.010-0.015 For very thin sections

0.015-0.025
0.025-0.035
0.035-0.045 For General Work
0.055-0.065



■ Fig. 5 – Stress concentration ratios in three types of splines

0.025-0.045 For Crankshafts

0.035-0.065 For Cams

Special conditions of each piece such as thickness of metal under gear or spline teeth, adjacent walls, or very thin sections must be taken into consideration when establishing depth of case.

The assembly of a ball or roller bearing may not function satisfactorily unless the outer corners of the outer rings and the inner corners of the inner ring, the radii in the housing, and the shafts are controlled properly. Fig. 4 shows a list with these dimensions for bearing sizes having a bore of from 5 to 300 mm. The maximum radii in the housing and on the shaft must be smaller than the *A* dimension on the bearing, but it is also very important to specify as large radii as possible on the shaft in order to decrease the stress concentration caused by the change in shaft diameter.

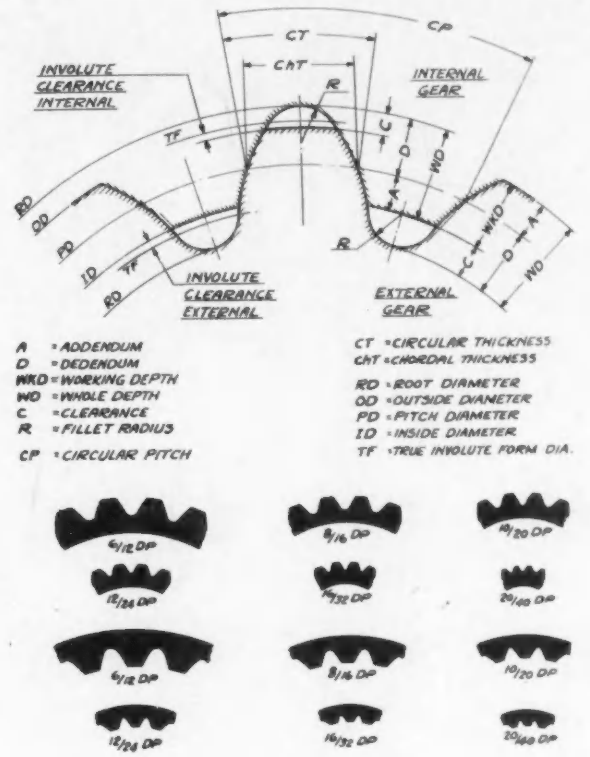
■ Shaft Ends Standardized Early

One of the first major parts to be standardized was propeller shaft ends and we are all familiar with the expressions: "SAE Shaft Ends Nos. 10, 20, 30, and so on." Shaft ends Nos. 10 to 50 inclusive are of the straight-sided spline type, the No. 60, of the involute-type spline. It is suggested that any new or larger shaft ends be of the involute type as it is more adaptable for large production with greater accuracy than the straight-sided type; also, the involute type with the full radius at the root has a lower stress concentration factor than the straight-sided type.

The 18-mm thread for the spark plug is an established standard and is also an international standard, one of the first international standards used in our aircraft engines. The following accessory drives and mountings have been standardized and are also SAE Standards: starters, magnetos, generators, governors, vacuum pumps, fuel pumps and tachometers of the mechanical and electrical type.

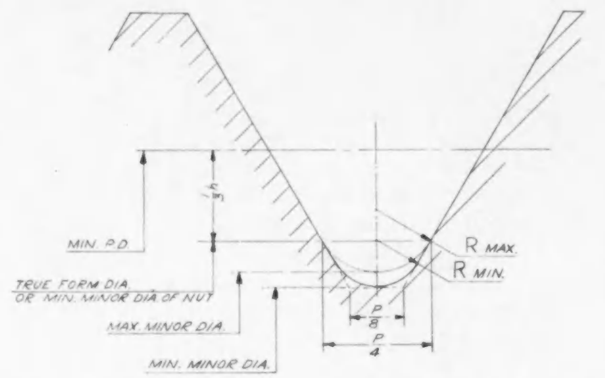
The method of transmitting power from a shaft to a gear through a key was early recognized as being unsatisfactory. The next step was to use SAE straight-side splines. For many applications, this type was not satisfactory; therefore, a 20-deg stub tooth form was used. Both of these splines have high stress concentration at the root of the spline and, to overcome this, work has been done on a 30-deg involute spline. The advantage of the 30-deg involute spline over the previous type is shown in Fig. 5.

The nominal dimensions for the external spline are shown on Fig. 6. To insure that the involute tooth form



DP DIAMETRAL PITCH	CP CIRCULAR PITCH	A ADDENDUM	D DEDENDUM	WKD WORKING DEPTH	WD WHOLE DEPTH		R FILLET RADIUS
6/12	.5236	.0833	.1506	.1666	.2339		.070
8/16	.3927	.0625	.1127	.1250	.1752		.054
10/20	.3142	.0500	.0901	.1000	.1401		.044
12/24	.2618	.0417	.0749	.0834	.1166		.037
16/32	.1964	.0313	.0625	.0626	.0938		.022
20/40	.1571	.0250	.0500	.0500	.0750		.016
24/48	.1309	.0208	.0417	.0416	.0625		.015
32/64	.0982	.0156	.0313	.0312	.0469		.011
6/9	.5236	.1111	.1802	.2222	.2913		.027
8/12	.3927	.0833	.1342	.1666	.7175		.025
10/15	.3142	.0667	.1071	.1334	.1738		.023
12/18	.2618	.0556	.0890	.1112	.1446		.020
16/24	.1964	.0417	.0666	.0834	.1083		.016
20/30	.1571	.0333	.0528	.0666	.0861		.014
24/36	.1309	.0278	.0441	.0556	.0719		.011
32/48	.0982	.0208	.0332	.0416	.0540		.009

■ Fig. 6 – Nominal dimensions for the external spline



THREADS PER INCH	1/3 OF h	R MAX.	R MIN. PREFERRED	R MIN. ACCEPTABLE GROUND	R MIN. ACCEPTABLE ROLLED OR MILLED	R MIN. ACCEPTABLE DIE CUT	THREADS PER INCH	1/3 OF h	R MAX.	R MIN. PREFERRED	R MIN. ACCEPTABLE GROUND	R MIN. ACCEPTABLE ROLLED OR MILLED	R MIN. ACCEPTABLE DIE CUT
80	.0027	.00180	.00135		.001	.001	16	.0135	.00902	.00677	.003	.003	.001
72	.0030	.00200	.00150		"	"	14	.0155	.01031	.00773	"	"	"
64	.0034	.00226	.00169		"	"	13	.0167	.01110	.00833	"	.004	"
56	.0039	.00258	.00193		"	"	12	.0180	.01203	.00902	"	"	"
48	.0045	.00301	.00226		"	"	11	.0197	.01312	.00984	"	.005	"
44	.0049	.00328	.00246		"	"	10	.0217	.01443	.01083	"	"	"
40	.0054	.00361	.00271		"	"	9	.0241	.01604	.01203	"	.006	"
36	.0060	.00401	.00301	.003	.0015	"	8	.0271	.01804	.01353	"	.007	"
32	.0068	.00451	.00338	"	"	"	7	.0309	.02062	.01546	"	.008	"
28	.0077	.00515	.00387	"	.002	"	6	.0361	.02406	.01804	"	.009	"
24	.0090	.00601	.00451	"	"	"	5	.0433	.02887	.02165	"	.010	"
20	.0108	.00722	.00541	"	.0025	"	4½	.0481	.03207	.02406	"	.011	"
18	.0120	.00802	.00601	"	"	"	4	.0541	.03608	.02706	"	.012	"

■ Fig. 7 – Modified thread form for use on highly stressed parts

is carried to the proper depth, a diameter known as the true involute form diameter has been established. On the external spline this diameter specifies that "the involute form must be true outside this diameter" and for the internal spline that "the involute form must be true inside this diameter."

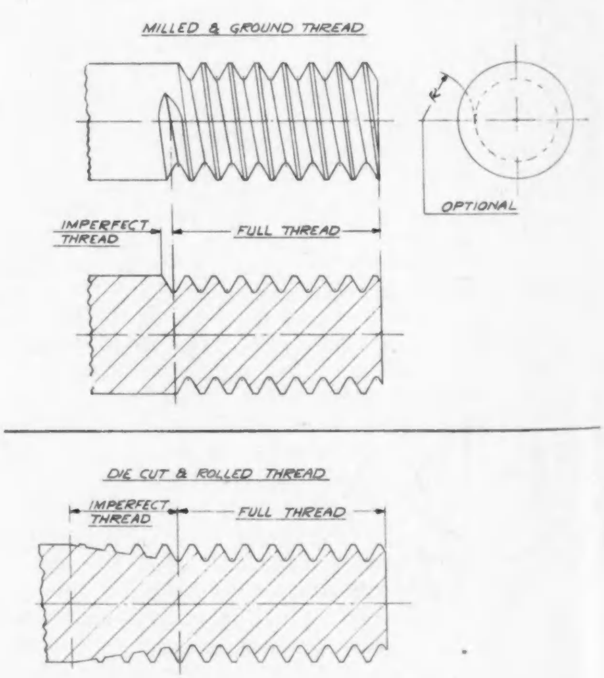
The following drives have been standardized and are using the 30-deg involute spline: generator drive, accessory-gear-box drive, fuel-pump drive, propeller-governor drive, vacuum-pump drive, and the No. 60 propeller shaft end. The 6/12 (2:1) series is recommended when a press-fit spline is used. The 6/12 is also used for a sliding-fit spline and the 6/9 (3:2) spline is used when additional area is required for spline sliding under load.

■ Modified Thread Developed

Parts are, in most cases, held together with threaded bolts, studs, or screws. The National screw form is satisfactory under fixed-load conditions but, under repeated load, has proved unsatisfactory. For this reason, a modified thread form should be used on all highly stressed parts. This thread form is shown on Fig. 7. Instead of allowing sharp corners as the national thread form permits, a maximum and minimum radius is used. The minimum radius is a radius formed between the side of the thread and the minimum minor diameter. The maximum radius is controlled by the sides forming a tangent at $P/4$ with a radius. The ideal thread would be the one with the radius at the root as shown in Column R "maximum" and R "minimum preferred," but available tools and equipment make it necessary to vary the minimum R according to the method used in producing the threads. This thread form has proved its usefulness for tapped holes in aluminum or magnesium as well as on studs or bolts. The National Screw Standard Class 2 and 3 does not allow for a neutral zone, so the low limit of the P.D. for the nut is the high limit of the screw. Nuts assembled with a size-to-size fit on the pitch diameter will seize. Many cases are on record where the stud had pulled out with the nut when disassembling an engine or if the nut is assembled on a bolt, the removal of the nut may prove costly, as the bolt may have to be cut off in order to remove the nut; also, many nuts must be assembled in places where speed wrenches cannot be used which makes it necessary to turn the nut by hand until it seats properly. For this reason, a size-to-size fit cannot be used and a neutral zone has been adopted. This neutral zone has proved to be useful as it will facilitate the assembling of nuts, bolts, and screws. The neutral zone is based on the diameter and not the pitch and is as follows:

0.0005 up to 1¼ in.
0.0010 – 1¼ in. up to 2 in.
0.0015 – 2 in. up to 3 in.
0.0020 – 3 in. and up

Special cases may require a neutral zone from 0.003 to 0.005, depending upon the design.



■ Fig. 8 – Milled and ground thread (above) and die-cut and rolled thread

Minimum dimension "B" for any of the following types of threads shall be as shown in TABLE 1 or TABLE 2.

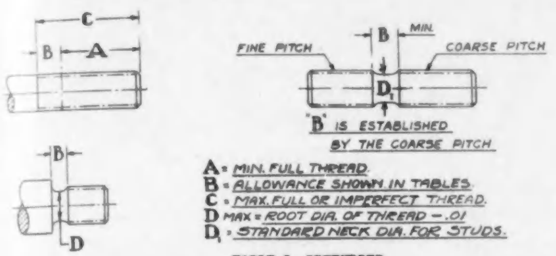


TABLE 1 PREFERRED

NO. OF THREADS PER INCH	B: DIE CUT THREAD (4 THDS.)	B: GROUND MILLED ROLLED THREAD (2 THDS.)	NO. OF THREADS PER INCH	B: DIE CUT THREAD (4 THDS.)	B: GROUND MILLED ROLLED THREAD (2 THDS.)	NO. OF THREADS PER INCH	B: DIE CUT THREAD (4 THDS.)	B: GROUND MILLED ROLLED THREAD (2 THDS.)
32	.14	.06	16	.26	.14	10	.40	.20
28	.14	.08	14	.30	.14	9	.44	.22
24	.18	.08	13	.32	.16	8	.50	.26
20	.20	.10	12	.34	.18			
18	.22	.12	11	.36	.18			

TABLE 2 MINIMUM

NO. OF THREADS PER INCH	B: DIE CUT THREAD (2 THDS.)	B: GROUND MILLED ROLLED THREAD (ONE THD.)	NO. OF THREADS PER INCH	B: DIE CUT THREAD (2 THDS.)	B: GROUND MILLED ROLLED THREAD (ONE THD.)	NO. OF THREADS PER INCH	B: DIE CUT THREAD (2 THDS.)	B: GROUND MILLED ROLLED THREAD (ONE THD.)
32	.06	.04	16	.14	.06	10	.20	.10
28	.08	.04	14	.14	.08	9	.22	.12
24	.08	.04	13	.16	.08	8	.26	.12
20	.10	.06	12	.18	.08			
18	.12	.06	11	.18	.10			

Minimum allowance B for any of the following types of holes shall be as shown in tables 3, 4 or 5.

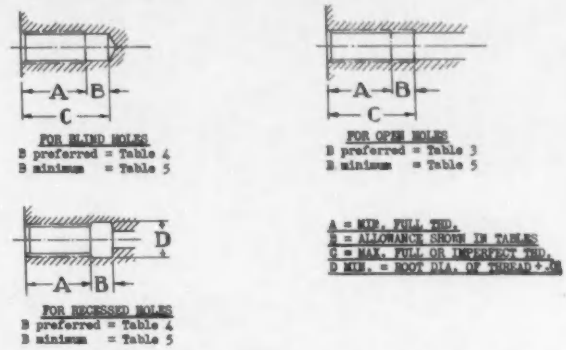


TABLE 3

THREADS PER INCH	B (6 Thds)
32	.20
28	.22
24	.26
20	.30
18	.34
16	.38
14	.44
13	.46
12	.50
11	.56
10	.60
9	.68
8	.76

TABLE 4

THREADS PER INCH	B (4 Thds)
32	.14
28	.14
24	.18
20	.20
18	.22
16	.26
14	.30
13	.32
12	.34
11	.36
10	.40
9	.44
8	.50

TABLE 5

THREADS PER INCH	B (2 Thds)
32	.06
28	.08
24	.08
20	.10
18	.12
16	.14
14	.14
13	.16
12	.18
11	.18
10	.20
9	.22
8	.26

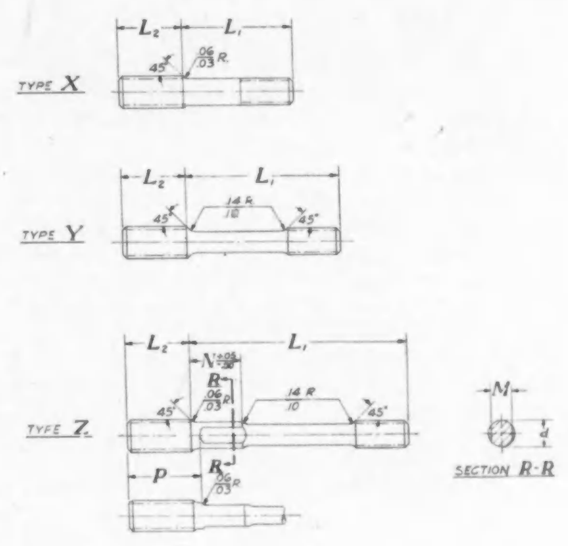
■ Fig. 9 – Tolerances for threaded parts

In the design of parts with threads, it is often very important to control the location of the end thread. Fig. 8 shows two types of threads. When threads are milled or ground, the end thread is as shown in the upper view and, when die-cut or rolled, threads are required as shown in the lower view. It is necessary to have a definite understanding of what is a full or imperfect thread and the picture as shown in Fig. 8 has proved helpful as it will save both time and cost where the end thread must be held to close tolerances in relation to a definite surface.

Fig. 9 shows the tolerances for threaded parts. A four lead thread is preferred on screws, bolts, and studs but may be decreased to one thread depending upon the type of thread and the design. On tapped holes six lead threads are preferred but may be decreased to four or two depending upon the design requirements. Long leads are preferred as they will cut the cost of producing the threads; also, they will produce smoother threads which are essential on steel parts when assembled in aluminum or magnesium parts.

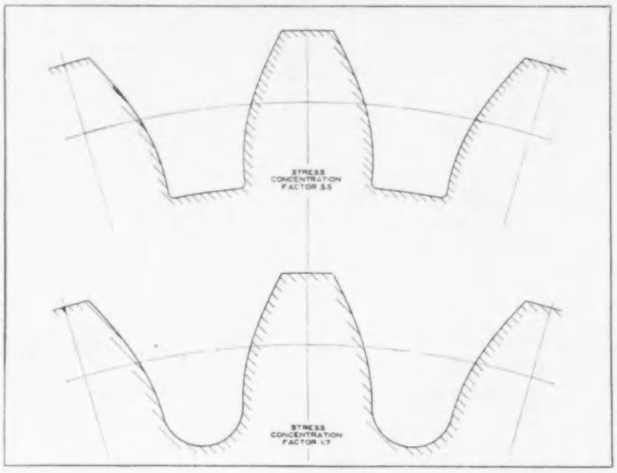
■ Stud Design

Studs are most commonly used for holding together parts in an aircraft engine as capscrews would not be satisfactory because the threads in the aluminum or magnesium would strip or wear due to the many assembling and disassembling operations that the engine has to withstand during its life. The stepped stud as shown in Fig. 10 has proved to be the most satisfactory type of stud. The short studs are of Type *X* but longer studs are necked to



NUT END	STUD END	L_2: 2 DIA	L_2: 1.5 DIA	d	M	N	P: 2 DIA	P: 1.5 DIA	L_1: TYPE X	L_1: TYPE Y	L_1: TYPE Z
.250-28	.313-18	.60	.45	.250	.20	.60	.65-.70	.50-.55	UP TO 2.00 INCL	DO NOT USE	DO NOT USE
.313-24	.375-16	.75	.55	.313	.25	.60	.80-.85	.60-.65	" " 2.50 "	" " "	2.55 AND UP
.375-24	.438-14	.90	.65	.375	.31	.60	.95-1.00	.70-.75	" " .75 "	.80 TO 2.00 INCL	2.05 " "
.438-20	.500-13	1.00	.75	.438	.36	.60	1.05-1.10	.80-.85	" " .85 "	.90 " 2.50 "	2.55 " "
.500-20	.563-12	1.10	.85	.500	.42	.75	1.15-1.20	.90-.95	" " .95 "	1.00 " 3.00 "	3.05 " "
.563-18	.625-11	1.25	.95	.563	.47	.75	1.30-1.35	1.00-1.05	" " 1.05 "	1.10 " 3.50 "	3.55 " "

■ Fig. 10 – Types of stepped stud



■ Fig. 11 – Stress-concentration factors for standard gear-tooth form and full-fillet gear-tooth form

eliminate fatigue failure, but long studs must have a flat at the stud end for assembling purposes (Type Z) to avoid overstressing the necked portion when assembling the studs. A coarse thread on the studded end and a fine thread on the nut end with the studded end 1/16 in. larger than the nut end has proved to be a satisfactory combination.

■ Reduction Gear Teeth

Fig. 11 – Many gears are used in an aircraft engine, and practically all of the large aircraft engines have reduction gears in addition to supercharger drive gears. The speed of the reduction gears is low compared with some of the supercharger gears which have gear ratios up to 14:1. Many types of gears have been tried but the 20-deg full-depth tooth form has proved to be the most satisfactory. The standard 20-deg tooth form has high stress concentration at the root as the fillet at the root is only a few thousandths of an inch. By using a 20-deg tooth form with a full radius at the root, the stress concentration will be decreased considerably. The stress-concentration factor for the full fillet tooth form is approximately one-half that of the standard form. Greater flexibility is also obtained with the full fillet tooth form thereby distributing the loading over a greater number of teeth. Maximum factor of safety is obtained by controlling the involute form tooth spacing and thickness.

■ Conclusion

In this paper the author has endeavored to show many items that need to be standardized, as very little information has been published on some of these subjects, also, that some definite steps should be taken to eliminate the fractional system and a purely decimal system used throughout the industry. The twenty odd gaging systems used today should be eliminated and only one system used, based on a decimal system. The number and letter drill sizes should be replaced by sizes based on a decimal system. It has been estimated by a firm using copper, brass, and aluminum, that the saving for this firm alone would be over $1,000,000 a year if a uniform gage system was used.

Serious consideration should be given to the adoption of the metric system for, if the metric system were adopted today, due to its simplicity, everyone using this system ten years from now would say: "Why didn't we start to use the metric system before?" I believe that, regardless of the outcome of the present war, this country will be the only country left that will use the inch system and the competition will be keener than ever. Further, more objections will be raised to the import of our products to countries using the metric system as England and the British Empire will be forced to adopt the metric system in order to compete with countries using the metric system on the world's market. Our exports to South America and Mexico would be facilitated as these countries are all using the metric system.

The Army and the Navy, in creating and publishing the AN Standards, have done the industry a most valuable service, but the industry should produce the new standards and submit them to the Army and the Navy and not rely upon the Army and the Navy to produce the standards. Most of the AN Standards are airplane standards and some are not satisfactory for engine use, so it is recommended that new standards be created that are used on engines only but, where parts such as nipples, hose and hose clamps, and so on, are used on both engines and airplanes, only one standard should be used. Many standards should be revised and brought up-to-date, for example, the nominal size of the I.D. of the hose is the nominal size of the O.D. of the nipple. This combination is causing trouble as the I.D. of the hose must be smaller than the O.D. of the nipple in order to insure a tight connection. Another standard which needs investigation concerns the many different types of nuts, bolts, and screws used today. A typical sample was brought to the author's attention some time ago where a bolt which costs 14¢ when made to a manufacturer's drawing could be purchased for 3.2¢ if an AN part could be used. If an AN part could have been used in this case, the saving per 10,000 parts amounts to $1080.00. The money spent on standardization will pay dividends beyond imagination, so it is recommended that the management of engine and airplane companies encourage their men to serve on SAE committees because, without the management's support, the work cannot be done satisfactorily.

Error in Table of "1940 Road Detonation Tests"

In Table 3 of the paper: "1940 Road Detonation Tests," published on p. 195, Transactions Section, of the May, 1941, SAE JOURNAL, a typographical error appeared in the fourth column: "Research Minus ASTM, fourth figure from the bottom, referring to Fuel 20-B. The figure 3.3 should have read 13.3, as is evident by subtraction of the value in the second column from that in the third.

BRAKES....

Their Analysis and Balancing

by S. JOHNSON, JR.

General Engineer,
Bendix-Westinghouse Automotive Air Brake Co.

IN its discussion of the analysis and balancing of air-brake systems, this paper places particular stress on brake rigging, or the foundation brake, and its maintenance.

Most of the chronic braking troubles could be eliminated, Mr. Johnson contends, if all brakes were balanced and then subjected to a well-planned and executed periodic maintenance program. In balancing, he explains, the brakes are modified so that all the brake shoes on the vehicle are contacting the drum at the same low pressure at the same time during the brake application. Benefits of such balancing and maintenance reported by the author include increased brake lining life; and reduction of: brake-drum breakage and checking, drum scoring, grease on brake linings, brake adjustments, and bearing failures and tire blowouts caused by heat.

The remainder of the paper is devoted to details of the brake analysis and maintenance program. Various points are emphasized by charts compiled from data obtained by testing vehicles in actual service. Although the tests reported were selected at random, Mr. Johnson avers that the data hold true for operations in any part of the country.

THE conditions under which motor vehicles are operated at the present time make it imperative that certainty and ease of control be developed to the maximum degree possible.

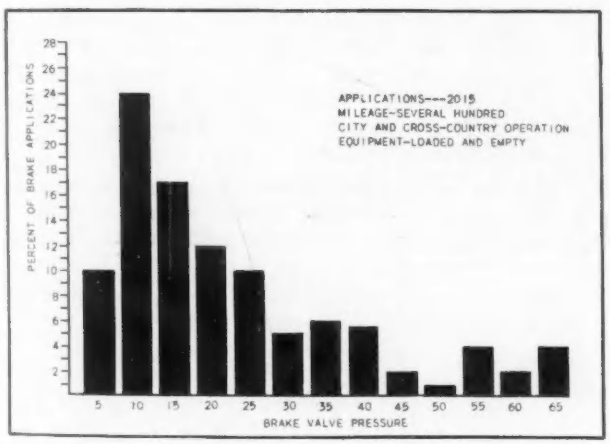
The functions of control are divided as to: (a) the application of power for acceleration and sustaining speed against the forces acting against the vehicle; (b) a braking system to reduce the speed, to stop the vehicle, and to hold it at rest; (c) a steering system to enable the operator safely to pilot the vehicle along the road.

It seems peculiar that, of these three functions between which it is hard to distinguish as to importance from the standpoint of safety, the one which has received most consideration from regulatory bodies is the function of braking.

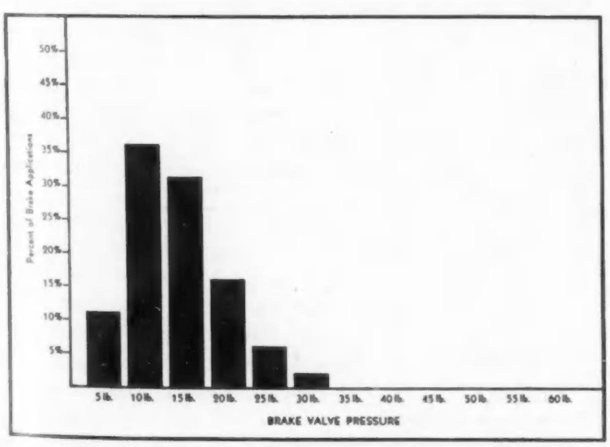
Furthermore, it seems equally peculiar that all regulations are pointed toward requiring the maximum obtainable braking capacity. This condition is in spite of the fact, which is well established, that extremely effective braking will cause serious interference with the function of steering and thereby will of itself contribute against safety on the road.

There is no function of the motor vehicle which operates under such widely and constantly varying conditions as does the brake system.

Since the design and maintenance of the air brake is so familiar to many, I am going to discuss in this paper the



▪ Fig. 1 – Brake application chart – 2015 applications in city and cross-country operation

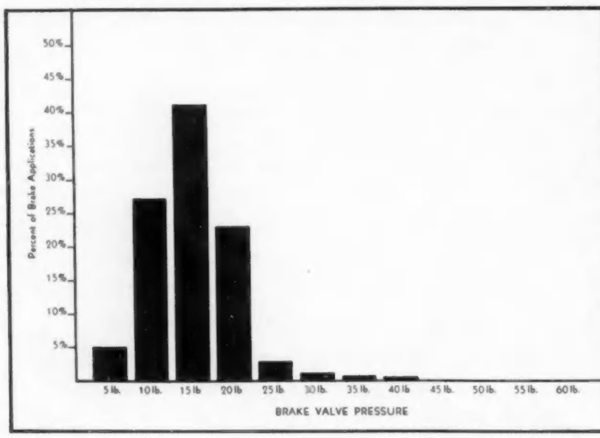


▪ Fig. 2 – Brake application chart – city operation

[This paper was presented at a meeting of the New England Section of the Society, Boston, Mass., Oct. 8, 1940.]

subject of "brake rigging," or the foundation brake as many call it, with particular reference to maintenance.

The subject of brake rigging and braking power is a comparatively little understood phase of the science of braking. We do not pretend fully to comprehend all of its essentials or details but, after having studied the subject for the past few years, we are able to reduce the underlying principles – both as to retarding force desirable for automotive vehicles and what must be embodied in a foundation brake-gear design – to a scientific basis. We used to think that practically all of the effectiveness of the air brake was in the performance of the air-actuated devices, and that only a little of the effectiveness or efficiency of the air brake came from the foundation brake gear.



■ **Fig. 3 – Brake application chart – hilly operation**

However, we discovered that this ratio might well be reversed; for, without an efficient and properly designed and maintained foundation brake gear, the most highly developed and most efficient power-brake mechanism is of little value. The brake which can transform the fluid pressure of compressed air into a very great mechanical force needs a foundation brake rigging sufficiently rugged for the development of this force; and, if it is not provided, every undesirable condition in braking that can be conceived may occur – resulting in inconvenience, increased maintenance, and in retardation of the development of one of the principal things that has helped make the present scope of truck and bus transportation possible, namely, the power brake.

For several years, many progressive operators have known that the most economical operation is obtained from vehicles that are operating constantly at peak efficiency, which could be obtained only by periodic inspection and maintenance. Consequently, they have established well-planned maintenance schedules on a mileage basis for their motors, transmission, electrical equipment, and steering. However, it is surprising to note that only a few of these operators have given similar attention to brakes. As a matter of fact, some have nothing more than a hit-or-miss brake maintenance program. This condition remains true despite the fact that modern traffic – which demands that trucks, buses, and so on, maintain the highest safety and operating standards established for pleasure-car operation – is throwing an increasingly heavy duty on braking equipment.

Naturally, as manufacturers of power-brake equipment, we are vitally interested in seeing that vehicles using our equipment have superior brake control, longer lining life and drum life, and the maximum safety at the lowest possible operating cost. We know that, while the air-brake system has sufficient quality and power to guarantee fine air-brake performance under all circumstances, the actual stopping distance, lining and drum life will depend to a great extent on the type and condition of the fundamental brakes.

About two years ago, we became convinced that most of the chronic braking troubles could be eliminated if the brakes were *balanced* and then subjected to a well planned and executed periodic maintenance program.

To prove this, we contacted an operator whose fleet was operating between Cincinnati, Pittsburgh, New York, and Boston, and who was complaining constantly of poor brake and drum lining life, unsatisfactory stopping ability, trailers bumping tractors, and jack-knifing. We arranged to have this operator send the vehicles that were giving the most trouble, to our service station. As each unit came in, we analyzed it carefully, and balanced the brakes (that is, we modified the brakes so that all the brake shoes on the vehicle were contacting the drum at the same low pressure at the same time during the brake application). We also made sure that the retarding force developed by each wheel closely approached the designed retarding force. (This is very essential to obtain proper lining and drum life based on our experiences to date.) We then arranged to have the vehicle returned at regular intervals (based on mileage basis) for brake inspection and maintenance. The results were amazing. Brake complaints ceased, lining life doubled and, in some cases, tripled, and brake performance improved 50 to 75%.

We were not satisfied, however, so in addition, we analyzed thoroughly a number of truck or bus operations in various parts of the country. The data obtained from these operations showed conclusively that a comprehensive brake analysis and maintenance program gave the operator both finer brake performance and lower maintenance cost.

■ Results of Program

We helped one well-known Southeastern company establish a brake analysis and maintenance program. This company is engaged in city service and operates over 100 vehicles. At first they were skeptical – saying that they were doing most of the things we had recommended and still had brake troubles. However, they agreed to try our program, and now everyone in the organization is enthusiastically boasting of the finer brake performance and lowered brake maintenance cost. This company's records reveal the following facts:

1. Brake lining life has increased from 8,000 to 30,000 miles, and is still increasing.

2. Brake-drum breakage and checking which was high before starting this program has been dropped to almost nothing.

3. Formerly they operated a drum lathe almost continuously. Now drum scoring has been practically eliminated and drums are only turned as normal wear occurs.

4. They have reduced grease on brake lining to one or two wheels per month.

5. It is no longer necessary to remove the wheels between their 10,000-mile inspection period.

6. Formerly they had to adjust the brakes on every vehicle practically every night. Now adjustments are made between 1500 and 2000 miles.

7. They have reduced their bearing failures due to excessive heat to practically nothing.

8. Tire blowouts caused by heat have been practically eliminated.

9. They have increased braking efficiency on all units. On some of the older units they report an efficient brake for the first time.

■ Details of Brake Program

The remainder of this paper will be devoted to the detail of the brake analysis and maintenance program. Wherever convenient, we have emphasized our various points by showing charts compiled from data obtained by testing vehicles in actual service. These tests are representative reports selected at random. Some of the data were secured from operations in the Pittsburgh area. However, subsequent tests indicate that the data are true for other operations in any other part of the country.

We know that most of the points considered in our maintenance program are not new discoveries. As a matter of fact, brake experts have known most of them for years. However, a program in which all parts are considered in relationship to each other has been overlooked completely by the majority of us. In our program we have attempted to achieve better brake performance, lower maintenance cost, and better lining life through a correct interpretation and application of these facts.

From the very start, test results indicated the incorrectness of two long-standing air-brake ideas, namely:

1. Due to the inherent action which permitted the building up of equal pressure in all brake chambers, air brakes automatically equalized brake performance.

2. That all insufficient brake troubles could be remedied by increasing the brake valve pressure.

Cowdrey or drum temperature tests showed very clearly that, while air pressure was equal in all brake chambers, the actual brake application produced by the various wheels would vary considerably. Obviously, this condition indicated an inequality in the resistant and friction factors (springs, camshafts, cams, hinge pins, and so on) both in the air-brake system and in the fundamental brake rigging. Other tests showed us that, though the full maximum power of the air-brake system was necessary for emergency stops, pressures in excess of 40 psi were rarely used in brake operations.

Addition of the first four columns in Fig. 1 will show that over 60% of the brake applications require 20 psi or less of brake chamber air pressure. The true significance of this can best be realized by comparing these low total brake-chamber pressures with the brake chamber pressures actually needed to overcome the various frictions, resistance, and so on, and to place the shoes against the drum ready for actual braking. It is immediately apparent that, while a 4 or 5 psi differential in the air pressure needed to expand the shoes is relatively unimportant when the total brake-chamber pressure is 60 psi, it becomes tremendously important when the average application pressure is 10 to 15 psi. For example, on a 15 psi total application, should one wheel be operating at 10 psi while the others require only 5 psi shoe expansion force, this wheel will be using only 5 psi pressure for actual braking purposes. Obviously, it will be delivering only 50% of its designed retarding force. Should the reverse of this condition be true, that is, the brake on one wheel be operating at 5 psi pressure and the brakes on the other wheels be operating at a 10 psi pressure; it is immediately apparent that the brake on this one wheel would be vastly overworked. This, of course, will result in increased drum temperatures, increased wear, poor brake performance, and very short lining life.

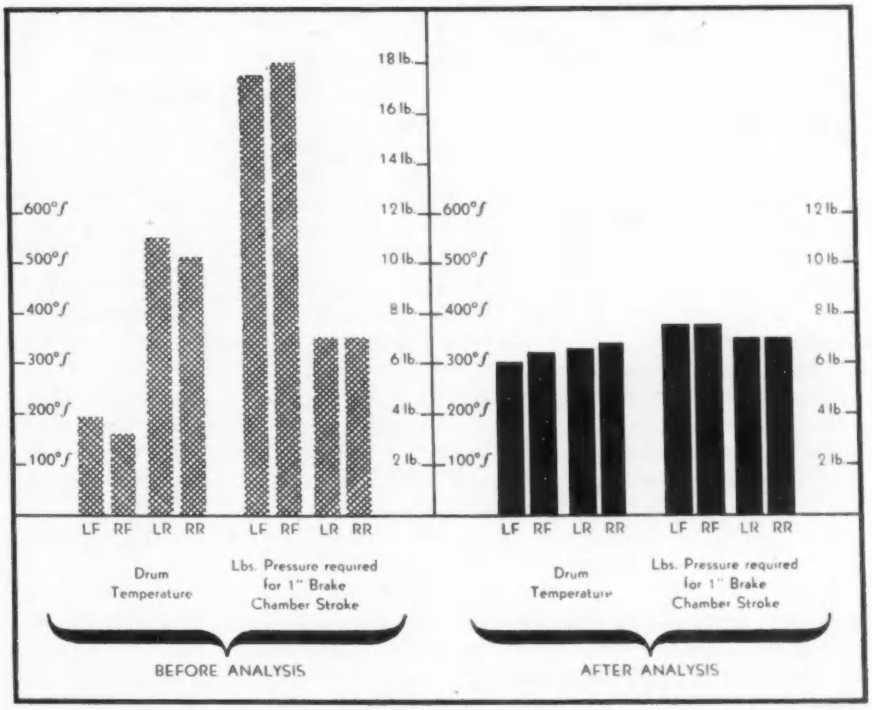
Fig. 2 is another illustration of brake valve air pressures employed in actual city operation.

Fig. 3 is still another illustration of brake valve air pressures. This operation is one where the topography is quite hilly.

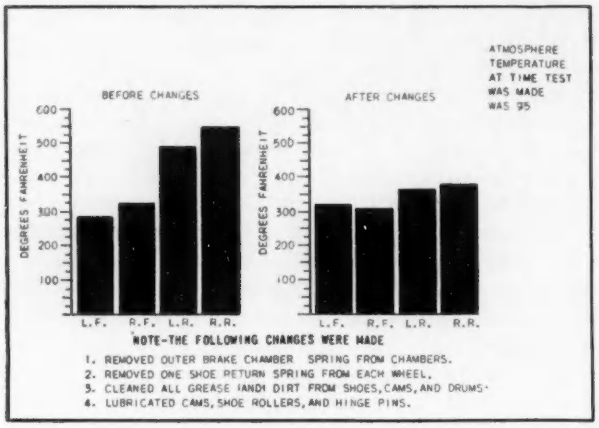
Fig. 4 illustrates a condition where "before analysis" front drum temperatures are running considerably lower than those of the rear. On the right side of the chart, note that the pressure required to expand the shoes against the drum is higher in the front than in the case of the rear brakes.

The "after analysis" portion of this chart shows that both temperatures and pressure required to bring the brake shoes in contact with the drums are equalized.

The result of analysis on this particular unit increased



■ Fig. 4 – Drum temperatures and pressures required for 1-in. brake chamber stroke before and after analysis

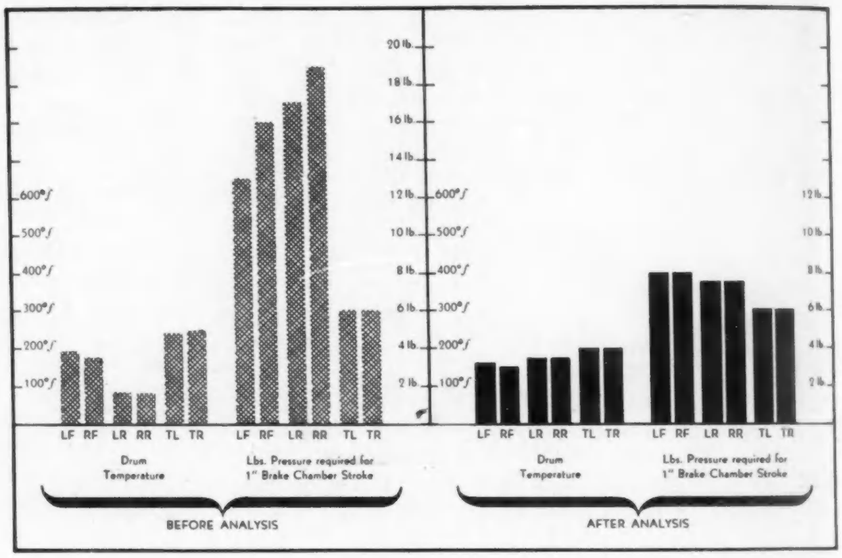


■ **Fig. 5 – Brake-drum temperatures before and after changes**

condition has been discovered. Sighting one instance, approximately 15 to 20 psi were required to engage the shoes against the drums of the trailer in direct relation to approximately 6 to 8 psi on the tractor. Naturally, lining life on the tractor was in for a sharp decline and a tendency toward jack-knifing was noticeable under adverse climatic conditions.

Fig. 7 – Note the difference in drum temperatures between front and rear and the variations in pressure previous to analysis. The benefits of analysis are no better indicated than the result shown here – lower drum temperatures caused by equalization of pressures required to bring the shoes in contact with the drums.

Fig. 8 – At a glance, this chart shows a condition common to vehicles previous to brake analysis. Note that



■ **Fig. 6 – Drum temperatures and pressures required for 1-in. brake chamber stroke for tractor-and-trailer unit before and after analysis**

rear lining life 100% without decreasing that of the front linings.

Fig. 5 shows a condition "before analysis" that is more general than the previous charts.

Fig. 6 is only shown as a matter of interest. By this chart it is notable that conditions similar to those of single vehicles exist in the case of tractor and trailer units. Evidenced here are the easily recognizable and inevitable combination of uneven drum temperatures and pressures required to produce a 1-in. brake chamber stroke, previous to analysis.

Obvious in this specific instance "before analysis," the trailer brakes were applying between 10 and 12 psi in advance of tractor brakes. Inasmuch as it has been conclusively determined and indicated in an accompanying chart, that a preponderant average of brake applications are made in the lower pressure brackets, it is at once apparent why the operator of this particular unit had been experiencing all-too-short a lining life on his trailer brakes.

The after-analysis portion of this chart shows an accomplishment which has done much to equalize both pressures and temperatures between tractor and trailer. In this one case, it is gratifying to note that, after analysis, a 200% increase in the lining life was effected on the trailer while that of the tractor remained unchanged.

It is worthy of note here that, in many cases, a reverse drum temperatures register excessively high on the rear brakes. This is caused by the fact that the rear shoes are applying with a pressure of 6 psi while the front brakes require 10 and 11 psi to expand the shoes against the drums. Inasmuch as the average application pressure is approximately 15 psi, it is readily seen why this heating condition exists. At this rate it is obvious that only 3 or 4 psi would be available for braking effort on the front wheels whereas approximately 9 psi of working air pressure would be lavished on the rear.

The after portion of this chart shows the results of an identical test following brake analysis. Note in this instance that drum temperatures were substantially reduced and evenly distributed. Thus is shown that, since all brakes have been caused to apply at approximately uniform pressure, an even balance of air pressure is available for all wheels.

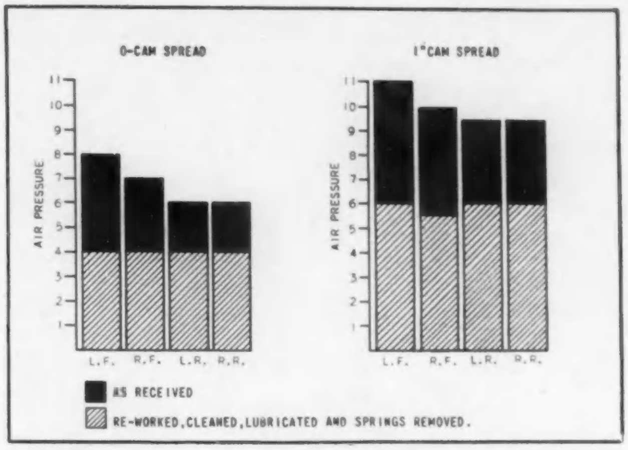
It is interesting to note that the net result of this specific case of analysis was an increase in lining life from approximately 8,000 to 10,000 miles on the rear brakes to approximately 30,000 to 40,000 miles without decreasing the normal life of front linings. The reason for this increase is equalized and proportionately lowered drum temperatures. High peaks which were causing burning and disintegration of the linings were eliminated.

Brake drum temperatures, due to the fact that heat is

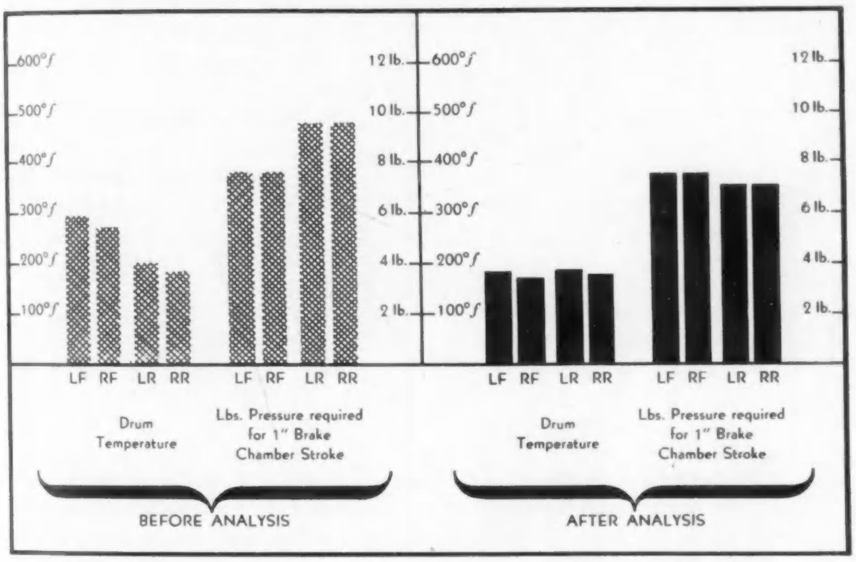
developed in proportion to the braking energy expended, are a true indication of the portion of the work being done by each wheel. It is, of course, needless to emphasize the relationship of heat to brake maintenance cost. All are aware of the vast investments that truck and bus companies, brake drum companies, brake lining companies, and so on, are constantly spending in a search for braking materials that have longer service life and higher heat-resistant qualities. Then, too, those who are operators have actually seen numerous cases where extreme heat has cracked or checked the drums causing wheel bearing failures and literally burning out the brake lining.

Fig. 9 – Without doubt, one of the most important factors to consider in brake balancing is the relationship of the brake chamber and brake shoe springs to the brake chamber pressure required to expand the brake shoes against the drum. Our experience shows that in many installations the load strength of these springs is excessive. Obviously, this condition has been caused by increasing the spring strength to allow the springs to function properly even though neglected maintenance had drastically increased the amount of friction they had to overcome. We feel that the best braking results will be obtained when the strength of these springs is not in excess of the amount necessary to insure performance of their function when the friction of the various brake parts is held to a minimum. The brake chamber return springs should have only sufficient strength to return the cam, camshaft, slack adjuster, and so on, to release position when all parts are in good operating condition. There is a general rule for the correct spring load, but in many cases we have produced results by removing one brake shoe spring and (when two were used) one brake chamber spring.

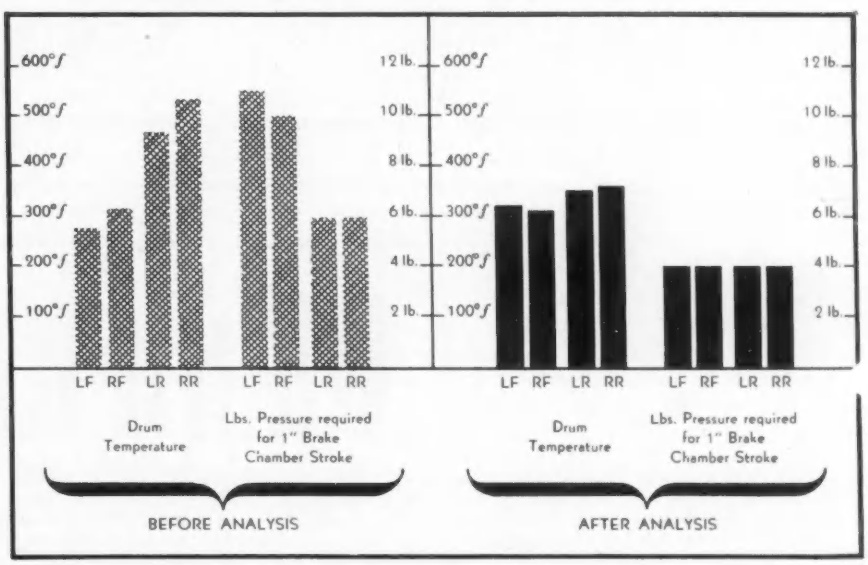
Fig. 10 – This chart shows pressure required to obtain 1 in. brake chamber stroke before and after analysis at zero cam position, or new lining condition, and 2 in. shoe spread, which represents a point where the lining is practically worn out. A ¾-in. block is used.



■ **Fig. 9 – Air pressures to cause 0 and 1-in. cam spread as received and as re-worked, cleaned, lubricated and springs removed**



■ **Fig. 7 – Before-and-after analysis drum temperatures and pressures showing lower drum temperatures caused by equalization of pressures after analysis**



■ **Fig. 8 – Drum temperatures, which before analysis were excessively high on rear brakes, are substantially reduced and evenly distributed after analysis**

Note that before analysis, with cam at zero position, a 4 psi variation at which the brakes applied and, at the 2-in. shoe spread, the pressure required to obtain 1 in. brake chamber stroke varied from 15 to 23 psi. On this particular vehicle, should the brakes on the rear be relined and not the fronts, note that the rears would apply with approximately 8 psi, whereas the fronts would not contact the drums until 23 psi of air was applied. This is undoubtedly the reason why, in so many instances, shorter life is experienced in the case of the rear wheels on the second set of brake lining than on the original. Note especially, after analysis, at the zero cam position, all brakes applied with 5 psi pressure and at 2 in. or with shoes spread to the point where lining is ready for replacement, the pressure increased to 6½ psi. In this condition should the rear brakes be relined and not the front, the rear brakes would apply at approximately 5 psi and the fronts with 6 psi, making a 1 psi differential where, before analysis, under the same condition there was a possibility of creating a differential of 15 psi. This same condition could result should the front brakes be relined and not the rears, which is one explanation why operators may experience locking wheels or grabbing front brakes under adverse weather conditions.

Fig. 11 – From five different vehicles selected at random from a fleet which was brake analyzed at the factory, this chart shows the relatively even, low temperatures at about 7000 miles of service. It is evident here that a unit properly analyzed will remain in proper condition from one inspection period to another when wheels are normally removed for bearing check – providing, of course, that these periods are within the mileage span of good maintenance practice.

Summarizing briefly, the steps to take to accomplish brake balancing are:

1. Clean brake shoes and brake rigging thoroughly and keep them cleaned. (See Figs. 12 and 13.)

2. Lubricate cams and cam followers at regular inspection periods with a good layer of heat-resistant grease.

3. Lubricate hinge pins at regular inspection periods. Avoid over-lubrication.

4. Blow dust out of brake drums at frequent intervals.

5. Use proper brake shoe return springs.

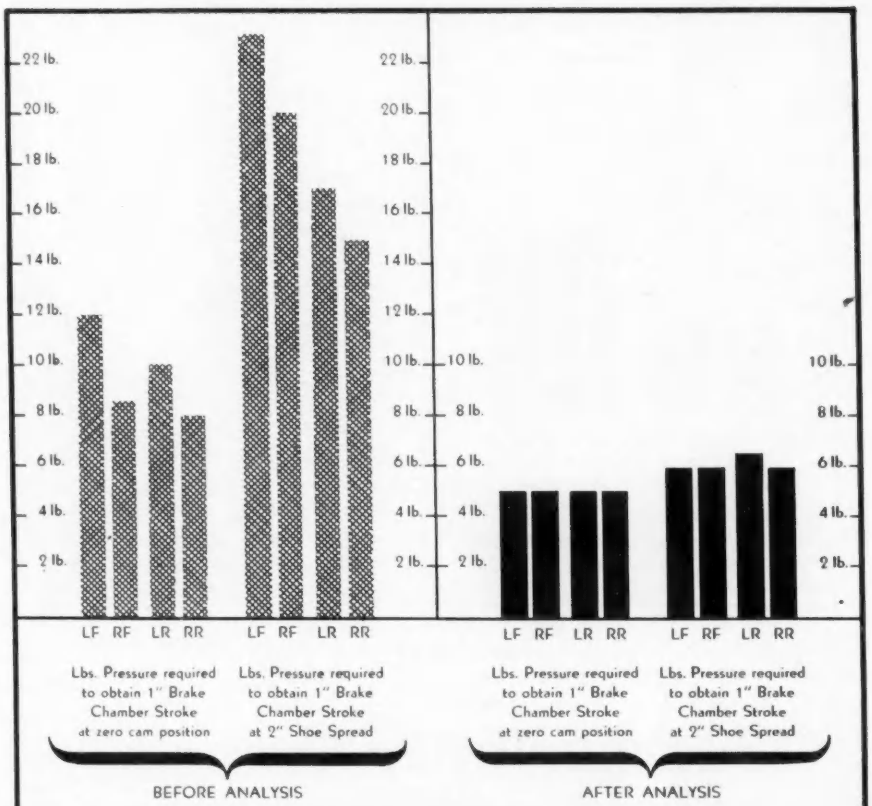
6. Use the same lining on all four wheels.

7. Replace lining on all four wheels at the same time. If this is not always possible, at least replace all blocks on one axle at the same time.

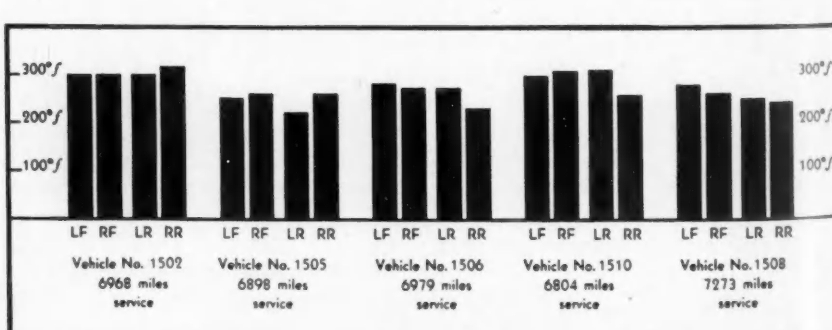
8. Turn drums frequently enough so that lining life and brake performance do not suffer due to poor drum condition.

9. When one wheel gives repeated trouble (such as high temperature, grabbing, howling, and so on) do not keep on pulling that wheel to find the trouble – look at the other three. The trouble is probably due to the complaining wheel doing all the work. Correct conditions on the other three wheels so they do their proper share of the work.

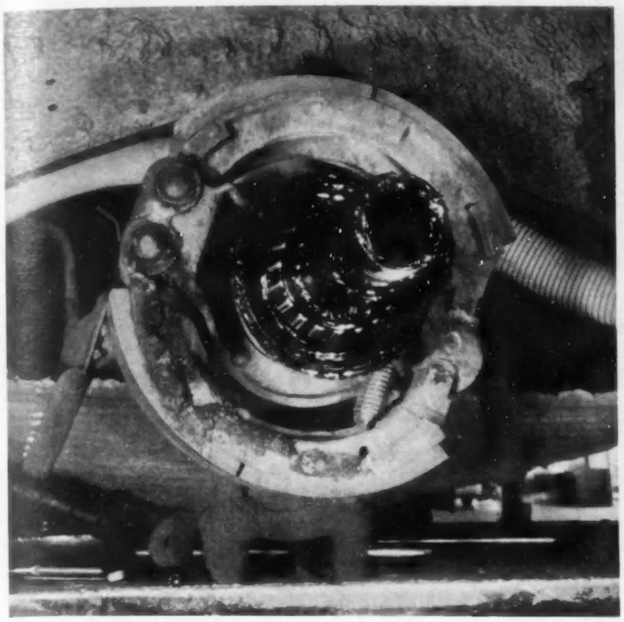
The foregoing discussion has covered the brake-balancing pro-



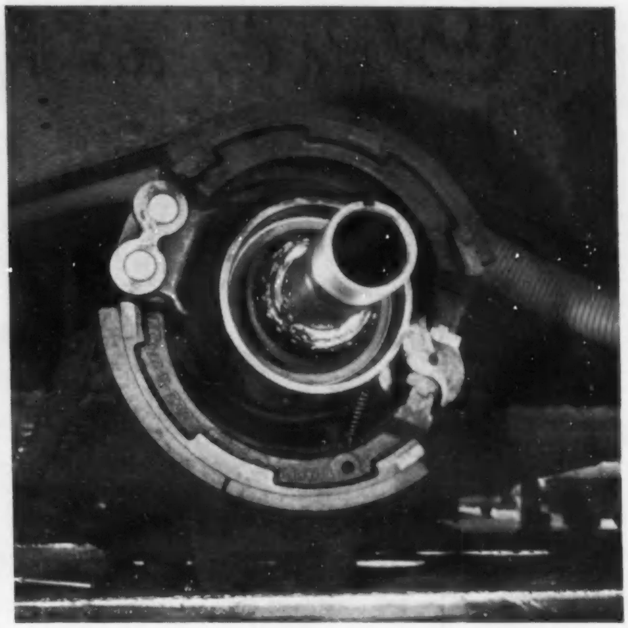
■ Fig. 10 – Pressures required to obtain 1-in. brake chamber stroke at zero cam position and at 2 in. shoe spread – Before and after analysis



■ Fig. 11 – Drum temperatures from five vehicles after about 7000 miles service following brake analysis



▪ Fig. 12 – Brake-shoe mechanism full of dirt



▪ Fig. 13 – Brake-shoe mechanism of Fig. 12 after cleaning

gram in its general aspects but we feel that a résumé of the technical details will also be of interest.

▪ Summary

We define brake balancing as the choice of the proper factors to improve safety in operation, increase passenger comfort, reduce damage to lading and passengers, and lower overall maintenance costs.

The factors just referred to fall under two general classifications; mechanical considerations and pneumatic details.

A. The primary mechanical consideration is the ratio of retarding force to weight.

The proper selection of this ratio is vital to good brake performance for it and little else determines the rate of deceleration. The actual retarding force developed by the brakes may be modified by the engine or motors, friction, rolling resistance, and wind, but it is the only force subject to design control for retardation purposes.

For the present-day automotive braking, experience dictates the choice of 6/10 ratio between the nominal brake retarding force based on 60 psi brake chamber pressure and the static weight between the wheel and the road under the gross load condition. This experience covers variations in adhesion, weight transfer, lining coefficient, brake rigging friction, return spring values, spring suspension, unsprung weight, and so on, and any attempt to segregate any one of these factors and apply compensation for its normal variations without analyzing all of the variables, is incorrect in principle.

B. A second factor to be considered is weight transfer. Braking forces tend to up-end a vehicle and develop a moment of mass times acceleration times height of center of gravity from road, which acts to reduce rear-wheel weight and increase front weight during deceleration.

On any one vehicle the percentage of this front increase will vary with the load, its distribution, and directly as the rate of deceleration; and, in addition to these variables, for different vehicles, with the wheelbase, gross weight, and location of center of gravity.

However, some manufacturers base their braking ratios on a 10% transfer of weight from rear to front during deceleration periods. We have approved braking on this basis provided the manufacturer accepted responsibility for steering control and front-end stability. In some cases, we have found that increasing the brakes on the front wheels to compensate for a 10% weight transfer increases the tendency of the front wheels to slide as much as 30%. As a matter of fact, it has been found advisable in many cases to add a limiting valve on vehicles which have higher braking ratios on the front axle to compensate for the 10% weight transfer. This permits reducing the air pressure delivered to the front brakes when road conditions lower the coefficient of adhesion between the tires and the road.

C. The length of the slack adjuster which is used with the operating brake chamber must be considered if a reasonable period between brake adjustments is to be obtained. The available stroke in the brake chamber and the brake cam radius determines the length of slack adjuster which should be used. Engineering tables, long established, are available to manufacturers and operators covering these relations.

D. The shoe return springs must be designed to just keep the shoe in contact with the cam at all times. Data to date indicate that the minimum acceptable values for this spring force for normal extension of the spring is equivalent to two to three times the weight of both shoes with a maximum pickup of 50 psi.

The pneumatic details which must be considered are the maximum pressures required to set the brake shoes against the drums and the uniformity of these pressures.

E. Brake-lining wear increases with temperature and uniform wear on all wheels can be expected only when operating temperatures are uniform. It is generally accepted that drum temperatures are an indication of lining

temperatures. Due to structural differences of various wheels, it is a practical impossibility to make the rate of heat dissipation, by radiation, convection and conduction, proportional to the work done at each drum, so a temperature balance may be obtained for practical purposes by checking drum temperatures under road service conditions and adjusting the K factor or unit lining pressure. To this end we have agreed to K factors as high as 0.7 for the front drums (when 0.6 is used on the rear) providing the customer accepts some measure of responsibility for steering control and front-end stability and arranges the foundation brake so that a return to 0.6 may be readily accomplished.

$$K = \frac{\text{Retarding Force at the Ground}}{\text{Weight on the Wheel}}$$

Testing Military-Type Motor Vehicles

THE proving grounds of the Army, one at Camp Holabird and one at Aberdeen, are interesting examples of thoroughness and practicability. The range of testing carried out and the recorded information are of special value to the automotive industry and have formed the basis for many innovations in design that are passed on automatically to the users of commercial vehicles. Their list of tests are a glossary of important items in the life of any motorized vehicle. A test of a new design type will fill a book of several hundred pages and will cover items such as:

- Cooling Test – 90 F Differential
- Drawbar Pull Evaluation
- Speed Tests
- Climbing Ability
- Acceleration Tests
- Cold Starting Test
- Steering Test
- Water Test – 32 In. Water
  - Mud Test
  - Sand Test
- Continuous Operating Test
  - Braking Test
  - Lubrication Test
  - Riding Qualities
  - Fuel Consumption
  - Distortion Test
  - Tire and Track Tests

Summarizing this impressive list, it means that the Army expects trucks to operate without overheating in at least 120 F atmospheric temperature. The carburetion, oil level in engine, the oil-bath air cleaner, transmission and rear-axle lubricant, and all other functional accessories must operate on grades as steep as 65%. Submerged in water almost to the top of tires, the vehicles must be sealed and ignition and lighting systems must be capable of functioning at reasonable speeds. The continuous operating tests at the speed of 2½ mph take into account keeping pace where necessary with marching troops. The Army is naturally fuel-consumption conscious, realizing as it does that even a small saving in the number of operating vehicles represents an accumulative total of large magnitude. An example of this point could be stated as follows:

Based on a saving of 1 mpg, 10,000 vehicles operating at maximum power for a 10-hr period could save as much as 245,000 gal. It is easy to estimate mentally the number of extra vehicles which would have to be employed to supply this additional fuel every 10 hr.

The distortion or block test represents conditions unusual but very informative for the automotive industry. Starting with 4-in. high blocks progressively increasing to 12 in. and staggered so opposite wheels are alternately raised, this test simulates cross-country conditions and demonstrates a flexibility in construction eminently desirable in hard operating conditions in some commercial operations.

The development of special tracks for high-speed vehicles is one that again will one day be applied to commercial vehicles. There are many operations where a flotation pressure of 4 psi would make the use of tracks reasonable and economic.

Quite apart from the pressing need for continuous progress in commercial automotive design, the Army specifications have further challenged the industry, and today engines with outputs way beyond what may have been considered possible two years ago are here. Horsepower output per cubic inch has increased 17%, a torque of 0.84 ft-lb per cu in. and a weight-horsepower ratio of 6.8 lb per hp have been achieved in a large measure by the encouragement of the U. S. Army. Just a couple of years ago 125 psi bmep seemed a goal a long way off. Today a bmep of 127 psi in commercial and Army engines is possible and practical. Engines must be built to run without governors and stand up under any unusual field service conditions.

The Navy requirements have also to be considered but the proportion of land vehicles is naturally quite small compared to the Army and resolves itself in a small range already covered in the main by similar Army specifications.

Furthermore, economy tests show a saving in fuel consumption represented by a decrease of 0.1 lb per hp-hr and may result easily in 1 mpg less fuel used in some actual operating conditions.

Excerpts from the paper: "Military-Type Motor Vehicles, Their Design and Uses," by Robert Cass, chief engineer, The White Motor Co., presented at a meeting of the Chicago Section of the Society, Chicago, Ill., April 1, 1941.

LUBRICATION of Severe-Duty Engines (DIESELS)

by J. G. McNAB, W. C. WINNING, B. G. BALDWIN, and F. L. MILLER
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THIS paper points out differences in diesel and gasoline engine design and operation which result in significant differences in heat flow and distribution in the engine, which may, in some cases, cause emphasis to be put on such problems as ring-sticking, wear, and piston scuffing, and in others, on sludge formation, varnish, coking, and so on.

Naphthenic and paraffinic oils of 40 and 100 viscosity index, respectively, to which have been added small amounts of a detergent-disperser-inhibitor type additive developed by the Esso Laboratories, have been subjected to extensive laboratory and field tests and have been found to meet the most severe requirements of heavy-duty diesel service in modern transportation and industry. Results of some of these tests were compared with results of similar tests on typical naphthenic and paraffinic mineral oils commonly used for diesel lubrication under moderate conditions of speed, load, and temperature where extreme detergency and oxidation stability are not required, on another detergent-disperser-inhibitor blend in a naphthenic base oil and on a commercial inhibited oil of the paraffinic type used in diesels requiring extreme oxidation stability but ony a little detergency.

The research forming the basis of this paper was carried out principally on three of the better-known makes of diesel engines, selected because they represented three rather distinct types of engine design and because each manifested certain of the characteristic lubrication problems which, considered together, included practically all of those peculiar to the diesel engine.

■ ■ ■

DIESEL engines, like gasoline engines, differ considerably among themselves in the burdens they impose upon the lubricating oil and, consequently, in the lubrication problems they raise. Differences in design, which result in significant differences in heat flow and distribution in the engine, may cause emphasis to be placed in some cases on such problems as ring-sticking, wear, and piston scuffing, and in other cases, on sludge formation, varnish, coking, and so on. Seemingly contradictory in nature, these problems have been extremely confusing, and it has been rather difficult to find any simple basic relationship among them. A great deal of progress toward solving them has already been made during the last few years by the engineers and technical men in both the automotive and petroleum industries, and it appears certain now that adequate solutions are or will be possible.

In general, all diesel engines are characterized by certain features from a lubrication standpoint which set them apart from gasoline engines and make them subject to consideration on a separate and distinct basis. Taken as a class, they (1) are more subject to ring-sticking, (2) form greater quantities of sludge, and (3) are subject to somewhat higher than normal rates of wear. Bearing corrosion should perhaps be included as a fourth problem, since the majority of the diesel engines are equipped with special alloy bearings, which are subject to chemical attack should the oil oxidize too rapidly or become contaminated with corrosive materials from any source. All of these problems are also known to gasoline engines but, except for bearing corrosion, they are generally of much less importance and are seldom major problems in present-day automotive operation.

The ring-sticking problem, which may be due either to varnish formation or to accumulation of deposits in the ring zone, is perhaps the most troublesome of the diesel-engine lubrication problems. It apparently occurs more frequently with the better refined paraffinic-type oils than with any others. Sludge formation, or separation of sludge

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 10, 1941.]

Table 1 - Analyses of Typical Diesel and Gasoline Engine * Sludge Deposits

	Sludge from Filter of Truck Diesel Engine **	Sludge from Filter of Truck Gasoline Engine **
Analysis of Insoluble (Chloroform) Portion:		
Ratio Chloroform Insoluble to Soluble	4	0.86
Carbon, %	70-90	30-35
Hydrogen, %	1-3	3
Ash, %	4-5	30
Weight Ratio of Carbon/Hydrogen	50/1	10/1

* Engines run for 84 hr on each oil at approximately 2000 rpm, 80%-90% full load, and same air, water, and oil temperatures.
** Averages of numerous runs.

deposits in the crankcase and on the filters, however, is probably almost equally a problem. The amount of sludge is generally less with the well-refined paraffinic oils, but is still considerable. Wear is perhaps not so common to all types of diesels, but it is more pronounced than with gasoline engines, and is more often noticed with oils of the naphthenic type.

Chemical analyses show that the diesel engine sludge deposits usually differ significantly from those found in the gasoline engine. Typical analyses of representative filter deposits taken from the two types of engines operating under approximately the same conditions are shown in Table 1. The results show that the insoluble material has a high carbon content and a carbon/hydrogen weight ratio of approximately 50/1, whereas the sludge from the gasoline engine has a lower carbon content and a much lower carbon/hydrogen weight ratio of approximately 10/1. It would appear that the deposits from the diesel are mostly of a soot-like nature, and that they originated largely in the combustion chamber, rather than in oxidation of the oil in the crankcase.

The quantities of sludge which are formed and find their way into the crankcase vary considerably from engine to engine, apparently because of differences in engine, particularly combustion-chamber, design. For any given set of speed, load, or other similar operating conditions, they may be relatively light in some engines and extremely heavy in others but, in any case, they are normally considerably heavier than those obtained in gasoline engines operating under approximately the same general conditions. This latter point is well illustrated by the used oil analyses (given in Table 2) on crankcase drainings of an SAE 40 naphthenic oil from three different automotive engines, two diesels and one gasoline, all operating at the same speed (2000 rpm) and about the same percentage loads. The naphtha-insoluble and the chloroform-soluble (asphaltene) sludges on the gasoline engine sample were 55 mg / 10 g and 38 mg / 10 g, respectively. For the diesel engines they ranged from 123 to 185 and 7 to 30, respectively.

Under normal operating conditions, where loads, speeds, and crankcase temperatures are not too high, it has not been impossible to lubricate most of the available diesel engines reasonably well. By proper selection of lubricants – for example, naphthenic in the case of some engines, paraffinic in the case of others, and certain types of compounded naphthenic oils in others – quite satisfactory performance has been possible. In many cases where engines are operating under these relatively mild conditions, very excellent results have been and are being obtained using high-quality paraffinic motor oils in all types of automotive diesel engines. As operating conditions become more severe, however, through increases in loads, speeds, crankcase temperatures, and other factors, straight mineral oils become less suitable and performance in general is unsatisfactory. Ring-sticking tendency is more pronounced, crankcase-oil contamination is much worse, wear is greater, and deterioration of the oil through oxidation becomes an important factor. These more drastic conditions, which are now widely termed "severe-duty" or "heavy-duty" conditions, often make it impossible to obtain satisfactory performance with some engines on straight mineral oils for more than a limited time before troubles such as excessive sludge formation, ring-sticking, piston scuffing or seizing, and so on, occur. Even more than in the case of gasoline engines, the tendency in the diesel-engine field recently has been to operate under the more severe conditions. This, together with the fact that inherently the diesel engine offers more lubrication difficulties, has made the problem doubly difficult.

The effect of increasing the severity of the operating conditions on performance is illustrated by the data which

Table 2 - Effect of Diesel and Gasoline Engines on Used Oil Conditions *

(84-hr Tests, 2000 rpm, 75%-90% Rated Load, 190 F Oil Temperature, 180 F Water Temperature, 7/3 Cycle **)

	Diesel Engines		Gasoline
	Engine A	Engine B	Engine
Used Oil Analyses:			
Naphtha Insoluble (mg/10 g)	123-185***	131	55
Chloroform Soluble (Asphaltenes - mg/10 g)	25-30	7	38
Neutralization Number (mg KOH/g)	2.0-2.4	1.7	1.9

* SAE 40 naphthenic oil (35 V.I.) used in all tests.
** Seven min at indicated load; 3 min idle.
*** Several tests run in this engine.

were obtained in several laboratory engine tests on two representative automotive diesel engines operating under what might be considered relatively normal conditions, and under what might be termed heavy load conditions. The normal operating conditions for the two engines consisted of a speed of 2000 rpm and a load of 50 bhp. For the heavy load conditions the speed was reduced to 1500 rpm, but the load was maintained at around 45-50 bhp. The percentage of rated loads for the two engines at the two respective speeds was therefore around 80% and 100%. The tests under these two types of operating conditions were run on three oils, one a well-refined paraffinic oil and the other two representatve naphthenic oils, both of which have found some application in diesel engine lubrication. The results of the tests are summarized in Table 3. In every case the operation at the higher percentage power output resulted in a much greater contamination of the used oil, the naphtha-insoluble content amounting to from four to twelve times that found when running at the lower percentage load. The paraffinic oil (Oil 1) also showed greater ring-sticking tendencies under the high load operation.

A satisfactory diesel-engine lubricant for severe-duty operation should therefore be able, to a greater degree than any other oil, to accommodate the contaminating sludge from the combustion chamber and from the oxidation of

the oil, to reduce ring-sticking by keeping the ring zone free of carbonaceous deposits, or by minimizng varnish formation, and to reduce wear by improving lubricating characteristics or by permitting the use of base stocks of the paraffinic type which have inherently more satisfactory lubricating properties. It is becoming increasingly apparent that straight mineral oils have for the most part failed to meet these requirements, and it is rather widely conceded at the present time that only certain specially compounded lubricants have thus far been really satisfactory under the severe-duty operation conditions. One diesel-engine manufacturer now recommends that such specially prepared lubricants be used whenever the severe operating conditions are encountered, where straight mineral oils are no longer able to lubricate the engine properly. Another recommends that they be employed in its equipment under all operating conditions. As the knowledge regarding the new types of compounds which are required for the purpose becomes greater, and the improvements in performance which are possible through their use are more completely evaluated, it is probable that other diesel engine manufacturers will adopt similar practices.

The additive compounds which have been employed in the manufacture of these new lubricants are of various types, termed "inhibitors," "dispersers," and "detergents," the term "detergent" normally covering both the detergent and dispersive properties. Some have one of these properties alone, while others combine two or more of them. The most successful have been the compounds combining at least the last two properties, and preferably all three. Compounds which are only inhibitors have quite often proved successful in one or more types of diesel engines but not in others. Similarly, compounds which are merely detergents and dispersers, but not inhibitors, are limited in application because they do not prevent or retard oxidation of the mineral oil base, and are therefore corrosive to alloy bearings. The first compounds of the detergent type to show promise were oil-soluble soaps which were reasonably good detergents, but which were oxidation accelerators rather than inhibitors. These could not be employed in engines containing alloy bearings of a corrosion-susceptible type. More recently, other metallic compounds or mixtures of compounds which have combined all three properties have been developed, and are being used successfully.

Until very recently, and even at the present time in most cases, one of the drawbacks to the new types of compounded oils has been the fact that their use was largely limited to low V.I. naphthenic-base stocks, either because of incompatibility of the additive with paraffinic-bases, or because of its inability to overcome some of the seemingly inherent weaknesses of the paraffinic-base stocks. It was thus impossible to provide the most satisfactory lubricant for the types of engines requiring the more paraffinic oils, or to provide for the other engines products having many of the desirable characteristics, such as low viscosity-temperature coefficient (high V.I.), good inherent oxidation stability, high resistance to scuffing and wear, and so on, which are much to be desired, and which are characteristic of the high V.I. paraffinic stocks. Such a situation has often made it necessary for an operator to carry two or more lubricants for lubricating the different types of diesel-engine equipment he may have on hand, or to make distinct concessions in lubricant quality and performance in some engines, in order to eliminate the necessity for handling more than one type of oil. Recent progress, however, has been rather rapid, and new developments have now made it possible to provide lubricants in the range of 100 V.I., which not only perform excellently in the types of diesels which have previously performed best with paraffinic lubricants, but also perform better in other engines than the compounded naphthenic lubricants which have heretofore shown to greatest advantage.

The present paper will review the progress that has been made thus far in this field, and indicate as far as possible the most important recent developments. The research which forms the basis of the paper was carried out principally on three of the better known makes of commercial diesel engines. They were selected because they represented three rather distinct types of engine design, and because each manifested certain of the distinct but characteristic lubrication problems which, considered together, included practically all the operating and lubricating problems peculiar to the diesel engine. A brief consideration of the three engines and a discussion of their individual peculiarities will perhaps better illustrate the nature of the problems and indicate the need for the modern lubricants of improved quality.

Engine I is particularly subject to ring-sticking, espe-

Table 3 - Effect of Severity of Operation on Engine and Used-Oil Conditions

	Diesel Engine A (84-hr Tests)				Diesel Engine B (70-hr Tests)			
	Oil 1 Paraffinic, 105 V.I.		Oil 2 Naphthenic, 35 V.I.		Oil 2 Naphthenic, 35 V.I.		Oil 3 Naphthenic, 50 V.I.	
	Moderate	Heavy Load	Moderate	Heavy Load	Moderate	Heavy Load	Moderate	Heavy Load
Engine Test Conditions:								
Speed, rpm	2000	1500	2000	1500	2000	1500	2000	1500
Load, bhp	50	45	50	45	50	50	50	50
Load, % of rated	80	100	80	100	75	100	75	100
Engine Demerit Ratings (% Based on Reference Oil):								
Overall	124	155	100	140	100	205	109	207
Sludge	100	175	100	247	100	295	121	343
Ring Grooves	109	153	100	139	100	122	105	133
Ring Sticking:								
Number of Rings Stuck	1	5	None	None	None	None	None	None
Analysis of Used Oils:								
Naphtha Insoluble, mg/10 g	28	354	123	447	131	896	285	1037
Chloroform Soluble, Asphaltene mg/10 g	2	8	27	35	6	19	10	26
Neutralization No. mg KOH/g	2.6	2.55	2.30	2.24	1.7	2.24	2.34	....

Table 4 - Operating Conditions for Engine Tests

	Engine I (6-Cyl)	Engine II (Single-Cyl)	Engine III-A (4-Cyl)	Engine III-B (3-Cyl)	Engine IV (6-Cyl)
Speed, rpm	1500	850	2000	2000	1600
Load, bhp	85	16.7	106	78.6	65
Load, % of Rated	95	100	100	100	100
Test Duration, hr	84	500-1000	500	500	84
Equivalent Miles	3800*	....	30,000**	30,000**	3800*
Water Out Temperature, F	180	180	180	180	180
Crankcase Oil Temperature, F	210	145	230	220	220
Type Operation	Steady	Steady	Steady	Steady	Cycle***
Oil Change	None	60 hr	None	None	None

* Calculated at 45 mph.
** Calculated at 60 mph.
*** Operated in a cycle of 7 min at indicated load; 3 min idle.

cially with paraffinic lubricants. It also forms rather heavy sludge deposits, and allows higher than normal wear, both of which are greatest with naphthenic oils. In spite of the wear and heavier sludge, however, naphthenic oils have been used most often in this engine in order to minimize ring sticking. Engine II has shown somewhat the same tendencies as Engine I, but usually not to the same degree. Carbonaceous deposits underneath the upper compression rings have also been a problem. Of the straight mineral oils, the naphthenic products have performed best in this engine. However, until just recently, only naphthenic oils containing special detergent additives have given really satisfactory performance. Engine III, unlike Engines I and II, is not troubled particularly with ring-sticking, and normally performs better with paraffinic lubricants. On the other hand, scuffing, and even seizing, of the pistons, wear, and formation of deposits underneath the piston crown have often been problems when employing naphthenic oils in Engine III.

Test conditions employed in the laboratory for evaluation of lubricating oils are outlined for each engine in Table 4. In Engine I, tests are conducted for a period of 84 hr at 1500 rpm and 95% full load. No oil change is made during the test. Reference runs are inserted at frequent intervals to allow for changes in mechanical condition of the engine with age, and results are corrected to a reference basis. Between each run the engine is dismantled for inspection and thoroughly cleaned. Engine II is generally operated for a period of 500 to 1000 hr at conditions (100% rated load) recommended by the manufacturer to simulate operation in the field. Sixty-hour oil change periods are employed in accordance with the recommended field practice. At intervals during the test, the engine is dismantled for inspection so that changes in engine condition with time may be recorded. For each test a new piston assembly and cylinder liner are installed. Engine III is normally operated for 500 hr without oil change at 2000 rpm and 100% rated load. These conditions are far more severe than encountered in field service, and are recommended by the manufacturer only for testing lubricating oils in the laboratory. At intervals during the test, rough examination of the engine condition is made by removal of the air-box covers, crankcase pan, and valve-chamber covers. At the same time, the connecting-rod bearings are removed and weighed to determine the extent of corrosion. Upon conclusion of the test, all engine parts are carefully inspected and rated. The engine is then thoroughly cleaned and overhauled, new pistons and cylinder liners being installed for the next test.

A few additional tests have been made in a fourth type of diesel engine which is operated for a total period of 84 hr with a cycle of 7 min under load (1600 rpm, 100% rated load) and 3 min idle.

In evaluation of the engine tests from the standpoint of lubricant performance, three fundamental factors are considered: (a) engine cleanliness (including ring-sticking); (b) mechanical wear, scuffing, and scratching; and (c) bearing corrosion.

Engine cleanliness is evaluated by means of a demerit rating system which is based upon the assignment of values from 0 to 10 to the different engine parts or surfaces, 0 representing a new or "perfect" surface, and 10 representing essentially the worst condition (from the standpoint of deposits) which could be expected to exist on that surface. Intermediate figures are recorded for conditions between these extremes.

For purposes of rough classification, an "overall" demerit is calculated from the individual demerits, different weightings being given to the individual demerits, depending upon the engine in question. For example, in Engine II, where ring-zone condition is the critical factor in successful operation of the engine, the ring-zone ratings are

Table 5 - Inspection of Test Oils

	Mineral Oils						Oils Containing Detergent-Disperser-Inhibitor Additive				Inhibited Oil
	Oil D SAE 30	Oil J SAE 30	Oil F SAE 30	Oil E SAE 20	Oil E SAE 30	Oil K SAE 20	Oil H SAE 30	Oil A SAE 20	Oil A SAE 30	Oil B SAE 30	Oil G SAE 30
Gravity, deg API	22.8	26.1	28.1	29.8	29.0	32.2	22.5	29.3	26.6	21.1	26.9
Flash Point, F	405	440	460	460	480	390	405	450	460	390	440
Viscosity at 100 F	514.0	517.1	489.4	299.0	500.3	256.1	547.4	309.1	517.0	669.0	567.0
Viscosity at 210 F	55.7	58.7	62.1	52.1	64.2	53.6	56.9	52.7	64.8	52.5	64.7
Viscosity Index	39	66	94	96	100	120	41	96	99	10	87
Color (Robinson)	12½	10¾	6¾	10	4½	10¼	9½	9	3¼	2	1¼
Carbon Residue, %	0.03	0.02	0.12	0.02	0.04	0.02	0.42	0.58	0.74	0.31	0.30

given considerably more weight than ratings of the piston skirt, cylinder liner, filter, and so on. Similarly, where a particular zone is important (such as the ring zone in Engine II), a composite rating for that particular zone is calculated.

The oils employed for the engine tests were chosen largely because they are representative of the different types which have been used in diesel engines and because they illustrate the effect of the newer-type additives on performance. Laboratory inspections of the oils are listed in Table 5. Oils D, E, F, J, and K are typical naphthenic and paraffinic mineral oils commonly employed for diesel-engine lubrication under moderate conditions of speed, load, and temperature where extreme detergency and oxidation stability are not required. Oil G represents a commercial inhibited oil of the paraffinic type which has found some application in diesel engines requiring extreme oxidation stability but only little detergency. Oils H and A are naphthenic and paraffinic oils of 40 and 100 V.I., respectively, to which have been added small amounts of a detergent-disperser-inhibitor type additive developed by the Esso Laboratories of the Standard Oil Development Co. These oils have been subjected to extensive laboratory and field testing, and have been found to meet the most severe requirements of heavy-duty diesel service in modern transportation and industry. Oil A is the only oil in the range of 100 V.I. which has fully satisfied the test requirements established by both the Caterpillar Tractor Co. and the General Motors Corp., for lubricants for their diesel engine. Oil B is another commercial detergent-disperser-inhibitor blend with a naphthenic-base oil.

The performance of these oils in the different engine tests is shown by the data in Tables 6 through 15.

■ Ring-Sticking and Ring-Zone Condition

Ring sticking has been an important problem in Engine I, even in the short 84-hr laboratory test. As shown in Table 6, for example, the paraffinic Oil E gave a ring-sticking demerit of 10.0, equivalent to complete sticking of two compression rings on each piston. With the naphthenic Oil D, however, ring-sticking was much less, giving a demerit of 4.78. This ring-sticking tendency of Oil E appears to be typical of that with many well-refined paraffinic oils, and seems to be associated with the greater tendency of such oils to form varnish or lacquer-like deposits which adhere strongly to the ring and groove sides. Possibly due to their greater volatility, naphthenic oils such as Oil D generally show less tendency toward varnish formation and tend rather to give ring-zone deposits of a drier, friable nature which are more readily removed from the ring and groove sides by the normal ring action. As a result, naphthenic oils have generally been preferred for the lubrication of Engine I, although certain paraffinic oils have been found satisfactory, as indicated by the results with Oil F (Table 6), where the ring-sticking demerit (3.93) is even lower than that with the naphthenic oil D.

With the newer detergent-dispersive-type oils it has been possible to reduce ring-sticking markedly, regardless of base stock type, indicating the ability of the detergent-dispersive type additive to prevent the accumulation of both varnish-type and carbonaceous-type deposits. The effectiveness of the detergent-dispersive type oils is shown in Table 6 by the tests with Oils A and H, where ring sticking has been reduced some 50 to 60% below that encountered with the naphthenic Oil D, and approximately 80% below that with the paraffinic Oil E (demerits of 2.20 and 1.96 for Oils H and A, versus 4.78 and 10.0 for Oils D and E). Fig. 1 shows the improvement in piston and ring zone cleanliness obtained with Oils A and H as compared with Oils E and D.

Ring-sticking likewise presents a problem in Engine II when straight mineral oils are used for its lubrication, and it is largely for this reason that the engine manufacturer has recommended the use of detergent or detergent-dispersive type oils under all operating conditions. As in Engine I, ring sticking in Engine II is generally much worse with the paraffinic oils and, at least in the lower ring zone, again appears to result from the accumulation of varnish-like deposits on the ring and groove sides. Particularly in the upper ring zone in Engine II, however, rather hard carbonaceous deposits tend to build up in the grooves behind the rings, and on the ring and groove sides. With paraffinic-type oils, such deposits generally build up quite rapidly, as shown in Table 7, where ring-

Table 6 - Performance of Different Oils in Engine I

	Mineral Oils				Oils Containing Detergent-Disperser-Inhibitor Additive	
	Oil D SAE 30	Oil J SAE 30	Oil F SAE 30	Oil E SAE 20	Oil H SAE 30	Oil A SAE 20
Viscosity Index	40	70	90	100	40	100
Cleanliness Demerits *						
Overall	3.03	3.36	2.35	3.04	1.42	1.26
Ring-Sticking	4.78	7.50	3.93	10.0	2.20	1.96
Ring Zone	3.97	4.10	3.38	3.94	1.96	1.84
Sludge	2.03	2.16	1.39	1.27	0.76	0.53
Oil Filter	4.50	5.16	3.50	2.32	2.00	1.25
Piston Underside	4.00	4.55	1.07	2.53	0.25	0.25
Piston Skirt	1.58	2.14	1.53	2.08	1.09	0.59
Average Maximum Ring Gap Increase per Cylinder, in	0.0050	0.0034	0.0030	0.0020	0.0022	0.0020
Copper-Lead Bearing Weight Loss, g per Bearing	0.005	0.021	0.049	0.068**	0.004	0.025

* Demerit ratings are corrected to a reference basis, using Oil D as reference.
** Estimated life of bearing corroding at rate of 0.068 g per 84-hr test period is upward of 3000 hr (equivalent to more than 120,000 miles).

Table 7 - Performance of Different Oils in Engine II

	Mineral Oils		Oils Containing Detergent-Disperser-Inhibitor Additive		
	Oil D SAE 30	Oil K SAE 20	Oil H SAE 30	Oil A SAE 20	Oil B SAE 30
Duration of Test, hr	500	126	1000	1026	1000
Cleanliness Demerits:					
Overall	2.45	2.33	1.05	0.97	1.94
Ring Zone	2.87	2.98	1.48	1.00	2.78
Piston Skirts	3.00	4.00	0.06	0.06	0.25
Oil Filter	7.00	1.00	0.75	0.75	1.0
Number of Rings Stuck	0	1	0	0	1

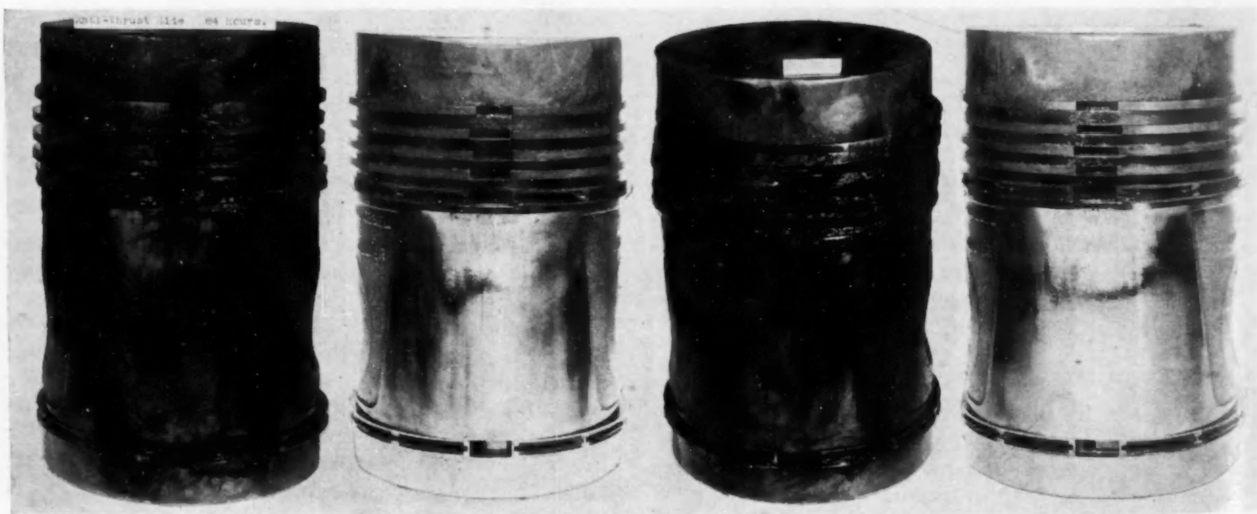
Table 8 - Comparison of Laboratory and Field Tests—Engine Type II—
Oil H (Containing Detergent-Disperser-Inhibitor Additive)

	Laboratory	Average of Five Field Units	Individual Field Units No. 1	No. 2	No. 3	No. 4	No. 5
Number of Cylinders	1	23 (Total)	3	6	6	4	4
Duration of Test, hr	1000	1064	1058	1024	1086	1168	1186
Cleanliness Demerits:							
Overall	1.05	1.16	1.17	1.23	1.14	1.09	1.17
Ring Zone	1.48	1.67	1.62	1.73	1.59	1.67	1.77
Piston Skirt	0.06	0.35	0.25	0.47	0.20	0.22	0.62

sticking and heavy ring-zone deposits (2.98 demerit rating) were encountered after only 126 hr of operation with Oil K. Under the same operating conditions with naphthenic Oil D, 500 hr of operation was obtained without ring sticking, although the accumulation of deposits in the ring zone (2.87 demerit rating) was of such magnitude that ring-sticking appeared to be imminent, and the test was discontinued. Considerably better performance was obtained with the compounded Oil B, and the period of operation could be extended to 1000 hr before ring-sticking occurred and ring-zone deposits became heavy.

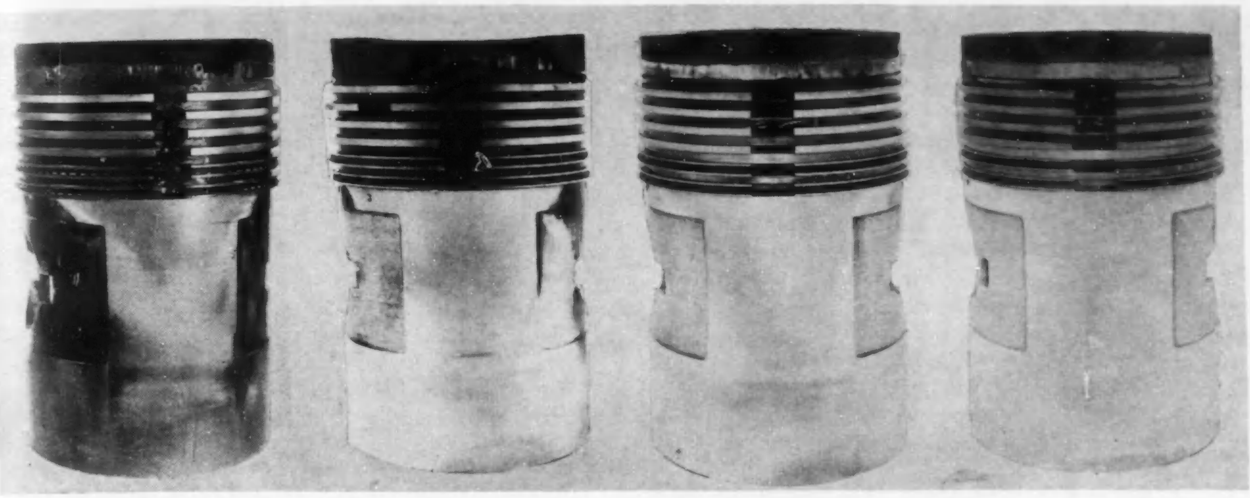
With the detergent-disperser-inhibitor-containing Oils A and H, however, no ring-sticking was encountered in runs of 1026 and 1000 hr duration and, on the basis of the excellent ring-zone and piston-skirt condition, many hundreds of hours of further operation would have been possible. The remarkable cleanliness of the piston and ring zone in the 1000-hr tests with oils A and H is illustrated in Fig. 2, where the pistons after 250 hr of operation with the straight naphthenic oil D and after 1000 hr of operation with Oil B are also shown for comparison. Enlarged photographs in Fig. 3 show even more clearly the improvement in ring-zone condition with the detergent-disperser-inhibitor type oils. The appreciably better condition resulting from lubrication with the paraffinic-base Oil A is clearly visible.

Remarkably similar results were obtained in 1000-1200 hr field tests under somewhat more severe conditions than employed in the laboratory. No ring-sticking whatsoever was encountered and, as shown in Table 8, the average overall cleanliness demerit for five field units was 1.16, as compared with 1.05 in the laboratory engine. Ring zone



OIL D OIL H OIL E OIL A

■ Fig. 1 – Appearance of pistons after 84-hr tests in Engine I



250-HR TEST
OIL D

1000-HR TEST
OIL B

1026-HR TEST
OIL A

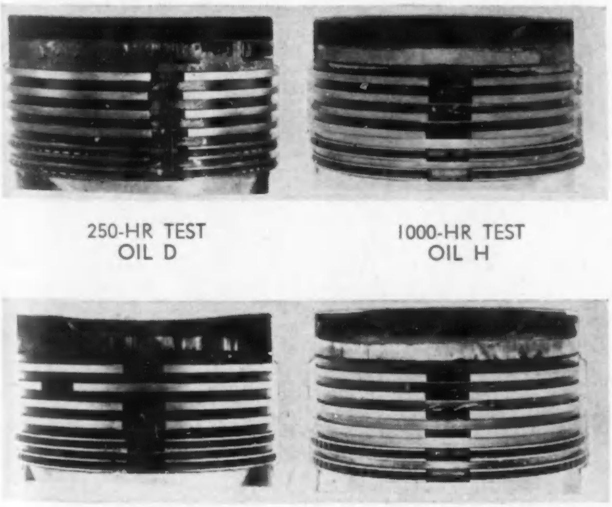
1000-HR TEST
OIL H

■ **Fig. 2 – Appearance of pistons after tests in Engine II**

demerits (1.67 versus 1.48) and piston skirt demerits (0.35 versus 0.06) were likewise in close agreement, which is shown by the comparison of field and laboratory pistons in Fig. 4.

Under ordinary conditions, ring-sticking is not a particular problem in Engine III, but in extremely severe field service and in the severe 500-hr laboratory endurance test without oil change, ring-sticking has been encountered. As shown in Table 9, for example, a ring-sticking demerit of 6.5 was obtained in 301 hr of operation with the paraffinic mineral Oil E in the laboratory Engine III-A, and a demerit of 6.0 in 500 hr of operation with the inhibited Oil G in the somewhat more severe Engine III-B. In 500 hr of operation with the detergent-disperser-inhibitor containing Oil A, ring-sticking demerits were reduced to 0.0 and 1.50 in Engines III-A and III-B, respectively.

More important problems in the operation of Engine III in heavy-duty field service have been the accumulation of



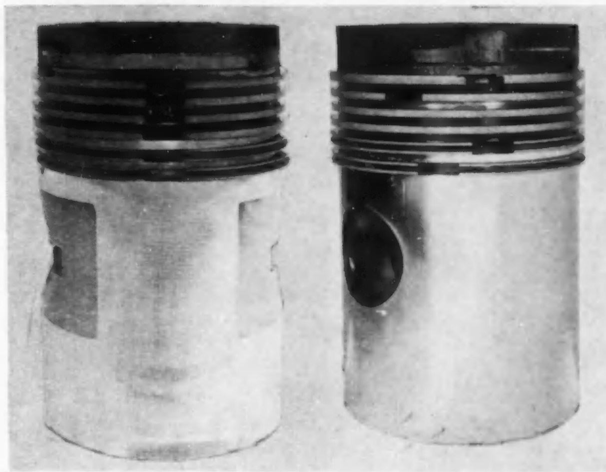
250-HR TEST
OIL D

1000-HR TEST
OIL H

1000-HR TEST
OIL B

1026-HR TEST
OIL A

■ **Fig. 3 – Piston-ring zones after tests in Engine II**



LABORATORY TEST PISTON
1000-HR
OIL H

FIELD TEST PISTON
1168-HR
OIL H

■ **Fig. 4 – Comparison of laboratory and field test pistons**

carbonaceous deposits in the ring zone, on the piston skirts, and on the piston under the crown. As these deposits continue to build up, heat transfer is greatly reduced and, in extremely severe cases, piston cracking or seizure has occurred. Although these difficulties have now been reduced to an appreciable extent by mechanical changes in the engine, emphasis continues to be placed on lubricants with the highest possible stability toward decomposition and oxidation. For this and other reasons, paraffinic oils are preferred to naphthenic oils for lubrication of Engine III. However, marked improvements over straight paraffinic mineral oils and even over strongly inhibited paraffinic mineral oils may be obtained with detergent-dispersive-inhibitor type oils, as shown in Table 9. In the severe laboratory endurance tests, ring-zone, piston-skirt, and piston-underside demerits are considerably lower after 500 hr of operation with Oil A than after 301 hr of operation with the paraffinic mineral Oil E or after 500 hr with the inhibited Oil G.

■ Sludge and General Engine Cleanliness

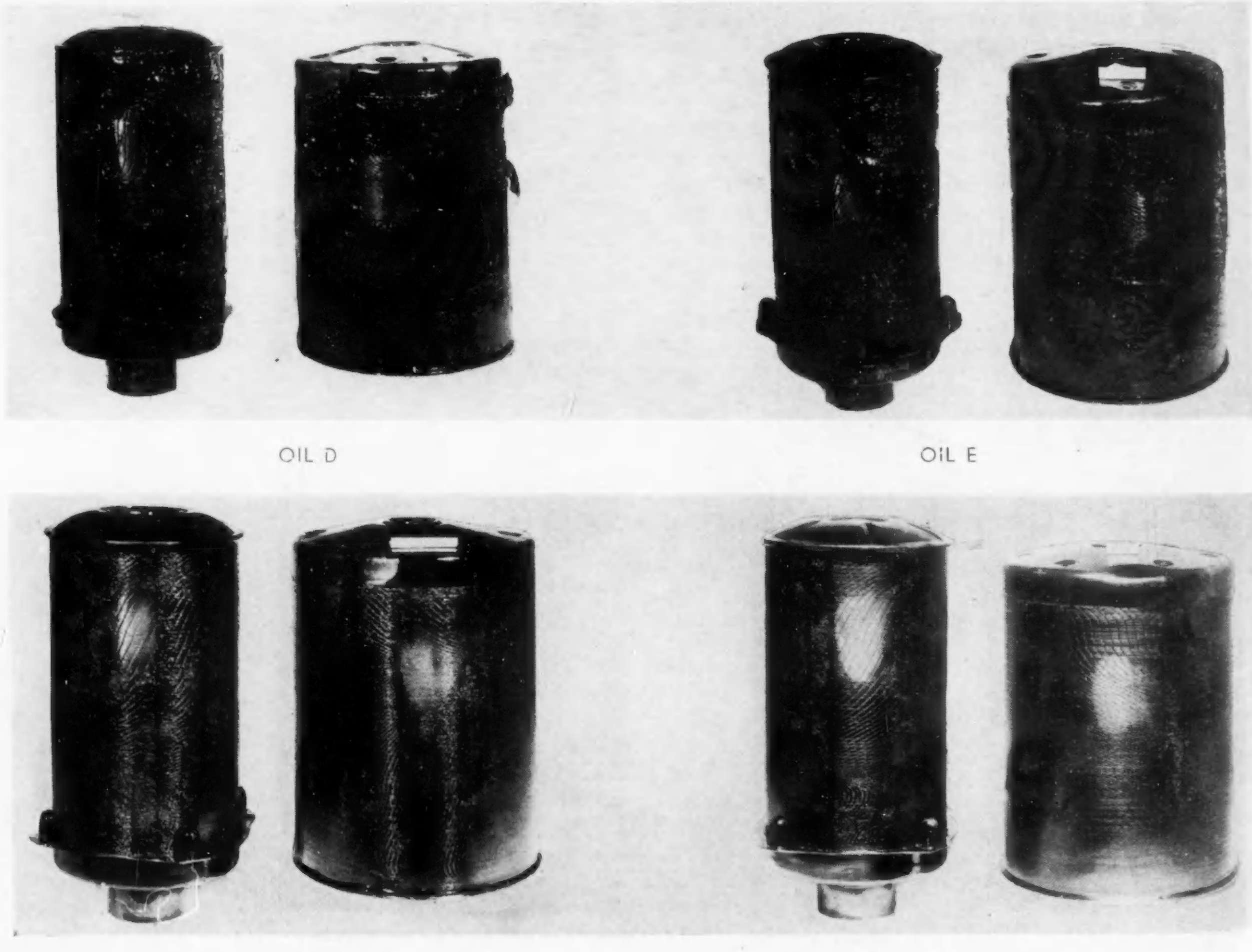
The detergent-disperser-inhibitor-containing oils have further shown a remarkable ability to prevent the accumulation of sludge and other contaminants on engine surfaces and on the filters (strainers). With such oils, the condition of the filter has generally been a rather good index of overall engine condition, the cleaner the filter, the cleaner the engine also. With straight mineral oils this may not always be the case.

When operating under heavy-duty conditions, Engine I forms relatively large quantities of sludge which build up to considerable proportions, even in the 84-hr laboratory test. As shown in Table 6, sludge and filter deposits are particularly heavy with naphthenic or low V.I. mineral oils such as D and J. Well-refined paraffinic oils such as Oils F and E give less sludging due to their inherently greater resistance to oxidation but, even with them, sludge accumulations are relatively heavy. The use of the detergent-disperser-inhibitor type Oils A and H results in a 60-75% reduction in sludge deposits and a 50-75% reduction in filter deposits when compared with the uncompounded mineral oils. The paraffinic-base Oil A is somewhat more effective than the naphthenic-base Oil H in this respect. Photographs of the strainers from 84-hr tests with Oils A, H, E and D are shown in Fig. 5 to illustrate the extent of improvement with the detergent-dispersive type oils.

Sludging is less severe in Engine II, although rather heavy filter deposits result from operation with straight mineral oils, particularly those of naphthenic origin. Reference to Table 7 and Fig. 6 will show the extremely clean filter condition maintained after 1000 hr of operation with the additive-containing Oils A, B, and H, as compared with that after only 250 hr of operation with naphthenic Oil D.

Under normal conditions, no particular difficulties from the standpoint of sludging have been encountered in Engine III, although plugging of intake ports with sooty and carbonaceous deposits is generally more troublesome with non-detergent mineral oils. In the 301-hr test with mineral Oil E (Table 9), for example, port plugging resulted in sufficient loss of power that cleaning was necessary at the 218-hr period. Likewise, with the inhibited Oil G, excessive drop in load necessitated port cleaning at 300 hr. In the tests with the detergent-disperser-inhibitor Oil A, the full 500 hr of operation was possible without shutdown for port cleaning.

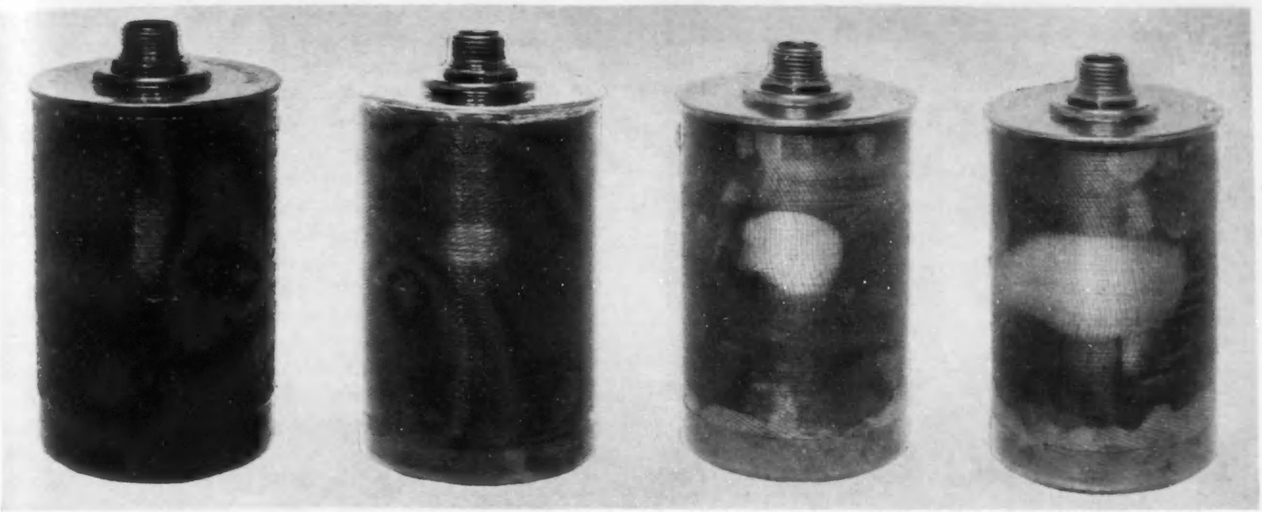
In the somewhat less severe 84-hr tests in Engine IV, little or no deposits of sludge or piston skirt varnish were observed, even with the naphthenic Oil D (Table 10). Carbonaceous deposits tend to form in the ring zone and on the underside of the piston crown, however, and it is probable that these deposits would be limiting factors in longer time or more severe operations. Marked reductions



OIL D OIL E

OIL H OIL A

■ Fig. 5 – Appearance of oil filters after 84-hr tests in Engine I



250-HR TEST OIL D | 1000-HR TEST OIL B | 1026-HR TEST OIL A | 1000-HR TEST OIL H

■ Fig. 6 – Oil filters after tests in Engine II

Table 9 – Performance of Different Oils* in Engine III

	Engine III-A (4-Cyl)		Engine III-B (3-Cyl)	
	Mineral Oil Oil E	Oil Containing Detergent-Disperser-Inhibitor Additive Oil A	Oil Containing Detergent-Disperser-Inhibitor Additive Oil A	Commercial Inhibited Oil Oil G
Duration of Test, hr	301	500	500	500
Cleanliness Demerits:				
Overall	3.45	1.83	2.84	3.55
Ring Zone	4.24	3.25	3.40	4.07
Ring Sticking	6.50	0.0	1.50	6.00
Piston Underside	2.00	0.50	1.00	1.50
Piston Skirt	5.00	3.75	2.25	6.50
Number of Rings Stuck	4	0	1	3
Intake Port Cleaning, hr	218	None	None	300
Copper-Lead Bearing Weight Loss, g per bearing	7.10	0.048	0.047	0.447
Used Oil Analyses at End of Test:				
Neut. No., mg KOH/g	2.50	0.35	0.70	0.42
Sap. No., mg KOH/g	10.5	3.10	5.6	7.4
Maximum Cylinder Liner Diameter Increase, in./500 hr	0.0015	0.0011	0.0014	0.0033

* SAE 30 grade.

in ring-zone and piston-underside deposits were obtained with the detergent-disperser-inhibitor type oils, the paraffinic-base Oil A again being more effective in this respect.

The very potent detergent and dispersive properties of the additive-containing Oils A and H, which minimize the accumulation of deposits in new engines to which the oils are charged, will also exert a pronounced purging action on dirty engines. Table 11 shows the results of tests with Oil A in Engine I. The engine was operated for three successive 84-hr periods (252 hr total) without oil change, during which time the overall and ring-sticking demerits had increased to 2.35 and 3.66, respectively. The oil was then changed and the engine operated for a period of 84 hr on the fresh oil charge. At the end of the period, the amount of ring sticking had actually been reduced to a demerit rating of 2.56, which is reflected in a reduction of the overall demerit of 2.07. Ring zone, sludge, piston skirt, and other demerits remained essentially constant during the 84 hr of operation with the fresh oil charge.

Very similar results have been obtained in Engine II. In this case (see Table 12), fresh Oil H was charged to a dirty engine having overall, ring zone, and piston skirt

Table 10 - Performance of Different Oils in Engine IV

	Mineral Oil	Oil Containing Detergent-Disperser-Inhibitor Additive		
	Oil D SAE 30	Oil H SAE 30	Oil A SAE 20	Oil B SAE 30
Cleanliness Demerits:				
Overall	1.62	1.25	1.19	1.41
Piston Underside	3.84	1.50	1.00	1.25
Ring Zone	3.18	2.70	2.45	3.05
Piston Skirts	0	0	0	0
Sludge	0.86	0.41	0.59	0.66
Number of Rings Stuck or Tight	0	0	0	0
Copper-Lead Bearing Weight Loss, g per bearing	0.015	0.037	0.034	0.114

Table 11 - Purging Action of Fresh Oil A*—Engine I

Successive 84-hr Periods:	No. Oil Change 1	2	3	Oil Changed Before Starting Period 4
Cleanliness Demerits:**				
Overall	1.08	1.59	2.35	2.07
Ring Sticking	0.0	0.33	3.66	2.56
Ring Zone	1.61	2.21	2.76	2.74
Sludge	0.79	1.33	1.87	1.40
Piston Skirt	0.50	0.81	1.08	1.21
Number of Rings Stuck	0	0	4	2
Number of Rings Tight	0	1	1	2
Used Oil Analyses:				
Naphtha Insoluble, mg/10 g	366	540	610	395
Chloroform Soluble (Asphaltenes), mg/10 g	20	23	28	15
Neutralization No., mg KOH/g	0.42	0.48	0.48	0.45
Saponification No., mg KOH/g	3.86	4.45	5.63	6.15

* Contains detergent-disperser-inhibitor additive.
** Demerit ratings at end of Periods 1 and 2 based on inspections of two pistons only. Demerits at end of Periods 3 and 4 based on inspections of all pistons.

demerits of 2.36, 3.13, and 2.62, respectively. Operation of the engine was continued for 240 hr on the fresh oil charge, during which time the engine was inspected at 60-hr intervals. During the first 60 hr following oil change, overall, ring zone, and piston skirt demerits had actually decreased to 2.03, 2.91, and 2.50, respectively. Almost identical ratings were obtained after the second 60-hr period following oil change. After 180 hr, a slight increase in deposits had been observed, but almost 240 hr operation was obtained before the engine deposits had finally reached those at the start of the test.

This purging effect of the detergent-disperser-inhibitor type oils has also been accomplished in the field. When operated on conventional mineral oils, one particular engine in bus service accumulated piston and ring-zone deposits to such an extent that high oil consumption necessitated engine overhaul every 10,000-15,000 miles. At a period when the engine was scheduled for overhaul, the detergent-dispersive-inhibitor Oil A was charged to the engine instead, and the bus was put back into service. Inspection of the engine after 6000 miles with Oil A showed all rings to be free and clean, slits in oil rings virtually free of deposits, oil screens completely clean, and no heavy deposits elsewhere in the engine. Oil consumption had, at the same time, been reduced to normal.

■ Wear

Diesel engines in heavy-duty service are generally subject to rather high rates of wear. In Engine I, wear seems to be somewhat greater with naphthenic oils, which may be due in part to their natural deficiency in oiliness or lubricating properties, and in part to their higher volatility which allows them to evaporate from the cylinder wall too quickly at the higher temperatures encountered in heavy-duty service. From Table 13 it will be observed that an average of seven widely separated runs in Engine I with naphthenic mineral oils gives an average maximum ring-gap increase for each 84-hr test of 0.0035 in., the range of individual values being from 0.0021 to 0.0057 in. Two paraffinic oils show maximum ring-gap increases of 0.0020 and 0.0030 in. In contrast to this picture, the average maximum ring-gap increase with nine detergent-dispersive type naphthenic oils is 0.0010 in. per 84-hr test, and with ten detergent-dispersive type paraffinic oils is 0.0007 in. The detergent-dispersive-inhibitor type oils have thus reduced wear in Engine I by some 60 to 70% over that with the uncompounded oils.

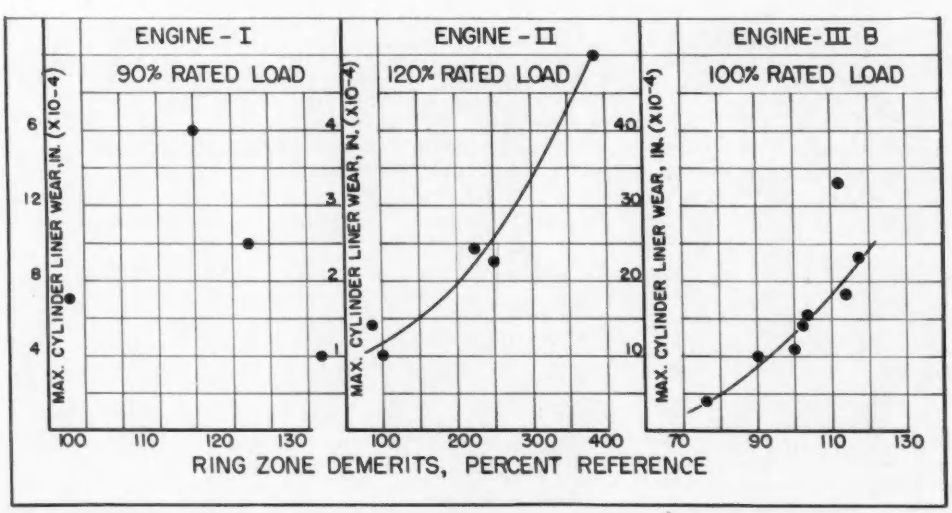
Table 12 - Purging Action of Fresh Oil H * - Engine II - Accelerated Tests at 140 F Air Temperature

	At Start of Test **	Hours After Oil Change 60	120	180	240
Engine Condition:					
Overall Demerit	2.36	2.03	2.04	2.18	2.34
Ring Zone Demerit	3.13	2.91	3.04	3.17	3.44
Piston Skirt Demerit	2.62	2.50	2.25	2.50	2.50

* Contains detergent-disperser-inhibitor additive.
** Fresh Oil H charged to engine in condition indicated by demerit ratings in this column. No oil change made during succeeding 240 hr. Make-up oil added as needed.

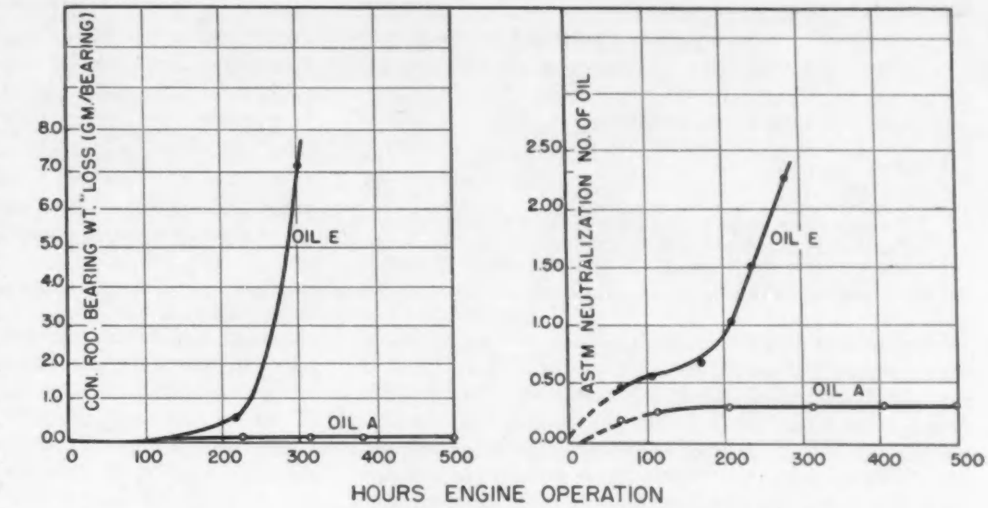
In Engines II and III, more than in Engine I, wear appears to be associated with ring-zone condition and ring-zone sluggishness. Ring-zone demerits as per cent of reference are plotted against maximum cylinder liner wear in Fig. 7 for 80-hr tests at high loads in Engines I, II, and III-B. A good correlation exists in Engines II and III-B, the wear increasing markedly with increasing deposits in the ring zone. The results in Engine I show no definite trend, indicating that oiliness or other factors are probably more important than ring zone-condition in determining wear in Engine I.

The effectiveness of the detergent-disperser-inhibitor type oils in reducing wear in Engine II en-



■ Fig. 7 – Relationship between ring-zone cleanliness and cylinder-liner wear – 80-hr engine tests

durance tests is shown by the data in Table 14. Cylinder-liner wear after 1000 hr of operation with the compounded Oils H and A was about 40% less than with the naphthenic mineral Oil D after only 500 hr. With all three oils a rather rapid break-in wear occurs during the first 60-100 hr of operation, due to the installation of a new piston assembly and cylinder liner for each test. Beyond the 100-hr point, however, wear with oils H and A proceeded very slowly whereas, with Oil D, the wear at 500 hr had more than doubled that in the first 100 hr.



■ Fig. 8 – Resistance of detergent-dispersive-inhibited Oil A to copper-lead bearing corrosion and development of acidity in Engine III-A

Remarkably low wear was also obtained with Oil H in the more severe field tests where dusty conditions of the atmosphere generally contribute to high wear in engines operating in this territory. Reference to Table 14 will show that the maximum cylinder-liner wear calculated as the average of five field units (23 cylinders) amounted to 0.0021 in. per 1000 hr, as compared with the laboratory value of 0.0017 in. per 1000 hr in a single run.

Although in laboratory tests wear has been less a problem in Engine III than in Engines I and II, the detergent-disperser-inhibitor type oils have shown reduced rates of wear, which is probably associated with the improved ring-zone condition. As shown in Table 9, the maximum cylinder-liner diameter increase per 500 hr with the compounded Oil A in Engine III-A was 0.0011 in., as compared with 0.0015 in. with the straight mineral Oil E. In the more severe Engine III-B, the maximum cylinder wear with Oil A amounted to 0.0014 in., as compared with 0.0033 in. with the inhibited Oil G.

Table 13 - Engine I Wear Measurements with Mineral and Compounded Oils, 84-hr Tests

Type Oil	No. of Tests	Average Maximum Ring-Gap Increase, in.	Range of Maximum Ring-Gap Increase, in.
Naphthenic	7	0.0035	0.0021-0.0057
Paraffinic	2	0.0025	0.0020-0.0030
Naphthenic+Additive	9	0.0010	0-0.0022
Paraffinic+Additive	10	0.0007	0-0.0021

■ Bearing Corrosion

In addition to detergent, dispersive, and wear-reducing properties, it is highly important that modern diesel lubricants be non-corrosive to the hard alloy bearings with which most diesel engines intended for high-speed or heavy-duty service are now being equipped. The degree of corrosion resistance required to give adequate protection of bearing surfaces, however, differs considerably from engine to engine, and even from one set of operating conditions to another in the same engine. In line with field experience, laboratory tests have shown Engine III to be particularly severe from the standpoint of oil oxidation, and lubricants of highly corrosion-resistant properties are required for satisfactory operation of the engine in heavy-duty service. Other engines such as Engines I and IV are generally not so severe in this requirement, and often can be lubricated satisfactorily with oils of lesser corrosion resistance. Indicative of the marked effect which operating conditions may have on corrosion, laboratory tests have shown that, with a number of oils, a temperature difference as little as 15 F in the range 200-350 F approximately doubles the rate of copper-lead bearing corrosion in the presence of air.

The problem has been complicated further by the complex nature of the chemical reactions involved, and by the failure of currently available laboratory corrosion tests to correlate with engine performance. This condition is particularly true when dealing with additive-containing oils, and has led to considerable confusion with regard to the possible effects of different additive types on bearing corrosion. Many of the earlier detergent-type additives, for example, greatly accelerated corrosion, apparently by promoting the rate of oil oxidation. Wolf[1] has recently pointed out, however, that such compounds may also attack alloy bearings, particularly of the copper-lead type, by a mechanism entirely independent of the base oil. With additives of this type, it is therefore necessary to provide inhibitors

Table 14 - Engine II Wear Measurements with Mineral and Compounded Oils *

Cumulative Maximum Cylinder Liner Diameter Increase, in.

	Laboratory Unit				Average of Five Field Units ***
Hours:	100	250	500	1000	1064
Oil D	0.0012	0.0021	0.0028	..	..
Oil H **	0.0012	0.0016	0.0015	0.0017	0.0021
Oil A **	0.0010	0.0012	0.0011	0.0016	..

* SAE 30 grade.
** These oils contain detergent-disperser-inhibitor additive.
*** Average of the maximum cylinder-liner diameter increase in each unit. Individual maxima ranged from 0.0013 to 0.0040 in.

[1] See SAE Transactions, April, 1941, pp. 128-137: "Crankcase Oils for Heavy-Duty Service," by H. R. Wolf.

Table 15 – Effect of Different Additives on Performance of Base Oil in Engine II (60-Hr Accelerated Tests at 140F Atmospheric Temperature)

Cleanliness Demerits, % Reference	Oil A* (Reference)	Experimental Oils **				Oil P***
		Oil L	Oil M	Oil N	Oil O	
Overall	100	127	120	131	119	137
Ring Zone	100	198	139	147	167	145

* Contains commercial detergent-disperser-inhibitor additive.
** These oils contain additives of some general type as that employed in Oil A, and are blended with the same base stock.
*** Another commercial detergent-disperser-inhibitor type oil.

which not only prevent corrosion by oil oxidation products, but also by the detergent additive itself. Some of the recently developed detergent-disperser-inhibitor type additives are in themselves non-corrosive, and in addition prevent the development of corrosive oil oxidation products. This is illustrated by the data in Table 9 and Fig. 8, where almost negligible bearing corrosion (0.048 g per bearing) occurred in the 500-hr endurance test with the detergent-disperser-inhibitor type Oil A in Engine III-A. In contrast, severe corrosion (7.1 g per bearing) developed in only 301 hr of operation with the uncompounded Oil E, resulting in discontinuation of the test. In the latter case, corrosion appears to have resulted from excessive oxidation of the oil since, as shown in Fig. 8, the sharp increase in bearing weight loss at about 200 hr coincides very closely with a rapid increase in neutralization number of the used oil. With the detergent-disperser-inhibitor type Oil A, neutralization number was maintained at a very low level throughout the 500-hr test.

Results of tests in Engine III-B (see Table 9) show further that the detergent-disperser-inhibitor type Oil A was considerably more effective than Oil G, which is simply an inhibited oil, the bearing weight losses after 500 hr of operation being 0.047 and 0.447 g per bearing, respectively.

Balance of Properties – It is apparent from the foregoing discussions that best overall engine performance can be obtained only with lubricants having a proper balance between detergency, dispersion, and inhibition. A lack of any one of these properties results in serious limitations of the oil for application to heavy-duty diesel service. Painstaking care in the selection of both additive and base stock is therefore necessary in compounding wholly satisfactory diesel lubricants, and both lengthy and extensive testing is required. As in the case of bearing corrosion, other laboratory test methods, such as those for sludging, oxidation rate, and so on, generally fail to correlate with engine performance, and are often misleading.

Table 15 shows the results of several tests in Engine II to indicate the relatively large differences in engine performance which may result from what appear to be only minor modifications in the additive compound. Oils L through O were all blended from the same base stock, and contained additives of roughly the same general types, but which differed in metallic and other substituent groups. Performance was compared with Oil A as reference. Overall cleanliness demerits ranged from 119-131% of reference. Individual ring-zone demerits showed much wider variations, ranging from 139-198% of reference. Oil P represents another commercial oil containing a detergent-disperser-inhibitor additive of the metallic type. Overall and ring-zone demerits were 137 and 145%, as compared with Oil A, again indicating that the metal-containing type of additive alone does not necessarily guarantee optimum performance.

Table 16 – Effect of Different Base Stocks* on Performance with Detergent-Disperser-Inhibitor Additive in Engine II (60-Hr Accelerated Tests at 140F Atmospheric Temperature)

Cleanliness Demerits, % Reference	Oil A (Reference)	Naphthenic			Paraffinic			
		Oil H	Oil Q	Oil R	Oil S	Oil T	Oil U	Oil V
Overall	100	129	118	120	114	117	140	169
Ring Zone	100	134	144	162	119	129	157	216

* All oils compounded with same concentration of the same commercial detergent-disperser-inhibitor additive employed in Oil A.

Table 16 summarizes data from several tests in Engine II to illustrate the differences which may result when employing different base stocks with the same additive. Overall demerits ranged from 114-169%, and ring-zone demerits from 119-216%, when compared with reference Oil A. With a given additive, therefore, many different base stocks may have to be evaluated to insure best overall performance.

Functions of Additives – It is rather remarkable that a relatively small quantity of added synthetic compound can improve the performance of a lubricant as effectively as demonstrated by the engine tests. Consequently, it would be of interest to know how the additive accomplishes its effects, particularly since an understanding of its action helps in predicting more completely what results may be

[2] See SAE Transactions, January, 1939, pp. 35-42: "Improvements in Diesel-Engine Lubricating Oils," by U. B. Bray, C. C. Moore, Jr., and D. R. Merrill.

[3] See SAE Transactions, December, 1937, p. 547: Excerpts from paper: "Effect of Addition Agents on Piston and Ring Performance," by C. M. Larson.

[4] See "Severe-Duty Engine Conditions as Related to Oil and Fuel," by C. M. Larson, presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 9-14, 1940.

[5] See SAE Transactions, November, 1939, pp. 485-500: "Recent Developments in Diesel Lubricating Oils," by G. L. Neely.

[6] See SAE Transactions, April, 1937, pp. 165-172: "Engine Temperature as Affecting Lubrication and Ring-Sticking," by C. G. A. Rosen.

[7] See *National Petroleum News*, Vol. 32, No. 18, May 1, 1940, p. R-152: "Deterioration Factors in Diesel Lubricants," by C. G. A. Rosen.

[8] See *National Petroleum News*, Vol. 32, No. 18, May 1, 1940, pp. R-149-150: "Utilization of Addition Agents," by G. A. Round.

[9] See *Oil and Gas Journal*, Vol. 38, No. 3, June 1, 1939, p. 44: "Engine Deposits, Field and Laboratory," by J. P. Stewart and B. W. Story.

[10] See *National Petroleum News*, Vol. 32, pp. R-88-89: "Use of Chemical Additives in Lubricating Oils," by J. P. Stewart, B. W. Story, and O. M. Reiff.

expected in different applications and under various conditions.

It is believed that the benefits obtained by the use of the detergent-disperser-inhibitor additive can be ascribed to three effects:

(1) The additive disperses and holds solid particles of deterioration and contamination products in suspension in the oil, and thereby prevents their deposition on engine parts as sludge, varnish, and ring-zone carbon;

(2) It retards the deterioration of the oil and the consequent formation of products corrosive to copper-lead and similar bearings, as well as substances otherwise objectionable;

(3) It increases the adhesion of the oil to the metal parts of the engine, thus helping the lubricant resist displacement by heavy rubbing loads, high temperatures, and corrosive vapors and, as a result, it reduces engine wear.

Table 17 - Effect of Detergent-Disperser-Inhibitor Additives on Accumulation of Sludge in Oil and on Filter (Engine I)

Oil	Naphtha Insoluble Sludge (mg/10 g) in Crankcase Oil	Sludge on Filter (Demerit Rating)
Naphthenic Oil D	462	7.0
Oil D+Inhibitor	468	7.0
Oil D+Disperser 1	687	2.0
Oil D+Disperser 2	707	1.0

Some of these observations have been made previously by other investigators in their study of related additives.[2 to 10, inc.]

Dispersing Action – The effect of the additive is immediately evident in the conspicuously cleaner filter condition noted in the different engine tests run on Oils A and H. It might appear at first that the additive enables the oil to keep cleaner, and that in consequence there is less material to filter out. This may be so in some cases. However, in most diesel operations the additive serves rather to carry the insoluble materials on through the filter. That such is the case is shown by the data in Table 17, obtained in a series of successive tests run in Engine I on various oils. As the quantity of naphtha-insoluble material accumulated on the filter decreased, that in the oil actually increased.

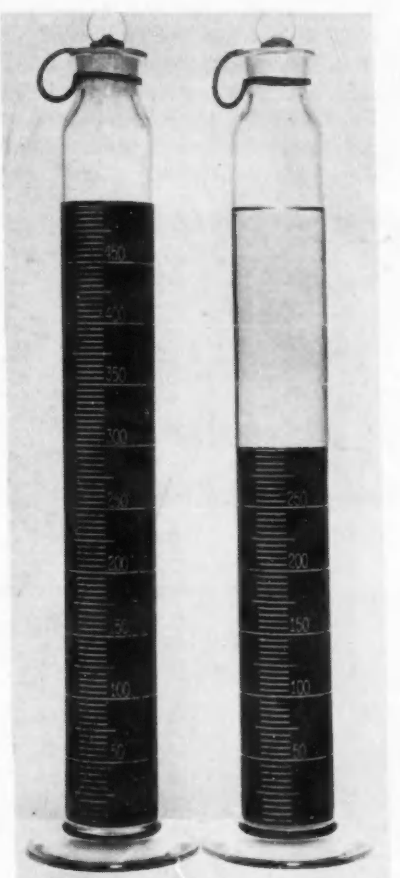
In carrying the insoluble materials through the filter, the additive functions as a disperser. Ordinarily the particles of insoluble material probably consist of aggregates of considerably smaller particles. The additive serves to break up these aggregates, or to retard their formation, thereby limiting more of the particles to dimensions which permit easy passage through the filter. For example, the individual carbon black particles, formed in the burning of gas flames, and such as might conceivably be formed in the combustion chamber of a diesel engine, are reported to have diameters of the order of 0.00001 in., which compares with a clearance of 0.003 in. in the strainer-type filters used in Engine I.

The ability of the additive to retard the settling of solids can be demonstrated quite clearly by simple laboratory tests. One example is illustrated in Fig. 9, which shows the degree of settling or separation of added carbon black in oils with and without the additive. Both of the cylinders shown were charged with 30 g of a representative solid, carbon black, thoroughly mixed with 450 g of oil, and permitted to stand for 22 hr at 200 F. In Cylinder A, the uncompounded oil was used as the vehicle, and marked setting of the carbon black is apparent. In Cylinder B, however, the additive blend has obviously greatly retarded the settling.

The dispersion effect is also further shown if the oil-carbon black mixtures are diluted with water white naphtha until both show the same degree of grayness in a color comparator. The oil containing the additive will be found to have required by far the greater dilution. This difference indicates that, in the presence of the additive, the carbon-black aggregates have been broken up to a far greater extent, and that the resulting greater number of carbon particles impart a greater apparent blackness to the additive blend.

Lubricants are of course not chosen because they are able to keep oil filters clean but, since the action on the filter is largely duplicated in other parts of the engine, a sludge-free filter is usually indicative of a sludge-free engine. This is particularly true of oils containing the detergent-disperser additives. Thus, as solid particles are more finely divided, they have less tendency to settle from the oil, and hence their deposition as sludge in the crankcase, valve chamber, and in oil lines is reduced.

The ability to prevent separation of insoluble substances is not limited to the effect of the additive on alien materials like carbon black; it also extends to the dispersion of such other insoluble substances as those formed in the chemical deterioration of the oil itself. For instance, Oil D was oxidized in the Indiana test for 72 hr and the quantity of naphtha-insoluble materials determined at 24-hr periods, both by centrifuging and filtering the usual naphtha solution through an asbestos pad. Standard ASTM precipitation naphtha was employed, and also this naphtha containing the additive in an amount proportional to that which would normally have been used in the oil. It is apparent from the data presented in Table 18 that the additive has reduced the separation of insoluble oxidation products by more than 50%.



CYLINDER B CYLINDER A

Fig. 9 – Dispersion of carbon black by oils in laboratory tests

A substance able to disperse contaminating substances as they accumulate in the oil may, under favorable conditions, also be expected to redisperse deposits which have already accumulated in the engine,

Table 18 - Effect of Detergent-Disperser-Inhibitor Additives - Dispersion of Insolubles from Oxidized Oil by Inclusion of Disperser in Precipitation Naphtha

Quantity of Disperser Added To Precipitation Naphtha, %	Relative Quantities of Insoluble Materials Separated By: Filtration	Centrifuging *
0	100	100
0.5	80	64
1.0	76	57
2.0	67	48

* 2000 rpm.

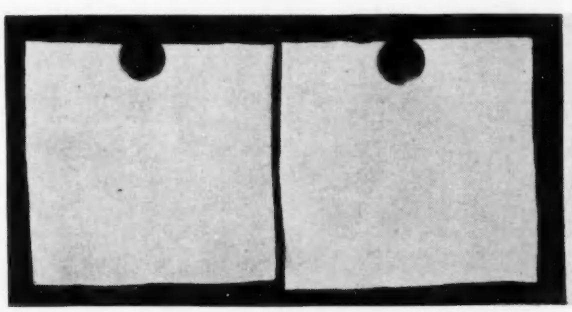
and thus exert a sort of cleansing action. This has already been shown to be the case in the engine performance tests. Bray, Moore, and Merrill[2] have likened the ability of the oil to clean up a dirty engine to a detergent or laundering action. A simple illustration of this effect is given in Fig. 10, which shows cloth specimens, dirtied by a carbon black-oil mixture, before and after washing in a Launderometer. In one case, conventional cleaner's naphtha was used; and in the other, the same naphtha containing the additive. In a clean engine the detergent effect probably operates concomitantly with that of dispersion.

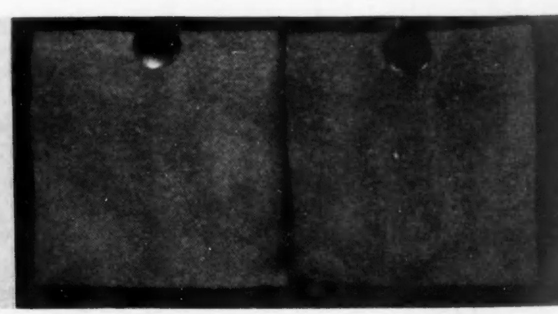
Prevention of Oil Deterioration – It seems probable that the effect of the additive in the foregoing different experiments was purely physical. In no case did the additive have an opportunity to retard the formation of the insoluble materials; it merely served to prevent their separation. Similar benefits can also be obtained, however, under conditions in which the additive may function chemically, or at least in which the action is not so clearly defined. Thus, an oil containing the additive exhibits a materially reduced tendency to form varnish deposits when it is subjected to oxidation. Fig. 11 shows glass microscope slides which had been immersed in Oils D, E, H, and A during the progress of a regular Indiana oxidation test, and it is apparent that the additive blends have left considerably the lighter varnish films. In the engine this effect is paralleled by lessened deposition of varnish on piston skirts, rings, and other engine parts, and hence reduced piston seizures and ring-sticking.

That the additive retards the chemical breakdown of the oil to form products which would be harmful, if not dispersed, is also illustrated by the data presented in Table 19, secured in a Chevrolet engine operated under the test conditions recommended by General Motors for studying heavy-duty oils.[11] Not only is the engine kept cleaner upon increasing the concentration of additive, but the oil deterioration indicated by the used-oil analyses becomes progressively less. Particular attention is called to the lessened increase in viscosity, neutralization and saponification numbers, since these oil characteristics vary with factors wholly

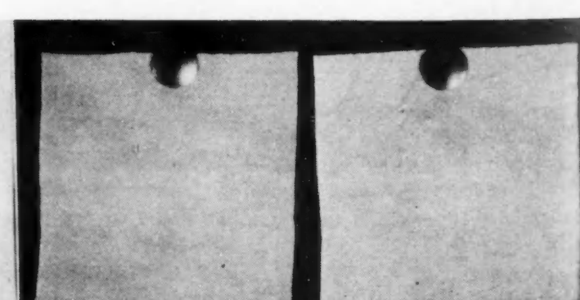
Table 19 - Effect of Additive Concentration on Engine and Crankcase Oil Deterioration in General Motors Chevrolet Tests

	Additive Concentration, % 0	0.5	1.0	2.0
Engine Condition:				
Ring Zone Demerit	2.38	1.46	0.89	0.89
Piston Skirt Demerit	1.33	1.21	0.04	0.00
Sludge Demerit	1.09	0.36	0.20	0.15
Used Oil Characteristics:				
Chloroform Insolubles, %	2.0	1.2	0.6	0.6
Chloroform Solubles (Asphaltenes), %	0.55	0.33	0.25	0.02
Neutralization Number	6.6	6.3	6.0	2.0
Saponification Number	25.6	25.6	23.8	7.2
Viscosity at 210 F, sec	68.1	....	67.4	63.8

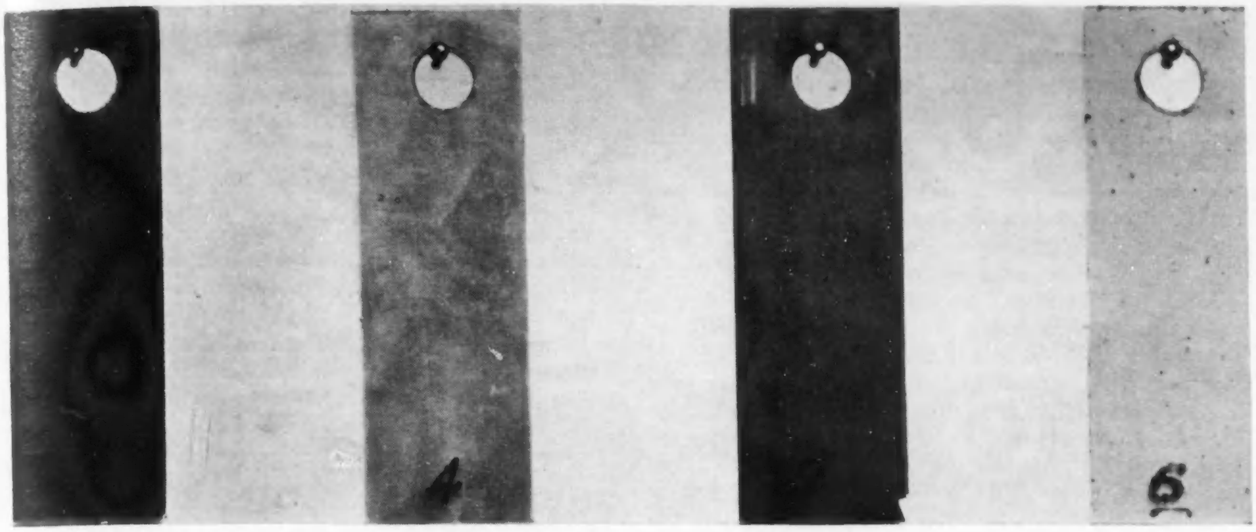

ORIGINAL UNSTAINED FLANNEL


ORIGINAL FLANNEL STAINED WITH CARBON BLACK


STAINED FLANNEL AFTER WASHING WITH CLEANERS' NAPHTHA


STAINED FLANNEL AFTER WASHING WITH CLEANERS' NAPHTHA CONTAINING DETERGENT-DISPERSER-INHIBITOR ADDITIVE

■ Fig. 10 – Launderometer tests on flannel specimens



OIL D OIL H OIL E OIL A

■ Fig. 11 – Deposition of varnish on glass microscope slides in laboratory oxidation tests

independent of dispersive action under the test conditions employed, and hence are a direct measure of the chemical stabilization effected.

It seems reasonable to assume that the same stabilizing effect which is reflected by the lessened crankcase oil deterioration should also have contributed to the greater engine cleanliness noted. The oil delivered to critical engine parts is in a cleaner condition, and its further breakdown in these zones may be retarded. In the upper ring zone, where only very small quantities of lubricant circulate and where the opportunity for detergent action in consequence appears slight, the additive may well function to prevent deposit formation and ring-sticking largely by a chemical mechanism.

Table 20 - Effect of Detergent-Disperser-Inhibitor Additive on Wear in Lauson Engine Tests

Lubricant	Oil D	Oil H	Oil D	Oil H
Cooling Water Temperature, F...	100	100	200	200
Hours on Test..................	75	75	75	75
Weight Loss of Top Ring, mg.....	84.0	56.7	45.7	44.0

Another result of the stabilizing influence of the additive is the reduced corrosion of alloy bearings, which already has been illustrated by engine tests. Laboratory studies indicate that the additive not only retards the formation of corrosive substances in the oil but, in addition, protects the bearings against attack by normally corrosive materials which enter the oil from extraneous sources.

■ Adhesion to Metal

The reduction in wear obtained with Oils A and H in engine tests characterizes the use of many detergents. It has been ascribed by Bray[12] to the greater "oiliness" of the compounded oils, and by Neely[5] to their superior "limiting adhesion temperatures." However, it is believed that certain benefits obtained with the additive can be attributed directly to the resistance of the compounded oil films to replacement by moisture and acidic combustion products which condense on engine parts in low-temperature service and in starting a cold engine. Such products initiate rusting and corrosion, which materially increase the wear of rubbing surfaces. Williams[13] has shown that, in intermittent stopping and starting operation, the corrosive effect may account for a major portion of the wear encountered.

Such wear is possible only because the usual oil films do not completely exclude the corrosive substances from making contact with the metal surfaces and, by the use of a suitable additive, highly protective oil films can be obtained. The additive in Oil A and Oil H imparts corrosion resistance, as is illustrated in Fig. 12. For this demonstration, the three steel plates shown had been exposed briefly to wet steam. That at the right, which had been coated with a film of the additive-containing Oil A, is entirely free from rust. However, the center strip, which was coated with the uncompounded Oil E, and the left strip, which had been left un-oiled, are both badly rusted.

The importance of this rustproofing action has been checked by tests run in a single-cylinder Lauson engine, with the results shown in Table 20. With a water jacket temperature of 100 F, ring wear was greater in tests on Oil D than on the additive blend, Oil H. Upon raising the jacket temperature to 200 F, and thereby eliminating the condensation of moisture, wear with Oil D was markedly reduced, and the additive in Oil H was no longer able to show as much advantage. Not only is the value of the additive in low-temperature operation indicated but, since the additive confers no benefits under the relatively mild conditions of the Lauson test at 200 F jacket temperature, it is apparent that the low-temperature effect should be ascribed more to corrosion prevention than to "oiliness," and so on.

[11] See API Proceedings, 21st Annual Meeting, Section III, Refining, pp. 63-78: "Heavy-Duty Motor Oils," by H. C. Mougey.

[12] See "Modern Diesel Lubricants for the Modern Diesel," by U. B. Bray, presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Okla., Nov. 7-8, 1940.

[13] See IAE Journal, Vol. 2, No. 10, 1934, pp. 19-34: "Further Experiments on Cylinder Wear," by C. G. Williams.

■ Operating Considerations

In the use of any lubricant, practical operating questions arise as to oil-change practice, the use of filters, and so on. Since the answers are to a large extent determined by service conditions which do not lend themselves to quantitative expression, recommendations must necessarily be indefinite. However, a discussion of the factors involved may help to clarify the problem, particularly with reference to the additive-containing oils, A and H, since little relevant information has been published on such oils.

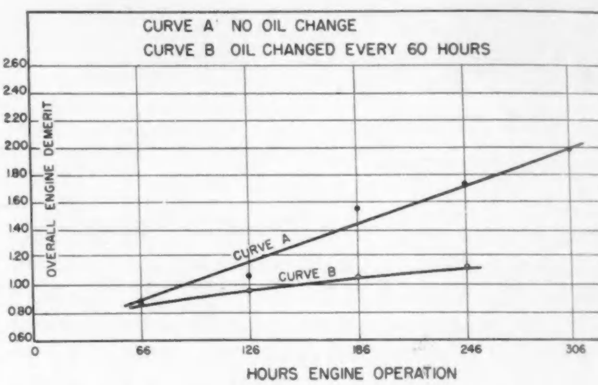
Oil-Change Practice – Conventional lubricants appear to deteriorate largely through the progressive chemical destruction of the bulk of the oil, and hence their loss in performance is quite often largely proportional to their length of service. In additive-containing oils, the deterioration of both the oil and the engine is retarded by the additive to provide the superior performance noted. However, during operation, the power of the additive may be gradually exhausted, and better results are therefore obtained if the oil is replaced regularly before the effectiveness of the additive is wholly spent. Consequently, for optimum performance a sound oil-change practice is even more important with additive blends than with conventional oils.

The benefits of changing Oil H are illustrated in Fig. 13. Curve A shows that, when Engine II was operated 306 hr without oil change, engine cleanliness demerits increased at a relatively rapid rate. However, when the oil was changed every 60 hr, as recommended by the manufacturer, accumulation of engine deposits was reduced greatly, and the superior overall performance represented by Curve B was obtained.

Make-Up Requirements – Make-up oil, used to replace that consumed, may also have an important bearing on engine performance. The make-up oil helps to replenish the supply of additive and to renew the dispersing, stabilizing, and wear-reducing properties of the lubricant. This effect has already been commented upon in connection with Fig. 13, and is illustrated rather well by the laboratory corrosion test data plotted in Fig. 14. In the laboratory test with no oil change, severe bearing corrosion set in after 100 hr but, when 20% of the oil under test was replaced with new oil every 24 hr, there was no evidence of corrosion, even after 192 hr, when the test was stopped.

These observations might make it appear at first that, the higher the rate of oil consumption, the less need there is for oil changes. However, this is not necessarily the case. If oil consumption is high as the result of the poor mechanical condition of the engine, the attendant blowby also may be high, and contamination of the crankcase oil by soot, and so on, will be severe, thus taxing the capacity of the oil to accommodate the sludge. In this event more, rather than less, frequent oil changes may be necessary.

Sometimes, however, oil consumption is higher without attendant greater blowby and crankcase oil contamination. Occasionally, consumption is increased upon changing to

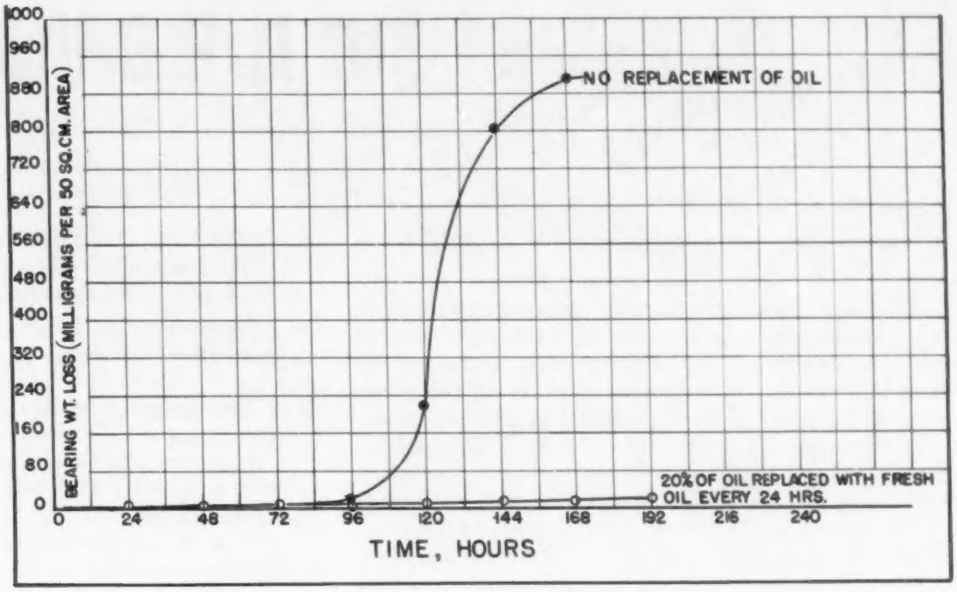


■ **Fig. 13 – Effect of oil-change practice in performance of Oil H in Engine II**

an additive-containing oil, probably because the rings are kept freer and hence let more oil pass, or because the oil wets the metal better and flows more readily to hot engine parts, which often act as a barrier to conventional oils. Such increased oil consumption is not ordinarily accompanied by an increase in blowby, as Table 21 shows, and hence it would seem possible to reduce oil-change frequency to take into account the increased oil consumption.

PLATE UNOILED — PLATE COATED WITH OIL E — PLATE COATED WITH OIL A

■ **Fig. 12 – Appearance of steel plates after steam corrosion test**



■ Fig. 14 – Reduction in copper-lead bearing corrosion by oil replacement in laboratory tests

Oil-Changeover Practice – It has been shown that when an oil like A or H is charged to a dirty engine, it may loosen sludge and varnish deposits. These loosened deposits may be very heavy and, if not removed, may become serious enough to cause trouble. Consequently, any engine changed over to a detergent blend should first be thoroughly flushed, and then, after running a nominal period on the new oil, the crankcase should be drained and recharged. Thereafter it should not require as much special attention. However, when engine deposits are being assimilated with difficulty by the blended oil, several oil changes may be necessary before the purging of the engine is completed.

Table 21 - Relationship Between Oil Consumption and Blowby in Engine I

(Comparable 84-hr Tests)

Test No.	Oil Tested	Oil Used, qt	Blowby cu ft hr
52	Oil D	9.6	93
54	Oil B (Compounded)	11.4	89
67	Oil D	7.8	135
65	Oil D+Detergent	10.8	129
72	Oil D	7.8	150
71	Oil D+Detergent-Inhibitor	8.9	142
70	Oil B (Compounded)	10.1	152

Mixing Oils – It is not ordinarily the best practice to mix the detergent-type oils with others. The addition of a detergent blend as make-up to a conventional oil may cause some cleaning action to occur. This would be beneficial to the engine, but the crankcase oil would have to be observed closely to make certain that accumulation of deposits was not sufficient to cause any difficulties. The addition of a straight mineral oil as make-up to a detergent oil, on the other hand, would tend to dilute the additive concentration, thus making it less effective. At least until more extensive field experience has been accumulated, it would also probably be best not to employ the detergent blend of one manufacturer as make-up for that of another. Although no particular incompatibility would be anticipated, it is possible that the balance between detergent, dispersive, and inhibiting properties might be altered considerably as a result of blending. These views upon mixing oils have not been checked in the laboratory or field as extensively as is desired, and are therefore offered only as precautionary suggestions.

Oil Filters – When operating an engine on a conventional oil, an effective filter may serve to remove a major portion of the contamination and deterioration products from the oil, and thereby delay the need for oil changes. Eventually the filter cartridge becomes choked with solids, and must be replaced. Some operators therefore change filters whenever the oil begins to appear fairly dirty, or when some definite limit of contamination is reached. Filter replacement schedules based on the oil condition require revision when applied to lubricants containing dispersers. Blends like A and H tend to carry finer contamination products through the filter, and consequently the apparent condition of the oil ceases to be a direct measure of the filter efficiency.

Strainer-type filters are of particular interest for use with detergent-containing oils. Since, as the power of the additive is spent, it loses its ability to keep such filters clean, some indication of the need for an oil change can be obtained by a quick examination of the filter unit. Once sludge starts to accumulate on the strainer, the oil should be changed. However, the additive may become spent in other ways, as in corrosion-preventive ability, and hence a clean filter is not a complete index of the oil condition.

It is obvious from these few observations that no blanket recommendation will provide optimum lubrication practice in all engines and in all services. What may be an excellent practice for one operator may not be satisfactory for another whose equipment, type of operation, and so on, are all different. In general, the engine manufacturers' oil-change schedules should be followed, and in no case should an additive blend be changed any less often than a conventional oil, until the operator has become quite familiar with the new product.

It should be emphasized that these improved oils provide better lubrication and keep engines in a better mechanical condition than conventional oils; thus, they help to keep equipment operating more efficiently, minimize costly and annoying breakdowns, and reduce maintenance expense.

COMPOUNDING —

THE 2-cyl "compound" engine on which the data reported in this paper were obtained is provided with a high-pressure cylinder, and a low-pressure cylinder which serves as a compressor as well as a working cylinder.

The primary consideration in this work was the extracting of additional work from the already partially expanded gases from the high-pressure cylinder; a secondary consideration was the use of the same low-pressure cylinder as a second-stage compressor to aid in supercharging the high-pressure cylinder.

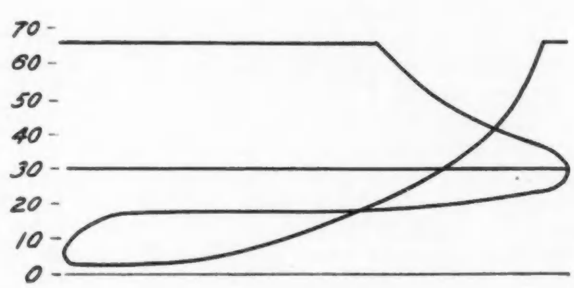
In reporting test results covering the development and testing program on the 2-cyl test unit, indicator diagrams showing the power developed in both high-pressure and low-pressure cylinders are presented. The method of summarizing the test results is by tabulating the representative runs and plotting the final runs. Some results and conclusions follow:

COMPOUNDING of internal-combustion engines is, by no means, new. Many attempts have been made to reclaim a portion of the energy wasted in the exhaust either by extracting energy in the form of heat or work. These attempts are, of course, inspired by the gains in efficiency realized with double-and triple-expansion steam engines. Most of the proposed internal-combustion compound engines utilize the low-pressure piston only for expansion. No attempt is made in this paper to review the history of such development. The engine on which the data reported in this paper were obtained was one in which the low-pressure cylinder served as a compressor as well as a working cylinder. The primary consideration was, of course, the extracting of additional work from the already partially expanded exhaust gases from the high-pressure cylinder. A secondary consideration was the use of the same low-pressure piston as a second-stage compressor to aid in supercharging the high-pressure cylinder.

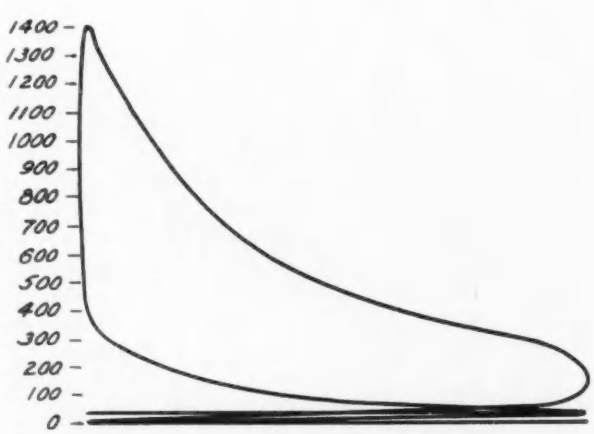
[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 13, 1941.]

■ History

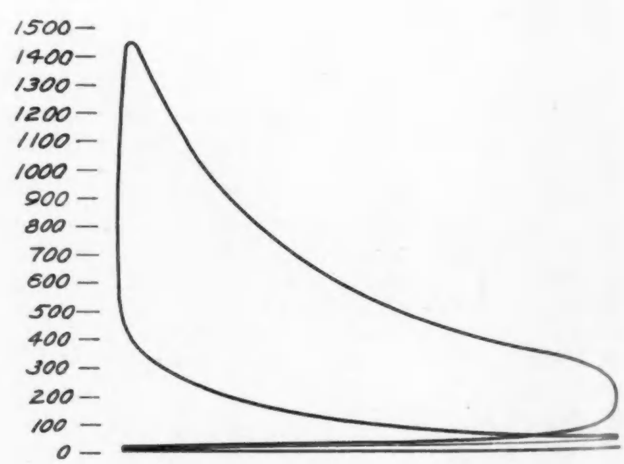
The high-output tests conducted on a 1-cyl engine several years ago by the author, and mentioned in *Mechanical Engineering,* March, 1936, were used as the basis for the



HIGH-OUTPUT CYLINDER LOWER LOOP DIAGRAM



HIGH-OUTPUT CYLINDER INDICATOR DIAGRAM



HIGH-OUTPUT CYLINDER INDICATOR DIAGRAM

■ Fig. 1 (left) and Fig. 2 (right) – Indicator diagrams – High-output cylinder

Facts and Fallacies

1. An indicated mep of 705 psi was recorded in the high-pressure cylinder, but the corresponding imep in the low-pressure cylinder was only 63 psi.

2. Fuels are available which make possible mean effective pressures in the high-pressure cylinder heretofore considered fantastic, resulting in release pressures to the low-pressure cylinder of 350 to 400 psi.

3. Two-stage expansion to a final ratio of 15:1 or more is feasible.

4. Modern internally cooled valves are satisfactory even under the unusual temperatures and pressure conditions encountered by the high-pressure exhaust valve.

5. Compounding the expansion results in a definite increase in thermal efficiency.

Although the results of the test are not as good as desired, Mr. Prescott concludes, it is considered that the project is worthy of further development and research.

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design of the 2-cyl compound engine. Inspection of the indicator cards, Figs. 1 and 2, shows that pressures of the order of 400 psi are available at the point of exhaust opening when the cylinder is operated at an output of 500 to 600 psi in imep. Obviously such pressures released to an efficient low-pressure cylinder should develop a very acceptable power increase at no added cost for fuel. In addition, the preceding up-stroke of the low pressure piston is adaptable to use as a final stage to compress the air required to supercharge the high-pressure cylinder. To this end a normal four-stroke high-output cycle was used for the high-pressure cylinder, air being introduced from a receiver supercharged by the low-pressure cylinder, and the exhaust being discharged directly into the low-pressure cylinder. Figs. 3, 4 and 5 show the arrangement of the test engine. Fig. 6 is a schematic diagram of the set-up. The low-pressure cylinder was equipped with an air-discharge valve, an air-intake valve, and an exhaust valve.

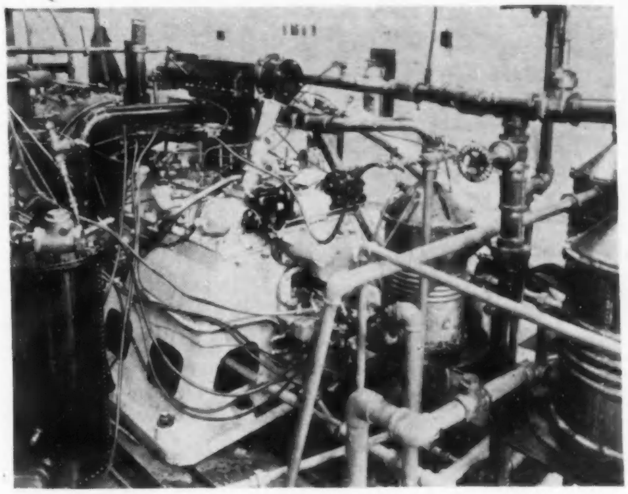


Fig. 4

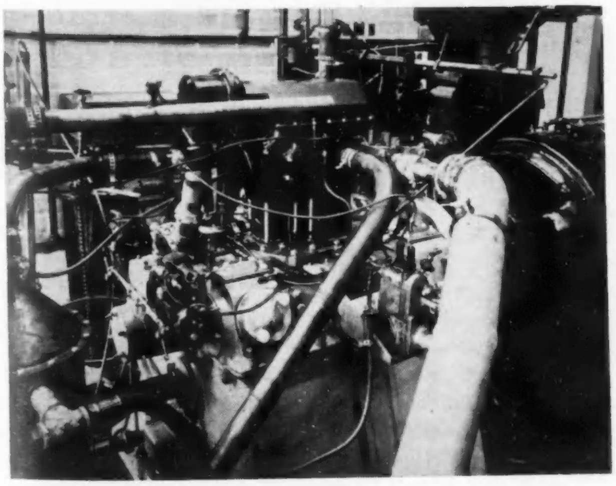


Fig. 3

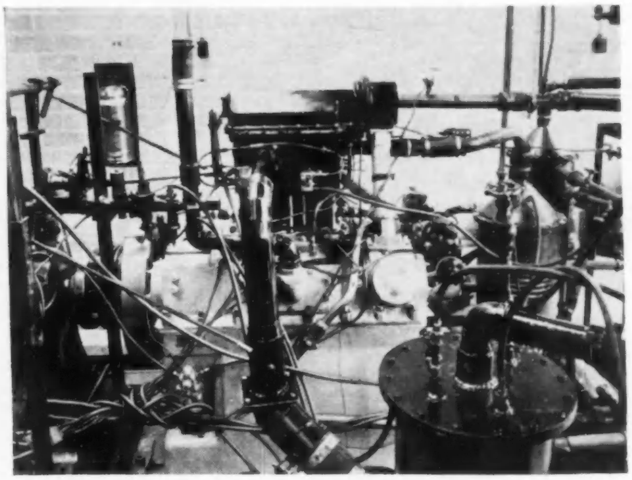


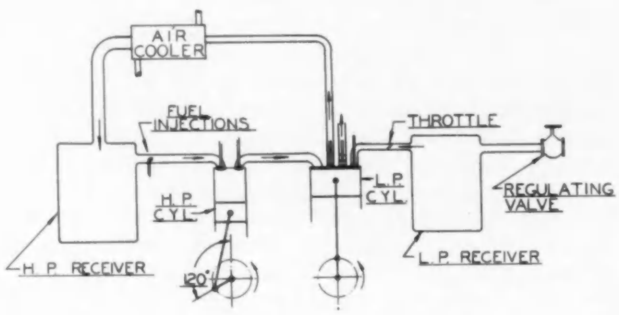
Fig. 5

■ Figs. 3, 4, and 5 – Three views of compound test engine

A port conducts the high-pressure exhaust to the low-pressure cylinder. The cycle is so arranged that the low-pressure piston is at top-center position when the high-pressure exhaust valve opens.

The events of the low-pressure cycle are as follows:

1. Intake, air only
2. Compression and delivery to air receiver
3. Expansion of exhaust from H.P. cylinder
4. Exhaust to atmosphere



■ Fig. 6 – Schematic diagram of compound engine system

The events in the high-pressure cylinder are as follows:

1. Intake of air from receiver with fuel added in intake pipe by an injector
2. Compression
3. Ignition and combustion
4. Expansion
5. Exhaust

The time of opening of the H.P. exhaust valve is 69 deg BBC. The H.P. crank leads the L.P. crank by 120 deg; hence, the H.P. exhaust opens 9 deg BTC in the L.P. cylinder. This event establishes the relation of the cycles in the two cylinders, and was arranged in this manner to prevent loss of energy by immediately expanding the exhaust gases in the L.P. cylinder. During the 69 deg following the opening of the H.P. exhaust the H.P. piston moves to bottom-center position. During the same period and the following 58 deg, the H.P. exhaust gases are expanded by the L.P. piston to the opening of the L.P. exhaust valve. Both H.P. and L.P. exhaust valves remain open for 67 deg, when the H.P. exhaust valve closes. The L.P. exhaust valve then remains open until 9 deg ATC.

Table I shows the valve timing in the H.P. and L.P. cylinders. This timing gives a normal high-output card such as shown in Figs. 7 or 8. Figs. 7 and 8 also show the cards obtained in the L.P. cylinder with this timing. It was expected that an imep of 500 psi would be obtained in the H.P. cylinder and 120 psi in the L.P. cylinder. The latter originally had 4 times the volume of the high-pressure cylinder, but this was later reduced to 3 volumes to reduce the friction H.P. in the engine. At 120 psi imep and 3 volumes, the equivalent imep of the L.P. cylinder would be 360 psi, referred to the H.P. cylinder.

It was easily possible to secure very high outputs from the H.P. cylinder, as the H.P. card, Fig. 8, shows. This card probably represents a world record, showing 705 psi imep. Incidentally, of course, a record for the pressure indicator also may be claimed. The corresponding L.P. cylinder indicator card, however, is most disappointing. It shows an imep of 63 psi, corresponding to 189 psi in the H.P. cylinder. The indicated output is 92 ihp in the H.P. cylinder and 25.5 ihp in the L.P. The total power is 117.5 ihp. The corresponding brake output is 96.8 bhp; consequently, the mechanical efficiency should be 82.5%. This value does not agree with the friction runs which indicate a mechanical efficiency of 62.3%. This discrepancy is probably due to the difference in pumping losses when firing and motoring. The disagreement was even worse before reducing the stroke of the L.P. cylinder to reduce engine friction.

The high- and low-pressure cylinders were cast in a 2-cyl block. The H.P. cylinder was 3⅜-in. bore x 5-in. stroke, while the L.P. cylinder was 6⅜-in. bore x 6¼-in. stroke. The displacement of the L.P. cylinder therefore was 199 cu in. or 3.84 times the displacement of the H.P. cylinder. The expansion ratio of the combination therefore was 4.84 times that of the H.P. cylinder. In this case it was 4.84 x 5 or 24.2:1. The air cycle efficiency for this expansion ratio is 70.1% as compared with 47.6% for the H.P. cylinder alone with 5:1 expansion ratio. It was known from previous tests that the specific fuel consumption of the H.P. cylinder alone could be as low as 0.5 lb per bhp-hr. Hence, it was reasoned that the fuel consumption of the compound engine should be of the order of $\frac{0.5 \times 47.6}{70.1}$ or 0.34 lb per bhp-hr. This would be at "maximum-power" setting, and gave grounds for the hope of a cruising specific consumption of about 0.3 lb per bhp-hr.

Table 1 – Final Valve Timing of Compound Test Engine

Valve Timing, High-Pressure Cylinder

Intake opens	33 deg BTC	Intake closes	53 deg ABC
Exhaust opens	69 deg BBC	Exhaust closes	19 deg ATC
Intake period	266 deg	Exhaust period	268 deg

Valve Timing, Low-Pressure Cylinder

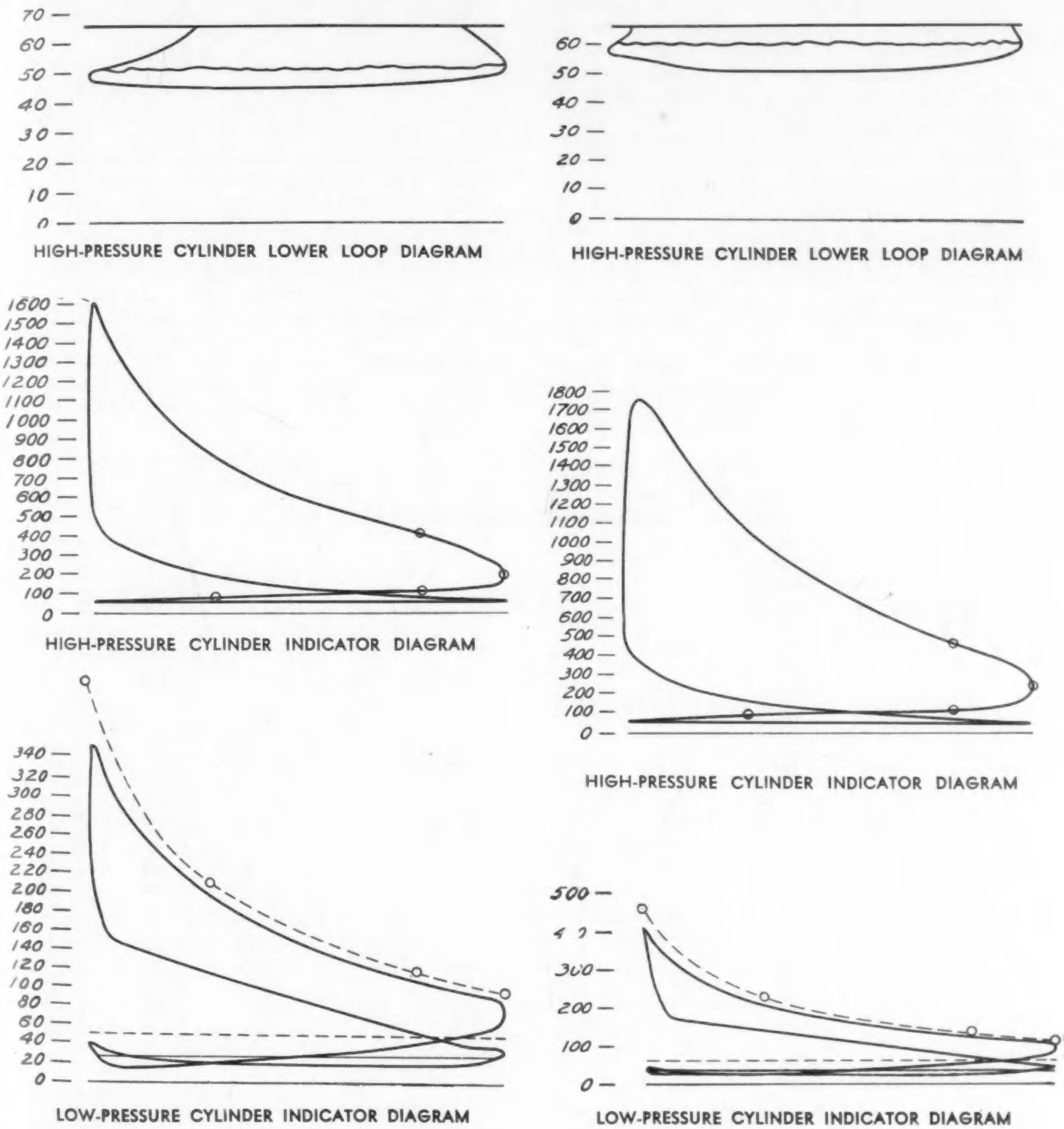
Intake opens	10 deg BTC	Intake closes	58 deg ABC
Discharge opens	22 deg ABC	Discharge closes	8 deg ATC
Transfer opens	9 deg BTC	Transfer closes	79 deg ABC
Exhaust opens	62 deg BBC	Exhaust closes	9 deg ATC
Intake period	248 deg	Discharge period	166 deg
Transfer period	268 deg	Exhaust period	251 deg

Crank interval 120 deg, with L.P. crank following H.P. crank.

When the engine was first operated, the air discharge valve in the L.P. cylinder was entirely inadequate. It had a port diameter of 1⅞ in. and a lift of only ⅛ in. The timing also was in error and, upon running the engine in, it was found to deliver only 23.9 bhp at 2000 rpm unsupercharged, that is, with only the receiver pressure developed by the L.P. cylinder as shown in A, Table 2. The fuel consumption was 21.9 lb per hr, giving a brake specific fuel consumption of 0.917 lb per bhp-hr. It was then discovered by means of indicator diagrams in the L.P. cylinder, that the air delivery valve was entirely too small. A new camshaft was made increasing the delivery-valve lift to ¼ in. and increasing the length of open period from 85 to 135 deg. At this time it also was found that the displacement of the L.P. cylinder was too great for the

attainable valve areas; consequently the stroke of the L.P. piston was reduced to 5 in. The immediate improvement is shown by the data in Run B, Table 2. Runs C and D, Table 2, were made at 5 and 7.4 psi supercharge to the L.P. cylinder. It was noted that the receiver pressure increased by only about the amount of the supercharge, showing that the air discharge valve was still too small. It was not possible to improve this condition except by building a new cylinder and valve gear; consequently the remainder of the tests suffer by excessive pumping losses.

In order to minimize the pumping losses it was decided to try what, for lack of a better term, was called parallel supercharging. In runs E and F, the inlet to the L.P. cylinder was open to atmospheric pressure, and the air pumped was discharged into the receiver to which additional supercharging air was supplied externally. This arrangement gave the lowest brake specific fuel consumption recorded, doubtless because of decreased pumping loss in the L.P. cylinder. It is also worthy to note that the indicated specific fuel consumption in run E is the lowest



■ Fig. 7 (left) and Fig. 8 (right) – High-pressure and low-pressure indicator cards obtained with valve timing shown in Table I

recorded, except for two readings of about 0.25 lb per ihp-hr which were not included because it was thought that an error may have been made in the original data.

In order to establish a standard by which to judge the performance, the L.P. connecting rod was removed and the piston blocked by a fixed support. Thus the H.P. cylinder could be operated as a high output single-cylinder test engine. Runs G to L were made in this manner. The indicated specific fuel consumption is approximately 50% higher than in corresponding runs with double expansion, but the brake specific consumptions remain about the same.

The final series of runs, M to R, was made after the defects in the system were remedied as far as possible without building a new engine. In these runs, the inlet air pressure was increased successively until all available facilities for supercharging were utilized. The brake specific fuel consumption is seen to decrease, in general with increased output, because the friction increases at a lower rate than the indicated output. Even at the highest output recorded there was no evidence that the limit of the fuel had been reached. Fig. 9 shows a graph of the final runs made.

The general operating conditions were: water cooling, 180 F, out temperature; 120-sec aircraft engine oil, 185 F in temperature; fuel, P.P.F. No. 628, especially for high output rated as octane plus 0.8 cc tetraethyl lead by the Aviation Method, but with a much better rating by an experimental supercharged engine method. Water injection or other detonation control was not required beyond the properties inherent in the fuel. The air entering the H.P. cylinder was kept at approximately 80 F by means of a water-cooled tubular cooler, the water surrounding the tubes and the air passing through the tubes.

■ Discussion of Data

Table 3 was compiled in order to study the combined effect of the two pistons during the expansion stroke. Column 3 shows the displacement volume of the H.P. piston for 300 deg from top-center firing stroke. Column 6 shows the corresponding L.P. displacement beginning at 120 deg from top-center in the H.P. cylinder. Column 7 was obtained by adding Column 3 and Column 6, while the displacement volume has been added to give Column 8. The maximum volume occurs at 284 deg and the expansion ratio is $\frac{194}{12.9} = 15.05$. The values in Column 8 are a fair approximation to a harmonic curve, considering the different sizes of the two cylinders and the out-of-phase relation of the cranks.

Table 2 presents test data covering several months of running under various conditions of operation. The 4th column presents the supercharge pressure in psi at the low-pressure receiver, while Column 5 gives the high-pressure receiver pressure. Column 6 gives values of the friction H.P. from motoring tests, with three values extrapolated from incomplete data. Columns 9 and 10 give the fuel consumption in lb per bhp-hr and also lb per ihp-hr. The values of specific fuel consumption are quite disappointing but, of course, a multicylinder engine would give better brake specific consumptions than those shown for the 2-cyl test units.

Table 2 – Compound Engine Test Data

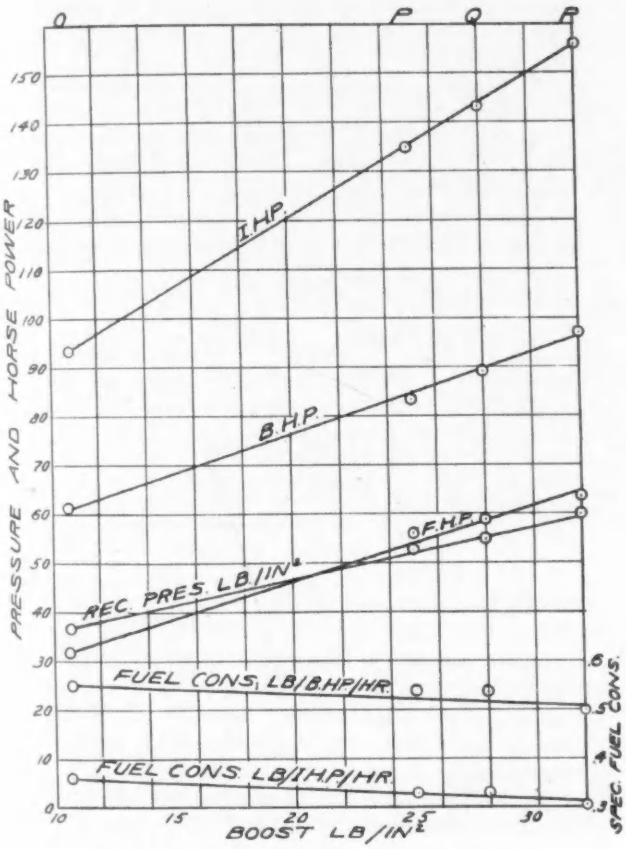
	Run	RPM	BHP	Boost, psi	Receiver, psi	FHP	IHP	Fuel Consumption lb/hr	Brake Specific Fuel Consumption lb/bhp/hr	Indicated Specific Fuel Consumption lb/ihp/hr
5.1 piston	A	2040	23.9	0	18.0	36.0	59.9	21.9	0.917	0.608
	B	2025	35.4	0	16.8	26.0	61.4	19.8	0.560	0.323
	C	2025	41.2	5	21.5	27.0	68.2	22.7	0.552	0.334
	D	2000	45.6	7.4	24.3	26.0	71.6	22.7	0.498	0.317
	E	1910	49.3	0	25.0	23.8	73.1	22.22	0.451	0.304
	F	1805	61.3	0	28.3	20.0**	81.3	28.3	0.461	0.348
	G	2525	51.3		20.0	10.9	62.2	31.3	0.609	0.503
	H	1985	41.7	1-cyl H.O.	20.0	6.34	48.04	22.25	0.533	0.462
	I	1895	48.0	with L.P.	30.0	5.9	53.9	23.8	0.496	0.441
	J	2215	61.2	piston	30.0	7.7	68.9	30.7	0.502	0.446
	K	2500	68.3	blocked	30.0	7.7	76.0	40.5	0.593	0.533
	L	1905	56.3		30.2	5.9	62.2	30.15	0.536	0.486
4.1 piston	M	2518	28.5	0	16.0	38.8	67.3	21.25	0.746	0.315
	N	1805	47.8	5.0	25.0	21.9	69.7	25.2	0.527	0.361
	O	2005	61.2	10.7	37.0	32.0	93.2	33.8	0.551	0.362
	P	2010	83.1	25.0	52.7	51.0**	134.1	44.8	0.539	0.334
	Q	2010	89.2	28.0	55.0	54.0	143.2	48.3	0.542	0.337
	R	2030	96.8	32.0	60.0	58.75**	155.5	48.2	0.498	0.310

A – 3⅝ x 5 and 6¾ x 6¼; B to F and M to R – 3⅝ x 5 and 6¾ x 5; E, F – parallel supercharging; G to L – 3⅝ x 5 cyl only; A to L – 5:1 C.R. in H.P. cyl; M to R – 4:1 C.R. in H.P. cyl.
**** Extrapolated from incomplete curves.**

Table 1 gives the final valve timing of the engine, and appears to be the best that can be done without redesigning the engine.

Fig. 1 shows a normal pressure indicator diagram on a 4⅞ x 7-in. 1-cyl engine at a manifold pressure of 30 psi gage, giving 494 psi bmep. The weak spring diagram shows an appreciable reduction in pumping losses due to the intake pressure exceeding the exhaust pressure during the greater portion of the pumping loop. Fig. 2 shows a similar card, at 539 psi bmep. These cards were the basis of design of the compound engine.

Fig. 7 shows a complete indicator analysis of the compound engine. The pumping loop no longer resembles that in Fig. 1 but the working portion of the card is quite normal. The lower card is from the L.P. cylinder, and



■ Fig. 9 – Plotted results of final runs on compound test engine

has superposed on it the corresponding H.P. cylinder pressure (see upper dotted curve), and the H.P. receiver pressure (lower dotted line). The compressor loss is clearly excessive, showing a pressure drop through the air discharge valve of approximately 100 psi maximum. The lower solid lines indicate the atmospheric and L.P. boost pressures.

Fig. 8 is similar to Fig. 7 but shows the highest recorded imep, together with the corresponding points replotted on the L.P. diagram. The H.P. card shows 705 psi imep, which is assumed to be a record value, since no reports are available that such an mep ever has been attained before.

Table 3 – Combined Effect of Two Pistons During Expansion Stroke

H.P. Piston			L.P. Piston			Combined	
Crank Deg	Piston Travel, %	Displ., cu in.	Crank Deg	Piston Travel, %	Displ., cu in.	Displ., cu in. (total)	Total Volume Incl. Clear, cu in.
0	0	0	....	....	....	0	12.9
20	3.7	1.9	....	....	....	1.9	14.8
40	14.3	7.4	....	....	....	7.4	20.3
60	29.7	15.3	....	....	....	15.3	28.2
80	47.5	24.5	....	....	....	24.5	37.4
100	64.8	33.5	....	....	....	33.5	46.4
120	79.7	41.2	0	0	0	41.2	54.1
140	90.9	46.8	20	3.7	5.9	52.7	65.6
160	97.7	50.4	40	14.3	22.8	73.2	86.1
180	100.0	51.6	60	29.7	47.5	99.1	112.0
200	97.7	50.4	80	47.5	76.0	126.4	139.3
220	90.9	46.8	100	64.8	103.5	150.3	163.2
240	79.7	41.2	120	79.7	126.5	167.7	180.6
260	64.8	33.5	140	90.9	145.5	179.0	191.9
280	47.5	24.5	160	97.7	156.5	181.0	193.9
284	....	....	....	....	....	....	194.0
300	29.7	15.3	180	100.0	160.0	175.3	188.2

$$\frac{194.0}{12.9} = 15.05 \text{ max. value of expansion ratio.}$$

The fuel used would permit somewhat higher output, but the limit of the supercharging facilities was reached at this condition, and the tests were terminated.

Fig. 9 shows a plot of the data under Runs O, P, Q, and R, of Table 2. The high value of friction horsepower is shown; also, the large difference between brake and indicated specific fuel consumptions. It is observed that both curves show decreasing values with increased power at constant rpm.

■ Conclusions

1. Fuels are available which make possible mean effective pressures in the H.P. cylinder heretofore considered fantastic, resulting in release pressures to the L.P. cylinder, of 350 to 400 psi.

2. Two-stage expansion to a final ratio of 15:1 or more is feasible.

3. The pressure and energy drop through the H.P. exhaust valve is not serious.

4. Modern internally cooled valves are satisfactory even under the unusual temperature and pressure conditions encountered by the H.P. exhaust valve.

5. Compounding the expansion results in a definite increase in thermal efficiency.

6. Air compression by a high-speed compressor piston is not efficient because of cylinder-wall friction and limited discharge-valve area. This results in low overall mechanical efficiency.

7. It is useless to expand the gases to a large volume because the friction and pumping losses in the L.P. cylinder more than compensate for the increase in thermal efficiency expected from the increased expansion ratio.

The Present Status of

A NUMBER of the more important contributions made recently to our knowledge of the process of combustion in the engine cylinder are reviewed briefly in this paper.

Of the possible lines of attack which promise to lead to further improvements in the control of gaseous combustion as a source of power, the following are stressed:

1. Finding new mixtures which are inherently more powerful or more economical;

2. Finding new methods of altering the mass rate of burning, the completeness of combustion and, hence also, the rate of increase in pressure;

3. Preventing preignition in spark-ignition engines and facilitating ignition in compression-ignition engines;

4. Suppressing detonation.

The need for an increasing amount of combustion research and the desirability for careful coordination are demonstrated by an examination of progress to date.

A BRIEF discussion of combustion research, which has been in progress for over three hundred years, must be limited to a specific subdivision of the field. For presentation to the Society of Automotive Engineers, it is logical that this report should be confined to that phase of combustion research which is concerned with explosions in gases, and particularly with explosions from which, through the medium of the internal-combustion engine, usable power may be derived.

From the standpoint of the automotive engineer, the practical objective of research on gaseous combustion will always be the improvement of the combustion process as a source of power in engines. The improvements that have been made heretofore in our control and utilization of this process for doing work are so great that the life of nearly every individual in the country has been profoundly affected. Yet it may be truthfully said that very little is actually known of the highly complex mechanism, on a molecular scale, by which the chemical energy latent in a fuel is converted into mechanical force upon a piston.

Despite the fact that combustion research has been inspired and its course directed by practical needs, the immediate objectives of individual studies are seldom highly practical. These studies, however, are responsible for the gradual accumulation of isolated bits of information that will some day lead to the evolution of a comprehensive picture of the explosion process. Since the advances in engine design and fuel technology have been so great, the question naturally arises as to whether a more complete understanding of gaseous combustion can lead to additional progress. That this question can be answered in the affirmative with certainty is at once evident from even a superficial examination of the gaps in our present knowledge of the combustion process and of the apparent possibilities for improving its usefulness as a source of power.

Both the National Advisory Committee for Aeronautics and the National Bureau of Standards have shown their belief in the need for, and their faith in the success of, combustion research by individual and cooperative support of a continuing program of study. Inclination to this same view by industrial and university laboratories is evidenced by the active participation of some of these groups in studies of a similar nature. There is room in the field for all of these, and many more, without duplication of effort, provided that the activities are properly coordinated.

The outstanding problems of combustion control are so varied and so complex that the need for coordination is great indeed. In fact it seems that the only hope of maintaining the rate of progress of the past three or four decades lies in a carefully planned and comprehensive program of study, executed on a national scale, yet having a flexibility that is adequate to provide for immediate attacks upon new and promising issues when these become apparent.

Under present circumstances, where practical accomplishments in a field have far outdistanced the development of the theoretical aspects, it is perhaps worth while to pause at times for a general consideration of what has been accomplished, what is being done, and what future trends are apparent. Such an attempt is made in this report.

General

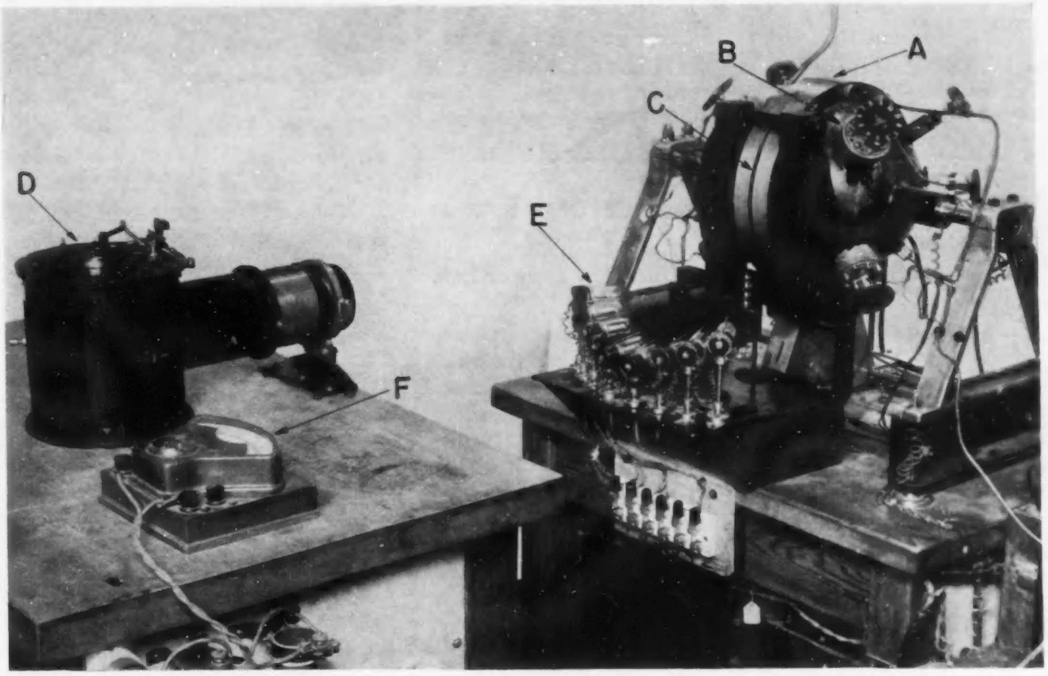
The indicator diagram is the recognized criterion for evaluating the power and efficiency of the explosion and it reflects also the degree of detonation and explosion shock. It is therefore logical, in endeavoring to improve the combustion process, to decide first what improvements are possible in the indicator card, and then to seek means of obtaining these characteristics through combustion control.

In the simple theory of the internal-combustion engine cycle, power and efficiency are a maximum for instantaneous and complete combustion at top dead-center, power is proportional to the rise in pressure at constant volume for a given compression ratio, and both power and efficiency increase as the amount of expansion upon burning is raised.

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 13, 1941.]

COMBUSTION RESEARCH

by ERNEST F. FIOCK
National Bureau of Standards



General view of the spherical bomb and accessories *A*, the bomb; *B*, a pressure indicator; *C*, the window slit; *D*, the camera; *E*, the neon lamps; and *F*, the electric tachometer

Application of these principles to the improvement of actual engines requires that higher compression ratios be made practical, that more powerful or efficient mixtures be developed, and that better control of the rate of rise in pressure during combustion be achieved.

The power and efficiency of spark-ignition engines do not increase as rapidly with compression ratio as do the explosion pressures, and there will be some value of compression ratio beyond which improvements in performance will be too small to justify the added cost and weight required by the higher stresses. However, before this point is reached, the useful compression ratio is limited by the tendency of the more plentiful fuels to detonate or preignite. Thus, one of the immediate problems of combustion control is to find means to suppress detonation and preignition in spark-ignition engines at higher and higher compression ratios.

In compression-ignition engines, the problem is not to prevent preignition but to accomplish ignition more readily. Difficulties are experienced in preventing an accumulation of fuel which suddenly burns to produce excessive pressures. For this type of engine the objectives are to facilitate ignition and to control the rate at which pressure is developed. Experience has shown that, if this rate is excessive, rough and noisy operation will be obtained with both types of engine. The optimum rate of rise is therefore the maximum rate consistent with smooth operation and reasonable peak pressures, and it seems probable that the actual rate in modern engines operating without detonation is still below the optimum.

For maximum power the fuel and charge proportions should be selected to develop the greatest pressure possible. Where economy is the primary requirement, the charge may have a very different composition. In either case both the quantity of fuel that remains unburned, or partially burned, and the heat losses should be reduced to a minimum.

For the purposes of the present discussion, it is convenient to make a more specific listing of the aforementioned general opportunities for improving combustion control. It has long been realized that practical improvement, which can be realized only through combustion research, is possible in the following ways:

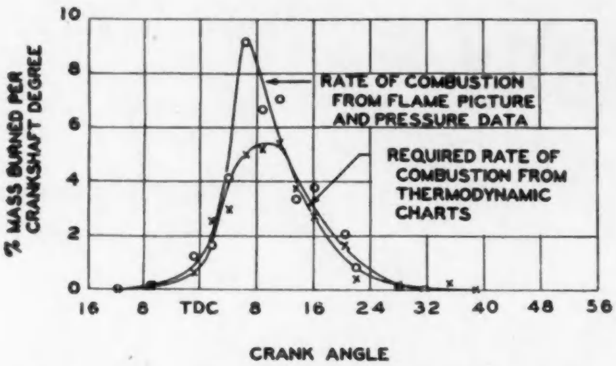
(1) By finding new mixtures which are inherently more powerful or more economical;

(2) By finding new methods of altering the mass rate of burning, the completeness of combustion, and hence also the rate of increase in pressure;

(3) By preventing preignition in spark-ignition engines and facilitating ignition in compression-ignition engines; and

(4) By suppressing detonation.

Although these subdivisions are not independent, one of another, they provide a logical basis for further discussion.



■ Fig. 1 – Typical curves from General Motors Laboratories, showing that the charge is consumed more rapidly during most of the power stroke than would be predicted from present thermodynamic charts

■ Past Results, Present Efforts and Future Trends

(1) New Mixtures – New and improved mixtures seem possible through the discovery of either new fuels or new additives.

If a new fuel is to be superior to those commonly available, it must have a higher heating value; or it must burn to give a greater number of product molecules; or it must burn more completely, leaving less combustible material such as CO in the exhaust gas; or it must permit the use of higher compression ratios without detonating. Recently there has been a notable increase in the intensity of the search for fuels which might have some of the foregoing characteristics. The present emphasis seems to be very strongly on the evolution of fuels of higher octane number, but some of the other possibilities may also be realized eventually. The American Petroleum Institute, various oil companies, the National Advisory Committee, and the military services are cooperating in the search for new and better fuels by sponsoring research at the National Bureau of Standards and at various universities. The importance of the results that may be forthcoming can scarcely be predicted at this stage of the project.

Research has shown that certain additives to motor fuels have a very great practical value in suppressing detonation. However, although the effects produced by tetraethyl lead are familiar to all, the mechanism by which the desired results are achieved is still a matter of conjecture. Who would be so bold as to deny the possibility that a full knowledge of this mechanism might aid in the development of other substances which would be more effective or cheaper? This then is one highly practical aspect of the more general aim of combustion research to discover the molecular mechanism of the burning process.

There are many other ways by which factors related to mixture composition might be controlled in the interest of greater power output or better economy of a given engine. Not a great deal of research effort seems to have been expended upon them so far, and about all that can be done at present is to pose a few questions for the dual purpose of stimulating thought and indicating some possible trends of future study.

Is it feasible to introduce additional oxygen into the cylinder, perhaps in combined form as solid or liquid, as a means of burning more fuel per explosion?

Fuel-oxygen ratios for maximum power are generally quite different from those for maximum economy. What are the causes of this difference and are any of them subject to control?

Radiation is responsible for a considerable fraction of the heat loss from the burned and burning gases, particularly during detonation. Can materials be found which will either absorb or prevent the emission of a portion of this radiation, thus retaining more energy in the charge for doing useful work?

While on the subject of radiation it may be well to point out that some studies of the emission spectra of flames and of the absorption spectra of the unburned gas within the engine cylinder have already been made in the General Motors laboratories[1, 2, 3], and similar studies have been made at the National Bureau of Standards at the request of the Navy Department, but these latter have not been publicly described. Through photographs of the spectra, certain intermediate products of the oxidation of the hydrocarbon fuels have been positively identified. Although it does not seem that any of the intermediates identified so far are of controlling importance in the burning process, there is some reason to believe that others might be more effective. Hence it is felt that the spectroscopic examination of engine flames is still a promising field for future study, even though no such experiments are known to be in progress at the present time.

As to thermal properties, the heats of combustion of the common fuels, the heat capacities of the product gases at high temperatures, and equilibrium data on the chemical reactions now known to be involved, are fairly well established. In fact, sufficient thermodynamic data are now available to permit the compilation of charts, similar in general nature and purpose to the more familiar Mollier diagram for steam, but applicable to explosions in the engine cylinder. Charts of this sort, prepared at the Massachusetts Institute of Technology, are described in References 4 and 5. The accuracy and usefulness of such charts will both improve, just as steam tables and diagrams have improved, as better basic data become available and as a more complete understanding of the explosion process is accumulated.

Recently an attempt has been made at General Motors[6] to correlate their observed values of flame volume with observed pressures, through the use of such charts. Fig. 1 shows a typical result of this attempt, indicating that, in all cases tried, it was necessary to burn a greater fraction of the total charge to produce a given increase in pressure than that called for by the charts. The deficiency is not inconsiderable, but "at the end of the inflammation process is about three times higher than the generally accepted value of 6% of the heat liberated by combustion." The challenge to combustion research to find a practical method of preventing this waste will be mentioned again in a later section.

A few years ago many explosion studies had as their primary goal the determination of the mean heat capacity

[1] See *Industrial and Engineering Chemistry,* August, 1933, pp. 923-931: "Absorption Spectra of Gaseous Charges in a Gasoline Engine," by Lloyd Withrow and Gerald M. Rassweiler.

[2] See *Industrial and Engineering Chemistry,* December, 1933, pp. 1359-1366: "Spectrographic Detection of Formaldehyde in an Engine Prior to Knock," by Gerald M. Rassweiler and Lloyd Withrow.

[3] See *Industrial and Engineering Chemistry,* December, 1934, pp. 1256-1261: "Formaldehyde Formation by Preflame Reactions in an Engine," by Lloyd Withrow and Gerald M. Rassweiler.

[4] See SAE Transactions, October, 1936, pp. 409-424: "Thermodynamic Properties of the Working Fluid in Internal-Combustion Engines," by R. L. Hershey, J. E. Eberhardt and H. C. Hottel.

[5] See *Chemical Reviews,* Vol. 21, March, 1937, pp. 438-460: "A Mollier Diagram for the Internal-Combustion Engine," by H. C. Hottel and J. E. Eberhardt.

[6] See SAE Transactions, December, 1940, pp. 526-545: "Effectiveness of the Burning Process in Non-Knocking Engine Explosions," by Lloyd Withrow and Walter Cornelius.

of some constituent of the mixture between the initial and final temperatures. Recently, however, more accurate values of heat capacity have been derived from spectroscopic data, and the usefulness of the explosion studies for providing basic information as to the burning process itself has thereby been greatly enhanced. This instance, at the same time, provides a striking example of the need for flexibility in any program of combustion research.

The methods of spectroscopy have also been employed in the measurement of temperatures of both stationary and explosion flames. In any explosion in a closed container, that portion of the charge which burns first is subsequently rendered much hotter by the compression resulting from burning the remainder of the charge. Thus, at the instant when burning is complete, the gas in the general region of the spark plug is several hundred degrees hotter than the gas that has just been inflamed. This non-uniformity of temperature is accompanied by a non-uniformity of composition, since the chemical equilibria change with temperature, and both of these factors contribute greatly to the complexity of any theoretical treatment.

Some measurements of the temperatures prevailing in particular zones of the combustion chamber have been made at Langley Field[7] and in the research laboratories of the General Motors Corp.[8] More general surveys, covering the entire combustion chamber, would doubtless be useful in studying the completeness of the reaction and in correlating flame and pressure records for the same chamber, but the surveys now available are too limited in scope for these purposes.

(2) *Control of the Mass Rate of Burning* – The second problem of combustion control listed previously is to attain the maximum rate of pressure rise consistent with smooth operation. The rate of pressure rise at any instant during combustion is nearly proportional to the mass rate of burning of the charge.

In a spark-ignition engine, where flame originates at the spark plug and spreads throughout the combustion chamber, the mass rate of burning at any instant is the product of the instantaneous area of the flame front, its velocity of advance into the unburned mixture and the density of this mixture. The density of the charge is not an independent variable. Therefore, if the rate of development of pressure is to be controlled, it must be done by changing flame area or flame speed.

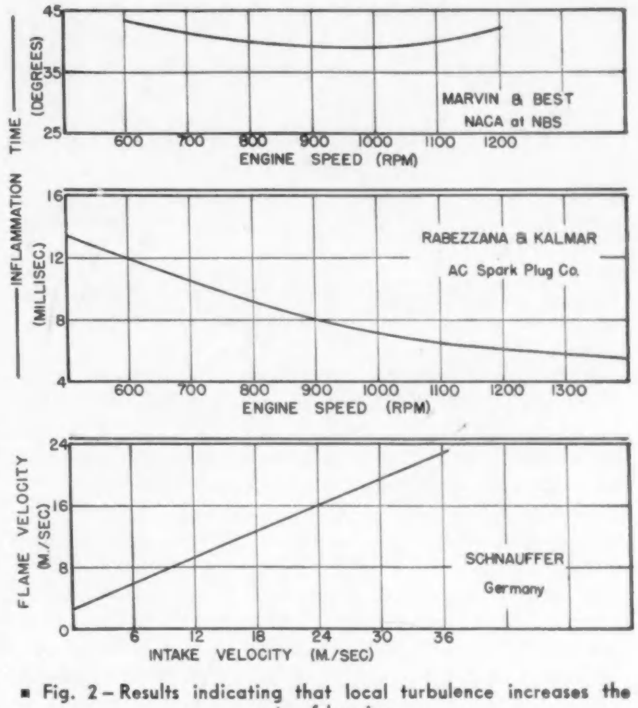
Flame in a closed vessel always tends to assume the shape of the restraining walls. Hence the area of the flame front at any distance from the spark plug will be influenced by the shape of the combustion chamber. Thus the rate of pressure rise is subject to some control by changing the contour of the chamber.

The movements occurring throughout the unburned charge also influence flame area. In addition to the movement of the entire body of gas, sometimes called general swirl, more important small-scale motions, which may be designated as local turbulence, appear to occur at random throughout the charge. These result in ragged flames of greatly increased area, and it is probable that their existence was first suggested by the fact that photographs of flame fronts in engines were much less sharp than those in a charge that was initially quiescent. It was also found that flame moves faster in an engine than it does in the same charge fired in a bomb. Finally, the speed with which flame moves in the combustion chamber was found to increase with engine speed.

Some of these points are illustrated graphically in Fig. 2. In the top strip, data obtained at the National Bureau of Standards[9] by direct observation of the flame, show that complete inflammation takes place over approximately the same angle of crank rotation, regardless of engine speed.

In the center strip, results of Rabezzana and Kalmar[10] show how the time for complete inflammation drops as the engine speed is increased. These data, plotted on the basis of crank angle instead of inflammation time, are in agreement with those of the top strip.

In the bottom strip, the results of Schnauffer[11] show a direct proportionality between flame speed and intake velocity, which, in turn is nearly proportional to engine



■ **Fig. 2 – Results indicating that local turbulence increases the rate of burning**

speed. The local turbulence seems to be the only factor which could reasonably be expected to change in this same manner with engine speed. Hence, through a group of independent observations, we have been led to believe that local turbulence in the engine is helpful in increasing the mass rate of burning by rendering the flame front irregular.

If it can be discovered how local turbulence is induced, perhaps a method can be found for intensifying it. If this can be done, a desirable effect upon power and efficiency will result, and it seems highly probable that the more rapid burning would decrease the tendency to knock, since less time would be available for preflame reactions.

[7] See NACA Technical Note No. 559, March, 1936: "Combustion-Engine Temperatures by the Sodium Line-Reversal Method," by Maurice J. Brevoort.

[8] See SAE Transactions, April, 1935, pp. 125-133: "Flame Temperatures Vary with Knock and Combustion-Chamber Position," by Gerald M. Rassweiler and Lloyd Withrow.

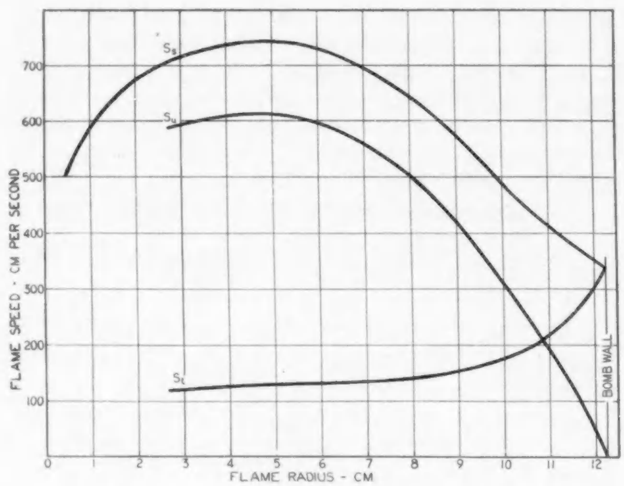
[9] See NACA Technical Report No. 399, 1931: "Flame Movement and Pressure Development in an Engine Cylinder," by Charles F. Marvin, Jr., and Robert D. Best.

[10] See *Automotive Industries*, March 2, 1935, p. 326: "Factors Controlling Engine Combustion," by Hector Rabezzana and Stephen Kalmar.

[11] See NACA Technical Memorandum No. 668, April, 1932: "Combustion Velocity of Benzine-Benzol-Air Mixtures in High-Speed Internal-Combustion Engines," by Kurt Schnauffer.

In addition to the area of the flame front, the other factor which determines the mass rate of burning is the velocity of the flame with respect to the unburned charge. Although it is a comparatively simple matter to measure the speed of flame in space, few experiments are capable of yielding reliable values of this more fundamental speed relative to the unburned gas. Moreover, there is as yet no satisfactory theoretical basis upon which values of this so-called transformation velocity can be calculated.

In an engine the pressure and temperature of the unburned charge vary continuously during the explosion.



■ Fig. 3 – Speeds of flame and unburned gas in chemically equivalent mixtures of CO and O_2, initially at a pressure of 1/3 atmosphere

Each of these variables is believed to affect transformation velocity, but much is yet to be learned about the magnitudes of these effects.

Although it does not seem feasible to measure transformation velocity in the engine cylinder, this can be don for explosions in a spherical bomb under carefully controlled conditions. Such an apparatus, shown on page 333, has been used at the National Bureau of Standards.[12]

Briefly, the bomb is a 10-in. sphere with central ignition, a window through which the progress of the flame can be photographed, and six openings through which pressure indicators may be attached. From simultaneously recorded time-pressure and time-displacement data it is possible to calculate both the transformation velocity and the expansion ratio at any chosen stage of the burning.

Results have been obtained for five different fuels at relatively low initial pressures, and experiments at higher initial pressures are contemplated.

Fig. 3 shows how the speed of flame in space (S_s), the velocity of the unburned gas ahead of the flame (S_u), and the transformation velocity (S_t) vary as the radius of the flame increases in a mixture of CO and O_2, initially at a pressure of 1/3 atmosphere. Note that the transformation velocity shows a progressive increase as the temperature and pressure of the charge go up. It has been demonstrated that both of these variables contribute to the increase.

[12] See NACA Technical Report No. 682, 1940: "Flame Speeds and Energy Considerations for Explosions in a Spherical Bomb," by Ernest F. Fiock, Charles F. Marvin, Jr., Frank R. Caldwell, and Carl H. Roeder.

[13] See NACA Technical Report No. 622, 1938: "A Photographic Study of Combustion and Knock in a Spark-Ignition Engine," by A. M. Rothrock and R. C. Spencer.

It has also been found that the transformation velocity increases in a similar way during explosions of the hydrocarbons, that the addition of ethyl fluid is without effect upon this property during the normal burning, and that there is no relation between transformation velocity under the conditions of the experiments and the tendency to knock.

Results obtained with the spherical bomb also indicate that the rise in pressure for a given mass of charge inflamed is less than would be expected from calculations based on thermal data and the assumption that reaction goes to chemical equilibrium in a very thin flame front. This fact is shown in Fig. 4, in which the dashed lines in the lower left-hand corner are calculated. Such values are reliable only during the early stages of the burning, while the temperature gradient in the unburned gas is negligible. Note that, for a given percent of the mass inflamed, the observed rise in pressure is considerably less than the calculated rise.

This result is interpreted to mean that reaction and evolution of heat must continue for some distance behind the flame front, which contention is supported by other evidence obtained with the spherical bomb. It is also supported by two independent studies of combustion in the engine cylinder.

From high-speed motion pictures of combustion in a spark-ignition engine at Langley Field, Rothrock and Spencer[13] find "evidence that reaction is not completed in the flame front but continues for some time after the flame front has passed through the charge."

The results of Withrow and Cornelius[6] at the General Motors laboratories, showing that the actual pressure prevailing at any instant during the burning in an engine cylinder is lower than that expected from the thermodynamic charts, have already been mentioned.

Here again there are three independent pieces of evidence which, taken together, clearly indicate that the time required to establish chemical equilibrium in the burned gas is an appreciable fraction of the total time of inflammation.

The presence of this so-called "afterburning" may be important in the course of future combustion research. In the first place it greatly complicates any theoretical treatment, as well as the compilation of thermodynamic charts. Again, Rothrock and Spencer suggest "that, in some cases, knock may occur in the burned gas as a result of sudden liberation of energy remaining after the initial passage of flame." Obviously the efficiency of the combustion process can be improved and the probability of this kind of knock can be decreased, if a method of promoting more complete reaction in the flame front can be evolved. A solution of this problem seems entirely within the bounds of possibility, and experimental work in this line might pay big dividends.

(3) *Control of Preignition* – Passing on now to the third problem of combustion control as previously enumerated, namely, preventing preignition in spark-ignition engines and facilitating ignition in compression-ignition engines, we find a field of growing importance, and one in which research is only well begun.

It seems probable that even today the two really distinct phenomena, preignition and detonation, are not always recognized as such. Indeed, this condition is not strange, since both may be accompanied by somewhat similar audible manifestations, and since both may occur, though

at different stages of the burning, during the same explosion.

No satisfactory method has yet been evolved for determining the relative tendencies of fuels to preignite in engines. The need for such a scale becomes greater as new fuels are being prepared, and it is encouraging to note that the National Advisory Committee has already undertaken studies of preignition in the combustion apparatus and in a CFR engine at Langley Field.

It is a personal opinion that the obscure processes leading to preignition or compression-ignition, and to detonation, are in some way produced by homogeneous reactions, or more specifically to unidentified products of such reactions, which occur between fuel and oxygen prior to the initiation or to the arrival of flame. Extending this theory a bit farther, the intermediates which may be responsible for preignition are probably different from those which lead to detonation. Both may be so shortlived that they do not exist at all as stable compounds, and hence may be difficult to identify, even by spectroscopic methods.

(4) *Suppressing Detonation* – The most familiar and most successful method yet devised for improving performance is that of suppressing detonation so that higher compression ratios may be employed. In addition to improvements in engine design, much of this progress has been made possible through new processes of refining and synthesizing hydro-carbon fuels, and through the use of antiknock additives such as tetraethyl lead.

A great deal of experimental work, most of which may with justice be classed as combustion research, has been and will continue to be done in an effort to evolve a satisfactory scale for the relative tendencies of various fuels to knock. Although all are aware of the great practical value of the present methods of rating fuels, it can hardly be denied that further improvements and extensions are of pressing importance.

The phenomenon of detonation or fuel knock in a spark-ignition engine is characterized by an abnormally high rate of inflammation of the last portion of the charge to burn, and is accompanied by excessive local pressures and rates of pressure rise, high heat losses, objectionable noise in the automobile engine, and stresses which may be destructive to the aircraft engine. Although its effects – all of them undesirable – are well known, and although numerous means for suppressing it have been discovered, the real nature of fuel knock is not yet understood.

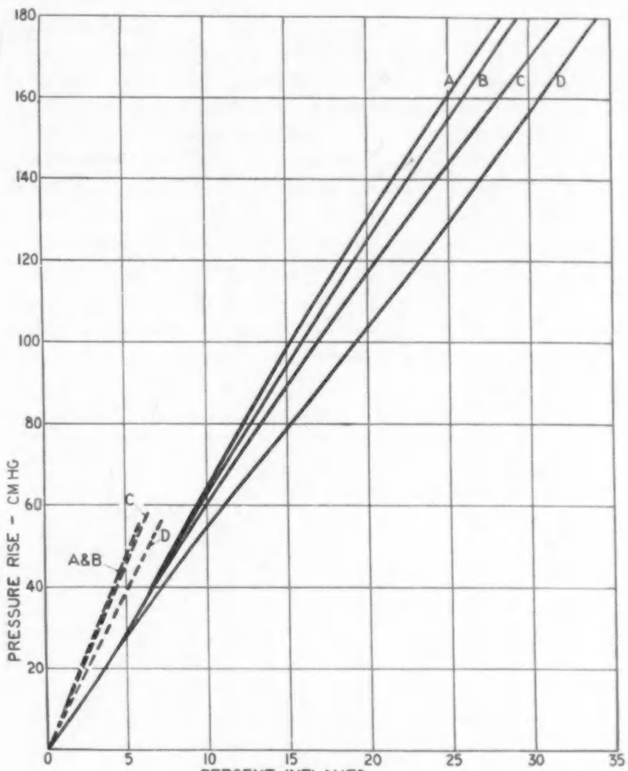
High-speed motion pictures of knocking combustion in the engine cylinder are one means of studying the phenomenon. Such records, taken in both the General Motors laboratories[14,15] and the National Advisory Committee laboratories,[13] show a great deal about the physical nature of the process, although a more complete and useful interpretation of the events which the camera shows transpiring during the processes of normal burning and detonation awaits the evolution of other basic information as to the principles of gaseous combustion.

The pictures show that the flame front suffers a sudden and very great increase in speed at the start of the detonation and then continues its travel at this same speed until it has traversed the remaining unburned charge. Some records have been interpreted as showing the origin of at least one other flame, separate and distinct from that started by the spark, and indicative of spontaneous ignition in the unburned gas. All the records seem to agree on one major point, namely, that detonation in the engine is not spontaneous burning throughout the entire mass of unburned charge, but is, instead, a progressive process.

High-speed schlieren photographs taken at Langley Field[13] show that detonation is followed by pressure waves of high frequency and intensity in the burned gas. These pressure waves, reflected back and forth across the combustion chamber at the velocity of sound in the very hot and highly compressed gas, give rise to the audible knock in much the same way as if a solid body were bouncing back and forth at the same frequency.

Since the pressure is so high in the moving wave, the temperature is likewise higher here than in the remainder of the gas. Hence more energy is radiated to the walls from the wave. The increased loss by radiation, together with the mechanical loss involved in imparting vibration to the solid parts of the engine, accounts for the increased heat loss which has long been recognized as one of the undesirable results of detonation.

Despite the fact that a fairly satisfactory picture of the physical manifestations of detonation has been evolved, the greater task of discovering why it occurs remains for the future. It is hoped that a part of the missing information can be obtained by giving more careful attention, not to normal burning and not to detonation individually but to the obscure and neglected transition from the former type of burning to the latter. It is in this transition stage that differences in the behavior of fuels should become ap-



■ **Fig. 4 – Rise in pressure produced during the inflammation of various mixtures. Solid curves are experimental and dashed lines are calculated. The fuels are *A*, normal heptane; *B*, 2, 2, 4 trimethyl pentane; *C*, benzene; *D*, carbon monoxide**

[14] See *Industrial and Engineering Chemistry*, June, 1936, pp. 672-677: "High-Speed Motion Pictures of Engine Flames," by Gerald M. Rassweiler and Lloyd Withrow.

[15] See SAE Transactions, August, 1936, pp. 297-303, 312: "Slow Motion Shows Knocking and Non-Knocking Explosions," by Lloyd Withrow and Gerald M. Rassweiler.

parent, and some of the vital factors governing detonation should be most readily discovered. To this end apparatus involving an untried method of study is currently being constructed at the National Bureau of Standards with the support of the National Advisory Committee.

Although little at present can be said of this apparatus, it is designed to determine the effects of various factors such as pressure, temperature, and time upon the reactions which take place homogeneously throughout mixtures of hydrocarbons and air, at conditions approximating those prevailing in the end gas of an engine. These experiments may or may not support previous evidence that preflame reactions are responsible for the start of detonation, and they may show the way to a new method of determining the relative tendencies of fuels to knock.

■ Conclusion

There have been presented a few examples of what has been done and what remains to be done in the field of combustion. That the discussion for the most part has been restricted to combustion in the engine cylinder does not mean that significant advances in other fields, such as safety, are of lesser importance.

After nearly ten years of association with combustion research there is an ever-growing impression that nothing short of a complete understanding of the whole process of ignition and combustion in explosive mixtures of gases will prove adequate to answer the highly practical questions that might, with reason, be asked by the engineer seeking to improve the design and construction of an internal-combustion engine. Such a picture can be evolved only by the protracted and cooperative efforts of interested groups, and active participation by new and larger groups is now highly desirable.

Although there has been a steady accumulation of a large number of experimental facts, either no mind capable of fitting these together to form a comprehensive whole has yet appeared or, as seems more probable, some important factors have so far been overlooked entirely. Since it is rarely possible to evolve individual results that are of immediate practical value, the student of the combustion process, as well as his financial sponsor, is subject to frequent attacks of discouragement at the apparent lack of progress.

However, such attacks are usually short-lived because the field is an intensely interesting one and because it is well known that collective effort along lines which are sound in principle, has always led to success.

In closing it is pertinent to recall two thoughts, coming from, and perhaps in a measure directed to, those who may be in a position to sponsor research of this sort. The first, attributed to Dr. Kettering, is that pure research is the kind that looks financially hopeless now but in 25 years pays the biggest dividends of all. The second is a statement made by E. C. Williams of the Shell Development Co. that "it is so extraordinarily difficult for a research worker to say with definiteness, while he is yet on the way, whether he will ever arrive, or when, or what he will have when he gets there."

DISCUSSION

More on Indicator Cards and Combustion Control

—Melville W. Beardsley
Mechanical Engineer

BESIDES his excellent direct presentation of the status of combustion research, Dr. Fiock made one statement introducing a subject worthy of further development. To quote him: "It is therefore logical, in endeavoring to improve the combustion process, to decide first what improvements are possible in the indicator card, and then to seek means of obtaining these characteristics through combustion control."

It is enlightening and of more than academic interest to determine what performance might be achieved if the shape of the indicator card diagram could be controlled. Consider these basic aircraft-engine characteristics:

1. High maximum power for take-off.
2. Low specific fuel consumption for cruising.
3. Low engine weight.

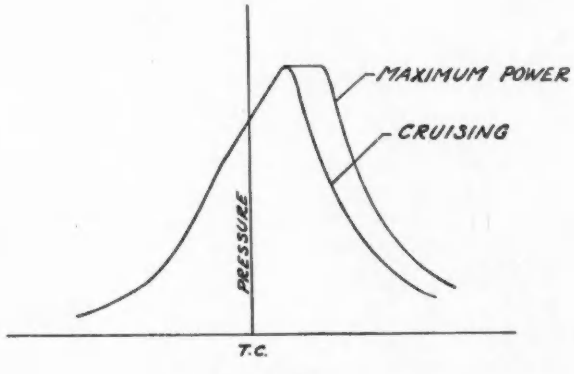
Now consider how these characteristics are related to the engine indicator card diagram:

1. Maximum power – diagram area (more area = more power).
2. Specific consumption – diagram shape (constant-volume combustion = low consumption).
3. Engine weight – diagram height (higher pressure = heavier engine).

It is obvious that these indicator-card requirements are mutually at odds, and that a compromise must be effected in the design of an engine.

Consider the engine characteristic of weight per horsepower. The allowable indicator diagram height is limited because the engine weight and strength of materials permit only pressures up to an established maximum. For efficient utilization of weight, the engine should operate under conditions at which the design maximum pressure is continuously developed. In other words, this design maximum pressure should be developed at cruising power output – thus making the engine weight roughly proportional to the cruising power instead of the maximum power output. In order that the design maximum pressure is not exceeded for maximum power output, the necessary increase in diagram area must be secured by having the latter part of the combustion take place at constant pressure. The increased combustion period lowers the cycle efficiency, but this is ordinarily of secondary importance for maximum output operation.

For an engine under consideration, determine the cruising power, and calculate the maximum pressure developed with constant-volume combustion. This constant-volume combustion will assure most efficient operation for cruising. Set this constant-volume maximum pressure as the design pressure for the engine structure. For maximum output conditions the combustion must be completed at constant-pressure – the constant-volume phase having raised the pressure to the design maximum allowable.



■ Fig. A (Beardsley discussion) – Superimposed indicator diagrams for maximum power and cruising for an aircraft engine

The superimposed indicator diagrams for maximum power and for cruising will then appear as in Fig. A. In both cases the maximum allowable pressure is reached by approximately constant-volume combustion – the rate of pressure rise being reduced to a value consistent with smooth running. For cruising, the combustion ends with the achievement of maximum pressure; while, for maximum power, an additional phase of constant-pressure combustion is required to utilize the extra fuel necessary for the increased cycle output.

These diagrams are admittedly hypothetical – showing what might be achieved, but indicating no process by which the desired results might be accomplished. Brief contemplation makes it clear that the realization of these two cycles in the same engine is virtually impossible (with current knowledge of combustion anyway) as long as the cylinder is charged with a complete fuel-air mixture before the start of combustion – such as in the case of a carburetor-equipped engine. The suggested solution is that of charging the cylinder with air – or a very lean mixture – and controlling the combustion by fuel injection during the combustion. For consistent operation under all conditions spark ignition will probably be necessary, and both combustion chamber design and airflow will be important considerations.

So far as I am aware, there have been only scant suggestions of this type of combustion-control system in the literature of this country or abroad – and no indications that any actual research has ever been performed. It might be anticipated that problems will arise – such as those concerning the fuel injection system and exhaust valve temperatures at maximum power – but these should not prove insurmountable.

Altogether, considering the possibilities suggested by indicator-card analysis, and considering the fuels that might be used, it appears that a combustion-control system incorporating spark ignition and fuel injection during combustion is worthy of research investigation.

Problem of Diesel Oxygen Utilization

– Robert F. Selden

National Advisory Committee for Aeronautics

1. Dr. Fiock's paper presents an interesting and thought-provoking array of questions that still bother those interested in the innermost secrets of the combustion process. From the standpoint of our ignorance, this subject bears a striking resemblance to the "science" of sociology – a great many facts are evident, but the missing links confuse the issue. Each expert accordingly feels free to draw his own conclusions. Some have thought that the detailed mechanism of the hydrogen-oxygen reaction was fairly well established. However, Oldenberg[a] et al. have sounded a recent note of uncertainty in their failure to discover hydroxyl radicals in the non-explosive reaction. In view of the uncertainties existing in the mechanisms of such simple combustion reactions, considerable fortitude and a genius for coordination will be necessary to untangle fully our engine combustion problems.

2. The main problem in diesel-engine combustion is not ordinarily that of getting a short enough ignition lag, as the author states, but rather in utilizing all the oxygen in the engine as effectively as in spark-ignition engines. It is my personal opinion that this is not entirely a matter of insufficient mixing, important as this may be. In fact, we have improved this mixing of fuel and air until the increased cost of doing so, measured in terms of increased heat transfer to the combustion-chamber walls and of increased mechanical friction, has begun to devour the improvement that might otherwise be expected.

The remaining difficulty probably arises from the formation of soot in the neighborhood of the fuel spray by virtue of the prevailing low oxygen concentrations and high ambient gas temperatures. If these soot particles are large enough, it is conceivable that they will not burn completely within the time available, even though they do come into contact with free oxygen late in the expansion stroke. If this picture is in any wise correct, it is evident that there is a limit beyond which we cannot go by increasing the rate of mixing of fuel and air.

3. In his discussion of Fig. 4, the author states that the transformation velocity increases with increases in charge temperature and pressure. It is my feeling that it would be better to employ the concept of charge density rather than of pressure, particularly where large temperature changes are involved. It is the concentration of the reactants, expressed in terms of molecules per unit volume, if you will, that is of real interest. Density has such a connotation, whereas pressure does not.

[a] See *The Physical Review*, Vol. 58, 1940, p. 1121: "The Spectra of the Thermal Reaction Between Hydrogen and Oxygen," by O. Oldenberg, E. G. Schneider, and H. S. Sommers, Jr.

Automotive Bearings

Fatigue Resistance

ANY metallic bearing, if run under completely ideal conditions and if run long enough, eventually will fail due to fatigue, provided the load exceeds the endurance limit of that particular metal or alloy. It might be said that the bearing dies of old age.

With respect to fatigue, we can rate bearing alloys in the following order:

1. Copper-lead
2. Cadmium Alloys
3. Tin-base & Lead-base Alloys (Conventional Type)

The evidence of fatigue is the presence of numerous fine cracks in the bearing metal. These cracks invariably start at the bearing surface. The first phase may be considered as the birth of the crack. In the second phase, the crack works inwardly toward the bond line, its actual path being influenced greatly by the location of the grain boundaries in the bearing metal. The third phase is quite astounding and is very probably the immediate cause of most bearing failures. Here the crack turns at right angles when it has proceeded to within a very short distance of the bond line. The crack will then run in a plane somewhat parallel to the interface between bearing metal and backing metal. When this crack meets another crack which has worked down from the surface, the effect is to produce a small section of loose bearing material. The disastrous results of a number of these small loose pieces can readily be visualized.

From the foregoing explanation on the mechanism of fatigue, it can be seen why the performance of the thin-layer bearings should be far superior to the conventional type bearing. The manufacture of thin layer bearings of the Trimetal and Micro type requires a precision and a technique not thought possible just a few years ago. However, there are many millions of both of these types of bearings in successful operation today.

Corrosion Resistance

With respect to resistance to corrosion by acids formed in the oxidation of oil, the four general alloys group themselves somewhat differently. Here the tin-base and lead-base alloys are practically immune to any type of corrosion so far encountered in the field. On the other hand, the cadmium alloys and copper-lead are definitely susceptible to attack if corrosive conditions are allowed to exist.

Future Bearings

It is quite evident that the trend of future bearing developments is toward a very thin lining of a recognized bearing material backed by steel or steel and an intermediate layer of bronze. As production methods with respect to bearings and engines are improved, we may expect even thinner linings than are in use at the present time. In closing, it may be of interest to mention that "Micro" and "Trimetal" bearings have been produced by laboratory methods which have successfully sustained mean unit pressures in excess of 10,000 psi.

Excerpts from the paper: "Automotive Bearings," by John K. Anthony, The Cleveland Graphite Bronze Co., presented at a meeting of the Detroit Section of the Society, Detroit, Mich., March 31, 1941.

Effect of RIM WIDTH on CAR and

by R. D. EVANS
Manager, Tire Design Research Development Department, The Goodyear Tire & Rubber Co.

IT is the purpose of this paper to report on the recent test program undertaken to evaluate the several effects of the use of rims of various widths and to appraise the advantages and disadvantages of the proposition to adopt rims substantially wider than those which have prevailed during the past few years.

The conception of wide rims for pneumatic tires is nothing new. In fact, about 1915 a tire was developed and patents were granted for its use on a rim as wide as the tire itself, but it did not prove successful. By 1925 when the balloon tire had "arrived" as passenger-car equipment, rim widths had become fairly well stabilized at approximately 50% of the widths of the tires mounted thereon. This value, hereafter referred to as rim-width ratio, is a basic index of the functional relationship between tire and rim.

This 50% combination of tire and rim prevailed for 7 or 8 years, and retrospect indicates that it was satisfactorily in harmony with the general status of automotive design of that epoch.

By 1933 the so-called low-pressure balloon tire, popularized under the name "Airwheel" by Goodyear, was finding acceptance as original equipment on passenger cars. The rims recommended for these tires averaged 65% in width ratio. This is a 15% increase as compared with the 50% ratio for balloon tires, and was a very striking increase indeed. It reflected the realization that, by using wider rims for these tires, the faster treadwear and the poorer car handling associated with the low pressure and the small rim diameter could be partially offset. In fact, some tire engineers favored still wider rims for these low-pressure tires. Their opinion has been responsible for including, in the Tire & Rim Association's Year Books to an increasing degree from 1933 to 1940, alternative recommendations for rim widths up to about 73%.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.]

During 1940 there has been great interest in exploring all the aspects of still further rim-width increase, and in determining whether, in this direction, substantial and enduring progress is to be found.

As a result of this interest and of the large testing program which it has caused to be carried out, the tire industry through the Tire & Rim Association has approved, on an experimental basis, the use of rims up to approximately 81%. These expanded recommendations are shown below:

Table 1 – Tire and Rim Combinations
1941 Recommendations of Tire & Rim Association

Tire Size	Present Rims: Rim	Present Rims: Ratio	Wide-Base Rims: Rim	Wide-Base Rims: Ratio
5.50	3.50D	62%	4½-K	74%
	4.00E	68	5-K	79
6.00	4.00E	64	5-K	75
	4.50E	69	5½-K	80
6.25	4.25E	65	5-K	73
	4.50E	68	5½-K	78
6.50	4.50E	66	5½-K	76
	5.00E,F	72	6-L	81
7.00	5.00E,F	68	6-L	77
	5.50F	73	6½-L	81
7.50	5.50F	69	6½-L	77
	6.00F	73	7-M	81

It is to be noted that new flange shapes, K, L, and M, which differ in shape from the current D, E, and F flanges,

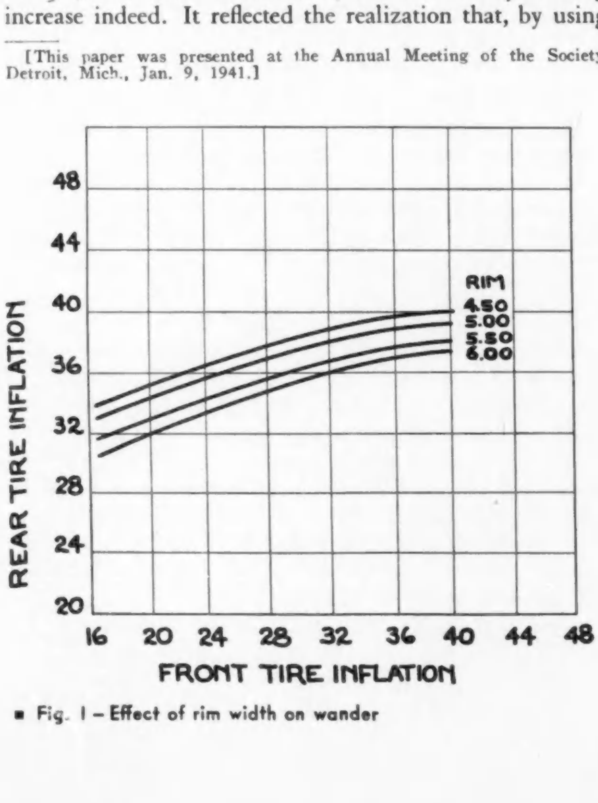


Fig. 1 – Effect of rim width on wander

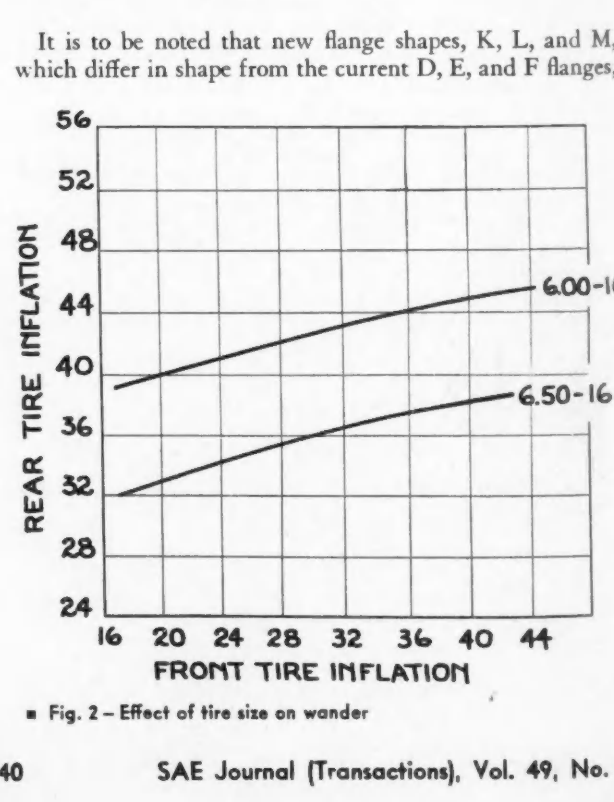


Fig. 2 – Effect of tire size on wander

TIRE PERFORMANCE

RESULTS of a recent test program undertaken to evaluate the effects of the use of rims of various widths are reported in this paper. Classifying effects of rim-width change into advantages and disadvantages, Mr. Evans shows first a 22% increase in treadwear by increasing the rim width from 65 to 80% of the tire width, and using an inflation pressure lower by 2 psi. Test results are presented to show the effect on the steering control of the car in terms of wander of the tire's progressively higher stability and cornering power as its rim width is increased.

are recommended where tires are used on rims for which the width ratio exceeds 73%.

An exploration of this kind involves extensively testing a wide range of equipment under a variety of conditions. Such a program necessarily requires a fairly long period of time, with adequate field and service experience. The results now to be presented must therefore be considered somewhat tentative, but are at least strongly indicative of the various effects attributable to the rim-width factor.

It has long been recognized that widening the rim on which a tire is mounted increases its lateral stability. In these respects, then, widening the rim should have effects closely corresponding to increasing the inflation pressure. Some of our data will, therefore, make use of inflation pressure as a sort of measuring stick.

It will be feasible as well as logical to classify into advantages and disadvantages the effects of rim-width change.

Advantages

1. Treadwear – As far as the tires themselves are concerned, the major, if not indeed the only discernible betterment, is that of treadwear.

Goodyear's results on treadwear are summarized in Table 2, in terms of treadwear ratings based on a 65% rim as 100% or standard. The higher the rating, the longer the treadwear. It is to be emphasized that these ratings apply only to *rate* of treadwear and do not include any index of *type* or relative *smoothness* or *uniformity* of wear.

Table 2 – Treadwear Ratings

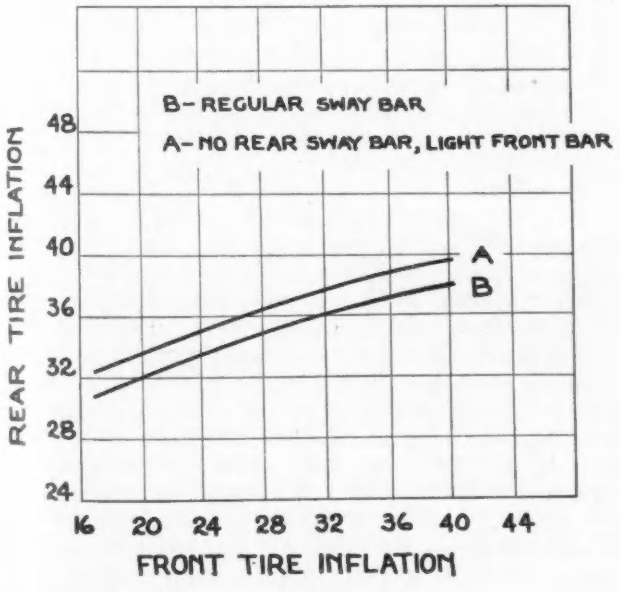
Rim Width, %	Inflation, psi	Treadwear, Rating, %
65	28	100
70	28	114
75	27	119
80	26	122

This table consolidates results on three sizes of tire, run on a total of 6 different cars of 3 different makes, and on routes in mountainous, in hilly, and in flat country. It averages both front and rear wheel tests, in some of which each tire maintained its wheel position throughout its test; in others, each tire "rotated" through the four wheel positions. But, in all cases, the four rims on a car at any one time were of the same width.

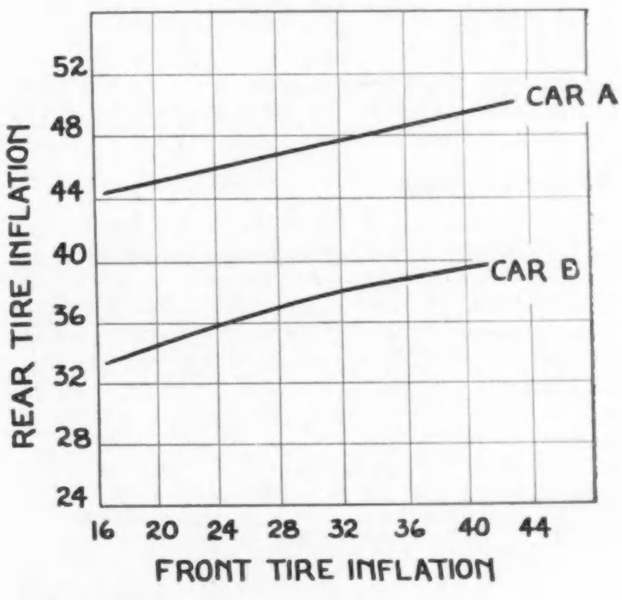
A degree of precision unjustified by the results should not be read into the foregoing ratings. For instance, the rating of 122% for the 80% rim was an average of tests in which individual results ranged from 112 to 130%. These variations are a matter of route, make of car, season of the year, and so on.

The reasons for the use of lower pressure with the 75% and 80% rims will be explained in a later paragraph.

2. *Steering Control* – The other advantage gained by widening the rim pertains to steering control of the car. The tire's progressively higher stability and cornering power as its rim width is increased are reflected in the wander and other handling characteristics of the car. Va-



■ Fig. 3 – Effect of sway-bar system on wander



■ Fig. 4 – Effect of make of car on wander

rious methods have been developed for evaluating these characteristics. Some methods use elaborate instrumentation; others depend on the "feel" of the car in the hands of expert drivers.

The data of Fig. 1 were obtained by a method of the latter type. In this method various conditions of front and rear tire pressure are determined which just eliminate a certain type of wander or "fish-tailing." With the same tires on a series of rim widths, the graphs of Fig. 1 are secured. Each line, such as the one marked 4.50 to indicate the rim width involved, marks the division between a "wander" and a "no-wander" behavior of the car. When values of tire pressure are below and to the right of the line, wander is present; when above and to the left, the car is stable.

The absolute location of such a line will vary somewhat with different drivers and observers, depending on the exact degree of "wander" they use as an index of the critical condition, but it is surprising how closely different observers will agree on the *slope* of any one line and the *separation* of two lines corresponding to different conditions, cars, tires, rims, or other factors.

Fig. 1 indicates that, at least as far as the type of steering control involved in this method of evaluation is concerned, the increase of rim width affords a proportionate advantage. Each inch of rim width is roughly equivalent to 2 psi inflation pressure.

It may be of interest to see what this technique of evaluating handling characteristics gives when factors other than rim width are changed. In Fig. 2 we see that a change of tire size only, with rims, car, and axle loads remaining the same, is "equivalent" to about 7 psi inflation.

Fig. 3 indicates the effect of changing the sway-bar system. It is to be understood that the degree of roll is not itself being evaluated; the observers attempted to determine the critical no-wander condition independent of the relative rolling behavior.

In Fig. 4 two different makes of car are compared. The same actual tires, the same rim widths, and same axle loads were involved in each case.

These other factors, such as tire size, sway-bar rigidity, and overall chassis design are portrayed with the idea of giving a sort of "importance rating" to rim width in relation to other aspects of automotive design.

In the preceding Figs. 1 to 4, the values of tire inflation pressure relate to special test conditions, and are not in any way to be considered as recommendations.

■ Disadvantages

These likewise may be classified into those which affect the car and those which are confined to the tire itself.

1. *Ride* – By stiffening the tire and reducing its deflectability, widening the rim causes a harder, harsher, or joltier ride. An increase from a 65% to an 80% rim corresponds to approximately 3 psi increase of inflation pressure, varying somewhat with the particular design of tire involved. As for the car itself, our tests indicate that it is necessary to decrease inflation 2 to 4 psi to get an equivalent ride with the 80% as with the 65% rim. This differential varies with different cars. Some observers have reported as much as 6 psi.

This differential, which all observers agree is not less than 2 psi, is the reason for the pressures used in the several series of tests summarized in Table 2.

It is clear that this pressure differential, believed to be necessary from considerations of ride, partially offsets the treadwear and stability advantages of wider rims. Indeed, the 6 psi differential indicated to be necessary by some engineers would entirely wipe out the stability advantage, and would largely offset the treadwear advantage previously described.

2. *Rim Damage* – With the wider rim the flange may be somewhat more exposed to curb damage because it is less protected by the bulge of the tire sidewall. Our evidence on this point is meager, but we do not believe it would prove to be an adverse factor of any importance.

3. *Uneven Front-Wheel Wear* – Goodyear's tests, as a general average, indicated slightly more irregularity of front-tire wear for the 80% rims, although the picture was not at all consistent. This is a most difficult characteristic to evaluate, since it depends on many variable factors, such as speed, type of route, design of car, brake and shock-absorber adjustment, driving habits, and possibly others.

It is to be kept in mind that this estimate of uneven wear is based on tests with a 2 psi difference in tire pressure as between the 65% and the 80% rim. Certainly if a 5 or 6 psi lower pressure were to be considered necessary for the wide rim, the unevenness of wear could be expected to be substantially worse.

4. Tire Durability – Except for treadwear, already discussed, widening the rim beyond the 65 to 70% range which has prevailed for several years is not necessarily favorable to better tire durability. The design of the bead structure, for instance, represents a long evolution in which the prevailing rim widths and flange shapes were intrinsically important factors. Any change in either width or flange shape has an effect on the distribution of stresses in the bead of the tire.

Early tests with rims in the 80% range gave some indications of poorer bead performance. However, with the modified flange shapes which are now embodied in the recommendations for rims 73% and upward in width, we believe that the bead picture will be entirely satisfactory. (See Table 1.)

There are several other aspects of tire performance which merit attention. These may be listed as tire hum or noise; squeal; the various aspects of tractive or nonskid action; steering-wheel effort at very slow speed and in parking. Rim width, within the ranges under discussion, has negligible effect on any of these items, except possibly to the extent corresponding to the pressure differential and directly caused by it.

In conclusion, attention may again be called to Fig. 4 and its implications. Here are two makes of car which exhibit considerable difference in certain handling characteristics. Possibly both of them are subject to some complaints about the manner in which they handle on the road. Yet both find wide acceptance by the public, the degree of which is dependent on many factors in addition to those which have been under discussion. It remains, therefore, the responsibility of the automotive engineer, guided by the more clearly marked signposts which have been set up by the past year's work, and yet remaining within the framework of Tire & Rim Association recommendations, to select those tire-and-rim combinations which best harmonize with his particular design and which afford the best all-around satisfaction to his customers.

WIDE-BASE Tire and Rim Construction

by SIDNEY M. CADWELL
U. S. Rubber Co.

WIDE-BASE rims impose different conditions of strains in tires and emphasize certain inherent performance differences, some of which are advantages and others disadvantages.

After a year of exhaustive testing by the combined car and tire industries, the wide-base rim proposal seems to have settled on the use of existing tire sizes on rims 1 to 1½ in. wider than at present, giving a rim ratio of 75 to 82% of tire width as compared with a ratio of 62 to 68% of the inflated width on existing tires.

The principal benefits of the proposed rim re-sizing combination, using present tire load-carrying capacity and 2 psi lower inflation pressure are: (1) considerably more stability in the car; and (2) a 20 to 22% increase in tire tread life.

In addition, the wide-base tire and rim combination will perform somewhat better for: tire cord fatigue, tire rim bruise resistance, tire groove cracking resistance; would perform equally well for: tire heat dissipation up to 75 mph, tire power consumption, tire tread and fabric separation, tire sidewall breaks at the rim, tire squeal on turns, and tire noise or hum on straight roads; but would be inferior for: tire ride, tire harshness, pavement seam bump absorption, tire and car parking effort, rim curbing, and tire tread shoulder cracking.

THE modern American automobile, having available top speeds of 85 to 90 mph, uses tires designed to function up to these speeds with good treadwear, economy, safety, and comfort on the standard rim at the same high standard attained in the automobile. This has always been the requirement of automotive engineers in the development of new model cars.

Soft ride, one of the important car qualities of the past five or six years, has been improved through the car engineers' greater use of suspension systems using lower spring rates, better shock-absorber control, better weight distribution, and softer-riding tires to give the overall soft, silent automobile ride we are accustomed to.

During the past year there has been considerable discussion among car and tire engineers of the benefits which might be derived from using tires on wide-base rims.

■ Present Rim Widths

The low-pressure tires used today at inflations of from 25 to 30 psi are used on rims having a width between the vertical rolled flanges of from 60% to 68% of the inflated tire width. All tire structural and chemical functions have, with this set-up, been balanced carefully over a period of years to complement the performance of the automobile in the average service, as well as extrêmes of service. Wide-base rims impose different conditions of strains in tires and emphasize certain inherent performance differences, some of which are advantages and others disadvantages.

■ Proposed Wide-Base Rim Width

The wide-base-rim proposal considers substantially the use of existing size tires on rims from 1 to 1½ in. wider than at present, giving a rim ratio of 75% to 82% of tire width. Numerous other proposals have been considered or tested in which the rim-to-tire-width ratio has been increased by varying amounts, as well as proposals to use smaller tire carcasses with deeper, wider treads. The present trend of development, after a year's exhaustive testing by the combined car and tire industries, appears to be settling at existing standard tires on rims 1 to 1½ in. wider.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.]

■ Wide-Rim Tire and Car Performance

The principal benefits of this rim re-sizing combination, using present tire load-carrying capacity and 2 psi lower inflation pressure, are: (1) Considerably more stability in the car; and (2) a 20% to 22% increase in tire tread life. The treadwear improvement is an average of all tests of all companies for tires on the widest rim combinations as compared with those on the present narrow rim combinations. These average figures were quoted by the Tire and Rim Association at a meeting held in Detroit, Nov. 8, 1940, with car manufacturing representatives, as a general statement of improvement resulting from a large number of tests covering both front and rear tires and several sizes. It is not to be considered as a specific prediction.

In addition to the better car performance, our tests have shown the tires will perform somewhat better for:

I – Tire cord fatigue
II – Tire rim bruise resistance
III – Tire groove cracking resistance
IV – Cross-wind handling

and would perform equally well for:

I – Tire heat dissipation up to 75 mph speeds
II – Tire power consumption
III – Tire tread and fabric separation
IV – Tire sidewall breaks at the rim
V – Tire squeal on turns
VI – Tire noise or hum on straight roads

but would be inferior for:

I – Tire ride
II – Tire harshness
III – Pavement seam bump absorption
IV – Tire and car parking effort (slightly inferior)
V – Rim curbing
VI – Tire tread shoulder cracking

■ Adjustments for Wide-Base Rims

Car and tire design engineers appreciate that the tire for modern motor vehicles is a complex mechanical and chemical elastic system enclosing the supporting air cushion on which the car rides. In readjusting this complex structure for those inferior properties just mentioned, it has been determined that they can be brought to the equal of present tires on existing rims by the tire manufacturer assuming responsibility for tire harshness, and tire tread shoulder cracking. Car engineers have, in general, indicated that future new cars can be brought to the equal of present performance for ride, pavement seam bump, and parking effort, by changes in the car steering, suspension and shock-absorber system. The problem of greater rim curbing with a sharp increase in number of rim edges with paint rubbed off and battered, rasp worn appearance may bring unfavorable customer reaction. Drivers in the future must exercise more care in parking, and they will be forced to cultivate the parking habits of owners of white sidewall tires.

■ Ride Definition

Soft ride, a feature of modern cars, has been lost to a great extent when tires are used on wide-base rims. The inability of the tire to absorb major road shocks as efficiently as the tire on the standard rim is inherent because the straighter sidewalls, acting as flexible supporting struts for the tread, require more effort to deflect vertically when major road-surface variations are to be absorbed, thus making the ride harder but the stability better.

Softness of ride may be defined as the ability of the tire to absorb those road shocks which are of sufficient magnitude to cause the whole tire, both tread and sidewall, to deflect a major amount.

Efforts to improve ride of wide-base-rim tires to equal the performance of the same tire on standard rims have not been successful to date. Tire ride softness alone may be controlled to an extent by use of tread patterns having open, flexible tread units in combination with narrow light-weight treads. Designs of this type, however, are invariably poor for treadwear life and stability, and would therefore cancel the principal benefits of wide-base tire usage.

After a year of testing by the entire industry it is generally agreed that tires operating on wide-base rims may use 2 psi lower inflation pressure for the same load capacity used for tires on standard rims because the larger volume of supporting air cushion seems to permit this reduction without harmful tire effects. This 2 psi reduction does not compensate for all the inherent ride difference, however, but further reductions have resulted again in loss of treadwear and stability to a point where the wide-rim tires performed no better than existing tires on standard rims.

■ Harshness Control

Harshness control, a factor previously combined with ride softness and measured in with overall ride, makes its appearance as an independent factor when tires are used on wide-base rims. It may be considered as the measure of a tire to absorb those road-surface irregularities which cause only minor displacements and are absorbed mostly by the deformation within the elastic tread. Harshness has been controlled experimentally by structural changes in the tire, and it probably will not be a difficult problem for the tire manufacturer if wide rims are used in the future.

■ Tire Re-sizing

Various re-sizing proposals have been considered in evaluating the possible benefits of wide-base rims such as using a wider, heavier tread with deeper anti-skid on a carcass of smaller size. Our company is not in favor of such extensive redistribution of tire structural balance because of the lower safety factor resulting from increased heat generation in tire-tread shoulders.

■ Future Tire and Car Trend

A new condition of tire and car service has made its appearance during the past year. The new non-stop, non-speed-limit Pennsylvania Turnpike has been completed and is now in operation. This 160-mile-long super highway has actually been traversed, with some care, by experienced operators at speeds of 110 mph. There is a certainty that car manufacturers will make cars to utilize fully the uninterrupted high-speed travel such a highway permits. In view of the future national-defense possibilities presented by extension of this type of road there may appear a new high-speed car and tire trend which cannot be ignored.

The problems of suitable tires to meet this new trend are being studied intensively. Tread shoulder heat generation and considerably shortened tread-wear life are the most important quality considerations. The use of wider rims by itself may not seriously affect tire service, but the use of full-sized tire carcasses, standard tread depths of nominal width, as well as some changes in compounds, are probably going to be the minimum requirements for this type of operation.

■ Conclusion

We approve the use of tires on wide-base rims if the combined efforts of the car and tire engineers to re-balance the changed tire performance properties results in future cars of at least equal comfort and safety.

General Approach to the FLUTTER PROBLEM

by S. J. LORING
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A GENERAL approach to the flutter problem is outlined, which can be used to investigate any mode of flutter of any structure provided the air-forces are known.

The method can be used to investigate the possibility of flutter on such structures as airplane wings, tail surfaces, and bomb doors, aircraft propeller blades, vehicular bridges, buildings, and so on.

With a well-known solution for the air-forces for two-dimensional flow over an airfoil with an aileron, equations have been derived which can be used to determine the flutter speed for the wing or tail surfaces of any conventional airplane.

The use of still-air vibration tests in obtaining the structural friction of structures, and in checking a part of the flutter computations experimentally is indicated.

Finally, to suggest the possibilities in the use of this method, a standardized procedure for its general application to aircraft flutter problems is outlined briefly.

I. Introduction

FOR the purposes of this paper flutter will be defined as a self-excited or unstable oscillation arising out of the simultaneous action of elastic, inertia, and aerodynamic "lift" forces upon a mass, or a system of masses. This type of vibration has been observed most frequently in aircraft, but is known in some other fields; the "galloping" of sleet-covered electrical transmission wires in high winds is one example, and the recent spectacular vibration of a suspension bridge[1], which eventually caused its failure, appears to have been flutter.

Flutter is an increasingly important factor in aircraft design because of the trend toward higher speeds and the use of the more flexible structures which result from cutouts for access, visibility, or equipment. At present the most useful theories for predicting flutter[2, 3, 4] consist of dynamic analyses, with the aerodynamic forces included, of very simple systems which are made to represent, by analogy, the more complicated systems occurring in practice. Success in predicting flutter speeds in this manner has been gratifying in cases where there has been a close analogy between the actual system and the simple one which had been analyzed; too often, however, there is little or no analogy between actual structures for which flutter characteristics are desired and the simple systems. In the latter cases the existing flutter theories cannot be used very satisfactorily. For example, there are several possible flutter modes of conventional airplane tail surfaces, involving the simultaneous movement of both horizontal and vertical surfaces and of the fuselage, that cannot be handled adequately by existing flutter theories.

The purpose of this paper is to outline a general approach to the flutter problem through the use of some well-known principles and relations of dynamics. It is hoped that this approach will eventually lead to methods of flutter analysis which will be as straightforward and dependable as are the methods of structural analysis at present. In the last section of this paper, a possible line of development in this direction is suggested. The paper is by no means self-contained. Flutter is, unfortunately, a problem of considerable complexity. There have been, however, numerous works devoted to various aspects of the general problem of vibrations, and it is through developments and extensions of these works that the results of this paper are obtained. The developments presented in this paper are based on Lagrange's equations of motion and the connected concept of generalized coordinates, and make use of matrix notation. The latter has proved so convenient and concise for writing general expressions that it has been adopted throughout. The air-force expressions of Reference 2 have been used in Sections VII and VIII. Previous work upon which the main results of this paper are based are fully covered in References 2, 5, 6 and 7.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 6, 1940.]

[1] Tacoma Narrows Suspension Bridge which failed Nov. 7, 1940, Tacoma, Wash.

[2] See NACA Technical Report No. 496, 1934: "General Theory of Aerodynamic Instability and the Mechanism of Flutter," by Theodore Theodorsen.

[3] See *The Journal of the Royal Aeronautical Society*, Vol. 41, No. 322, October, 1937: "The Two-Dimensional Problem of Wing Vibration," by Kassner and Fingando.

[4] See NACA Technical Report No. 685 (Restricted), 1940: "Mechanism of Flutter, A Theoretical and Experimental Investigation of the Flutter Problem," by Theodore Theodorsen and I. E. Garrick.

[5] See "The Theory of Sound," by Lord Rayleigh, Vol. I, Second Edition, Macmillan & Co., 1896.

[6] See "Vibration Problems in Engineering," by S. Timoshenko, Second Edition, D. Van Nostrand Co., Inc., 1937.

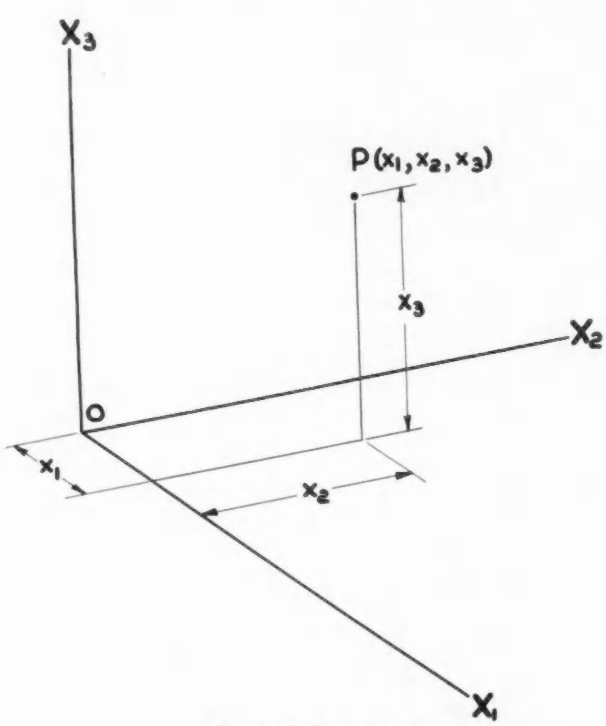
[7] See "Elementary Matrices," by R. A. Frazer, W. J. Duncan, and A. R. Collar, Cambridge University Press, 1938.

An attempt has been made, in preparing this paper, to make the arguments and derivations as general as possible so that they may be applied to any problem. For the purpose of making each step in the derivations clear, an example of the application of the general methods to a particular structure is included. In Sections VII and VIII, where the air-forces are introduced into the equations of motion, the lack of general air-force expressions has made it necessary to restrict the generality of the equations to the limits of applicability of the air-force expressions used. The general method of approach in these sections, however, should be usable for any more general (linear) air-force expressions which become available in the future.

To avoid the use of specific terms in the general arguments, and to aid thinking in mechanical terms, the structure which is being analyzed for flutter will be termed "mechanical system" or simply "system."

II. Specification of Displacements

A mechanical system which has its mass distributed along a line, over a surface, or within a volume, is described as "continuous." Most of the structures to be analyzed in connection with aircraft flutter investigations, for example, wings, fuselages, tail surfaces, constitute continuous mechanical systems. Since any structure in which



▪ Fig. 1 – Reference axes

the masses are concentrated can always be treated as the special case of a continuous system in which the masses are distributed over very small regions, there will be no loss in generality in restricting the arguments in this paper to continuous systems.

To describe the state of deformation of a continuous system it is necessary to specify the displacement of each of the infinite point masses of which it is composed. If the system is referred to a set of mutually orthogonal axes, OX_1, OX_2, OX_3, any point mass P may be identified by the coordinates x_1, x_2, x_3, of its undisplaced position (see Fig. 1). The displacement of the point P is completely described by its three components u_1, u_2, u_3, measured along the coordinate axes; and the coordinates of the displaced position are $x_1 + u_1$, $x_2 + u_2$, $x_3 + u_3$. The displacement components u_1, u_2, u_3, are, in general, functions of x_1, x_2, x_3, and of time t.

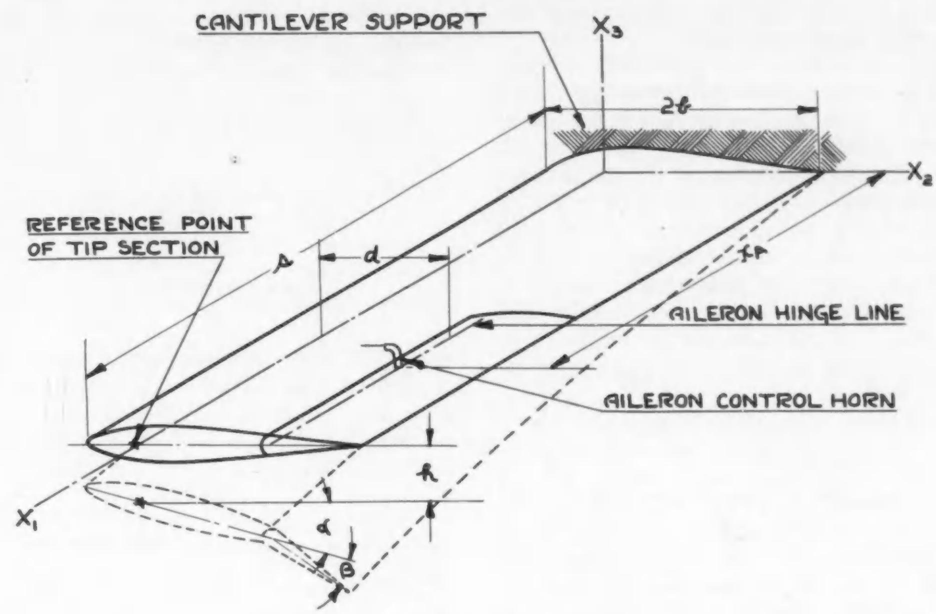
The displacement components can always be expressed as the sums of infinite series of products:

$$u_1 = \sum_{i=1}^{i=\infty} q_i \phi_{1i},$$
$$u_2 = \sum_{i=1}^{i=\infty} q_i \phi_{2i}, \qquad (1)$$
$$u_3 = \sum_{i=1}^{i=\infty} q_i \phi_{3i},$$

where the quantities q_i are an infinite set of scalars which are functions only of t, and ϕ_{1i}, ϕ_{2i}, ϕ_{3i} are the three components of an infinite set of independent, specified displacement forms and are functions only of x_1, x_2, x_3. It can be made clear that expressions of the form of (1) are possible for any displacement components u_1, u_2, u_3, by observing that the infinite set of quantities q_i can be considered as defined by the equations (1). The real significance of (1), however, is that the configuration of a system can be specified by a set of quantities q_i, which are functions only of t, through the use of a set of specified functions ϕ_{1i}, ϕ_{2i}, ϕ_{3i} of x_1, x_2, x_3. The time functions q_i will be called the generalized displacements of the system, and the space functions ϕ_{1i}, ϕ_{2i}, ϕ_{3i}, the generalized coordinates. The distinction between the term "generalized displacement" and "generalized coordinate" should be carefully noted; this distinction in terminology will be followed throughout this paper. The vector character of the generalized coordinates, which represent displacement configurations, should be noticed. It will be convenient to retain the dimension (length) of the displacements u_1, u_2, u_3, with the generalized coordinates ϕ_{1i}, ϕ_{2i}, ϕ_{3i}, so that the generalized displacements q_i will be dimensionless quantities. The use of generalized displacements and coordinates in describing the configuration of continuous mechanical systems has great advantages which will be discussed in the following paragraph:

Actually, a continuous mechanical system has an infinite number of degrees of freedom, and is theoretically capable of an infinite variety of displacement forms. Practically, experience has shown that only the very few of the possible displacement forms associated with the fundamental and the first few harmonic natural frequencies, that is, the first few "normal modes," are of importance. It is possible, then, in practical applications, to express the configuration of a system with sufficient accuracy by retaining only a very few terms of the infinite series of (1), provided that the generalized coordinates are chosen properly. The displacement components u_1, u_2, u_3, can then be expressed as finite sums of products, or, introducing matrix notation, as matrix products:

$$u_1 = \sum_{i=1}^{r} q_i \phi_{1i} = q'\phi_1 = \phi_1' q$$
$$u_2 = \sum_{i=1}^{r} q_i \phi_{2i} = q'\phi_2 = \phi_2' q \qquad (2)$$
$$u_3 = \sum_{i=1}^{r} q_i \phi_{3i} = q'\phi_3 = \phi_3' q$$



■ Fig. 2 – Wing-aileron system

where: q is the column matrix with the set of generalized displacements q_i as elements,
q' is the transposed row matrix of q,
ϕ_1, ϕ_2, ϕ_3 are the column matrices with the sets of functions ϕ_{1i}, ϕ_{2i}, ϕ_{3i}, respectively, as elements,
ϕ_1', ϕ_2', ϕ_3' are the transposed row matrices of ϕ_1, ϕ_2, ϕ_3, respectively.

In (2) the entire set of displacement components u_1, u_2, u_3, are specified in terms of a finite number r of generalized displacements. Thus, through the use of generalized displacements and coordinates, the problem of a continuous mechanical system, which theoretically has an infinite number of degrees of freedom, can be reduced to one which has only the few degrees of freedom which are of practical importance. Another advantage of the use of generalized displacements and coordinates is that, if the generalized displacements q of a system with whatever constraints or boundary conditions exist can be considered as independent, the equations of motion of the system can be expressed in the very simple and general Lagrangian form (see Equation (11) following).

The analytical advantages described in the preceding paragraph depend very largely upon a proper choice of generalized coordinates. First, in order to express the displacement components most satisfactorily with the fewest number r of generalized displacements, the r generalized coordinates should be chosen so that at least the approximate configuration of the first r normal modes can be expressed in terms of them. This can be done by choosing for each generalized coordinate a configuration which resembles a normal mode or a component of a normal mode. A knowledge of only the approximate shape of the first few normal modes or their components is sufficient for the proper choice of coordinates; with some experience, this can be obtained by inspection for a great majority of structures encountered. For example, the conventional airplane is made up of a wing, fuselage, and tail surfaces, each of which may be considered as a beam; the significant components of the normal modes of the whole airplane or of any part of it can be recognized by inspection if it be viewed as an assemblage of beams with various interconnections and support conditions. Second, to use the Lagrangian equations of motion each displacement must be independent; that is, it should be of such a nature that it can exist by itself without violating the conditions of restraint or the boundary conditions. Since each of the normal modes satisfies the restraint or boundary conditions, this restriction on the choice of the generalized coordinates is compatible with the one just discussed, and is, in fact, complementary. These two complementary requirements still leave considerable latitude in the choice of the actual functions, ϕ_{1i}, ϕ_{2i}, ϕ_{3i}, for the generalized coordinates. It will be found of great advantage in obtaining numerical results to choose functions of the simplest possible form; for example, the use of polynomials in one or more of the position coordinates x_1, x_2, x_3, will often provide simple functions which are satisfactory. The use of polynomials will be illustrated in an example.

For illustration, generalized displacements and coordinates will be used to specify the displacements of an elastically supported cantilever wing with an aileron. The wing semi-chord and root-to-tip distance are denoted respectively by b and s and the distance of the aileron control horn from the wing root, by x_A; these dimensions are indicated on the sketch of the wing-aileron system in Fig. 2. As shown in this figure, the system is referred to the axes of Fig. 1, with OX_1 running spanwise out from the support, OX_2 running aft from the support, and OX_3 running vertically up from the support; the axis OX_1 will be called the reference axis of the wing, and its intersection with each cross-section plane will be called the reference point of that cross-section. It will be assumed that the chordwise cross-sections of wing and aileron are rigid, and that only vertical and twisting movements of the wing and rotation

of the aileron about its hinge take place. The displacements of each cross-section of the wing-aileron system can, therefore, be specified by the three deflections,

h = downward movement of reference point,
α = rotation of wing chord, positive when angle of attack is increased,
β = rotation of aileron chord with respect to wing chord, positive in same sense as α,

which are indicated in Fig. 2. If the deflections h, α, β, are considered as functions of the reference axis coordinate x_1, and of time t, they specify completely the displacements of the whole system. For small motions the displacement components u_1, u_2, u_3, can be expressed linearly in terms of h, α, β, as follows:

$$\begin{aligned} u_1 &= 0, \\ u_2 &= x_3\alpha, \\ u_3 &= -h - x_2\alpha \qquad \text{(for wing proper)}, \\ u_3 &= -h - x_2\alpha - (x_2 - d)\beta \ \text{(for the aileron)}, \end{aligned} \tag{3}$$

where d is the coordinate x_2 of the aileron hinge. Because of the discontinuity in the deflection at the aileron hinge, separate functions are required to express u_3 for the wing proper and for the aileron. Restrictions imposed on the general displacement components u_1, u_2, u_3, by limiting the motions to those expressible by h, α, β, are made clear from (3) when it is remembered that h, α, β, are functions only of x_1 and t. Since h, α, β, are functions only of x_1 and t, it will be simplest to express them directly in terms of the generalized displacements; thus,

$$\begin{aligned} h &= \Sigma\, q_i\phi_{hi} = q'\phi_h = \phi_h' q, \\ \alpha &= \Sigma\, q_i\phi_{\alpha i} = q'\phi_\alpha = \phi_\alpha' q, \\ \beta &= \Sigma\, q_i\phi_{\beta i} = q'\phi_\beta = \phi_\beta' q. \end{aligned} \tag{4}$$

The notation in (4) is similar to that in (2). The generalized coordinates ϕ_{hi}, $\phi_{\alpha i}$, $\phi_{\beta i}$, in (4) are functions of the single variable x_1. It would be possible subsequently to express u_1, u_2, u_3, in terms of the generalized displacements by the use of (3) and (4).

It will be assumed that the cantilever support of the wing is rigid for all motions except a rotation about OX_2, and its stiffness with respect to this motion will be designated by K_h. Only the deflections resulting from the elastic yielding of the support, from the bending and twisting of the wing, and from deflection of the aileron control system will be considered. The elastic forces acting within the system can be expressed in terms of the deflections by the following equations:

$$\begin{aligned} &\text{Bending moment in wing} = EI\,\frac{\delta^2 h}{\delta x_1^2}, \\ &\text{Bending moment at support} = -K_h\left(\frac{\delta h}{\delta x_1}\right)_{x_1=0} \\ &\text{Torsional moment in wing} = GJ\,\frac{\delta\alpha}{\delta x_1}, \\ &\text{Aileron hinge moment} = -K_\beta\,(\beta)_{x_1=x_A}, \end{aligned} \tag{5}$$

where EI is the flexural rigidity of the wing beam,
GJ is the torsional rigidity of the wing beam (wing beam assumed to be of "torque-box" type),
K_h is the stiffness of the support in bending,
K_β is the stiffness of the aileron control system,

and the "flexural axis" of the wing is assumed to be straight and to coincide with the reference axis. The restraints or boundary conditions at the supported and at the free ends of the wing, expressed in terms of h and α, are:

$$\begin{aligned} \text{for } h:\ & x_1 = 0;\ h = 0;\ K_h\frac{\delta h}{\delta x_1} = EI_o\frac{\delta^2 h}{\delta x_1^2}; \\ & x_1 = s:\ \frac{\delta^2 h}{\delta x_1^2} = \frac{\delta^3 h}{\delta x_1^3} = 0; \end{aligned} \tag{6a}$$

$$\begin{aligned} \text{for } \alpha:\ & x_1 = 0,\ \alpha = 0 \\ & x_1 = s,\ \frac{\delta\alpha}{\delta x_1} = 0 \end{aligned} \tag{6b}$$

where EI_o is the flexural rigidity at $x_1 = 0$. It will be assumed that the aileron bends with the wing, but remains untwisted. Since the angle between the aileron chord and the fixed reference plane $X_1\ OX_2$ is $\alpha + \beta$, the condition that the aileron be untwisted is

$$\begin{aligned} &\alpha + \beta = \text{constant} \\ &\text{or } \beta = \text{constant} - \alpha. \end{aligned} \tag{6c}$$

Coupling between the bending, torsion, and aileron motions will, in general, cause their co-existence in each of the normal modes; nevertheless, these modes can be built up very approximately from components that resemble the normal modes which would occur in each single motion if there were no coupling. The approximate shapes of the first two pure bending modes of a flexural cantilever with a support which yields elastically in rotation are shown in Figs. 3a and b; the approximate shapes of the first two torsional modes of a torsional cantilever with rigid support are shown in Figs. 3c and d. Two functions which satisfy the boundary conditions (6a) and resemble respectively the shapes of the deflection forms of Figs. 3a and b, can be chosen for the components ϕ_{hi} of two of the generalized coordinates; simple functions which would be satisfactory are the polynomials:

$$\begin{aligned} P_1\left(\frac{x_1}{s}\right) &= 4\sigma\left(\frac{x_1}{s}\right) + 2\left(\frac{x_1}{s}\right)^2 - \frac{4}{3}\left(\frac{x_1}{s}\right)^3 + \frac{1}{3}\left(\frac{x_1}{s}\right)^4, \\ P_2\left(\frac{x_1}{s}\right) &= 3.2552\,\sigma\left(\frac{x_1}{s}\right) + 1.6276\left(\frac{x_1}{s}\right)^2 \\ &\quad - (13.8197\,\sigma + 4.1621)\left(\frac{x_1}{s}\right)^3 \\ &\quad + (13.8197\,\sigma + 3.3483)\left(\frac{x_1}{s}\right)^4 \\ &\quad - (4.1459\,\sigma + 0.9231)\left(\frac{x_1}{s}\right)^5 \end{aligned} \tag{7}$$

where $\sigma = \dfrac{EI_o}{K_h s}$ is a measure of the stiffness of the support. Similarly, two other generalized coordinates can be taken with components ϕ_{α_i} which satisfy the conditions (6b) and resemble the shapes of the deflection forms of Figs. 3c and d; again satisfactory polynomial functions can be found:

$$\begin{aligned} P_3\left(\frac{x_1}{s}\right) &= 2\left(\frac{x_1}{s}\right) - \left(\frac{x_1}{s}\right)^2, \\ P_4\left(\frac{x_1}{s}\right) &= 10\left(\frac{x_1}{s}\right) - 23\left(\frac{x_1}{s}\right)^2 + 12\left(\frac{x_1}{s}\right)^3. \end{aligned} \tag{8}$$

It is evident from (6c) that the only part of the aileron deflection function β which is independent of α is a con-

stant; the corresponding generalized coordinate would be:

$$\phi_{\beta i} = 1 \tag{9}$$

All configurations which can be built up by superposition of the deflection forms represented by (7), (8), and (9) can be expressed in the form (4) in terms of five generalized displacements which refer to the following five generalized coordinates:

$$\begin{array}{lclll}
\phi_{h1}, \phi_{\alpha 1}, \phi_{\beta 1} & = & P_1\left(\frac{x_1}{s}\right), 0 & , 0 & \\
\phi_{h2}, \phi_{\alpha 2}, \phi_{\beta 2} & = & P_2\left(\frac{x_1}{s}\right), 0 & , 0 & \\
\phi_{h3}, \phi_{\alpha 3}, \phi_{\beta 3} & = & 0 & , P_3\left(\frac{x_1}{s}\right), & -P_3\left(\frac{x_1}{s}\right) \\
\phi_{h4}, \phi_{\alpha 4}, \phi_{\beta 4} & = & 0 & , P_4\left(\frac{x_1}{s}\right), & -P_4\left(\frac{x_1}{s}\right) \\
\phi_{h5}, \phi_{\alpha 5}, \phi_{\beta 5} & = & 0 & , 0 & , 1.
\end{array} \tag{10}$$

From the process by which $P_1, \ldots P_4$, have been obtained it is evident that each of the generalized coordinates above satisfies the boundary conditions, and that the first few normal modes of the system can be very approximately expressed by them.

In the usual applications, and in all preliminary computations, it would be sufficient to use only three generalized coordinates to describe the motion of the system in Fig. 2; the "second order" coordinates ϕ_{h2}. $\phi_{\alpha 2}$. $\phi_{\beta 2}$ and ϕ_{h4}. $\phi_{\alpha 4}$. $\phi_{\beta 4}$. above will not ordinarily be of very great importance.

The lack of a direct calculation for obtaining the polynomials $P_1 \ldots P_4$ may cause some confusion; it is a result of the fact that the conditions which the functions must satisfy do not determine them completely, but leave considerable latitude in their choice. The same is true of the generalized coordinates in general. The latitude in the choice of the generalized coordinates is not an obstacle in choosing them, but actually presents the opportunity of taking suitable elementary functions which will simplify the subsequent mathematical manipulations. The choice of functions for the generalized coordinates is the step in this theory which requires the most judgment and care; it is also the step which makes possible the general treatment of vibration or flutter of any structure. Familiarity in the use of the theory will make the choice of the generalized coordinates easier and more direct.

III. Lagrange's Equations of Motion

The Lagrangian equations of small motion of a system about a position of equilibrium are:

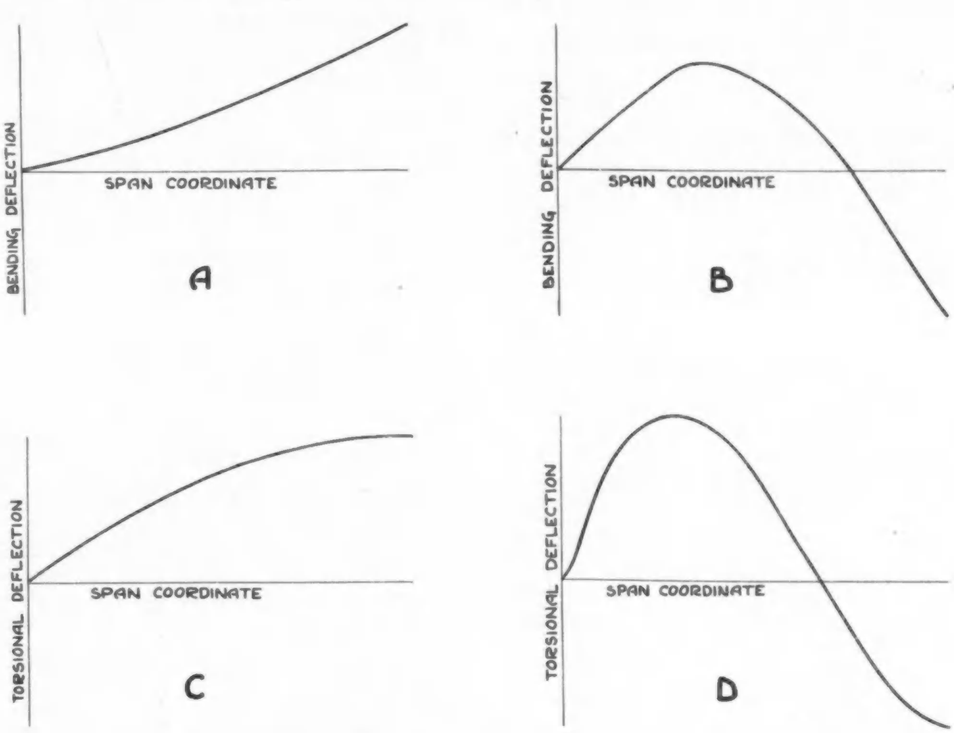
$$-\frac{d}{dt}\left(\frac{\delta T}{\delta \dot{q}_i}\right) - \frac{\delta V}{\delta q_i} + Q_i = 0, \ (i = 1, 2, \ldots r.), \tag{11}$$

where q_i = independent generalized displacements which specify the displacements of the system about an equilibrium position,

$$\dot{q}_i = \frac{\delta q_i}{\delta t},$$

T = kinetic energy of motion of system, which is a quadratic in $\dot{q}_i$ with constant coefficients as a result of the assumption of small displacements,

V = potential (elastic strain) energy of system, which is a quadratic in q_i with constant



■ Fig. 3 – Normal modes for flexural and torsional cantilevers

coefficients as a result of the assumptions of small displacements about an equilibrium position,

Q_i = generalized forces corresponding to externally applied forces.

There is one Lagrangian equation (11) for each displacement q_i; therefore, the motion of the system is completely determined by the full set of equations. Each equation expresses the equilibrium of all the forces on the system with respect to a generalized displacement q_i, and each term in the equations may be identified with a particular generalized force. The term $-\frac{d}{dt}\left(\frac{\delta T}{\delta \dot{q}_i}\right)$ is the generalized inertia force corresponding to q_i, the term $-\frac{\delta V}{\delta q_i}$ is the generalized elastic force, and in a flutter problem Q_i is the generalized air-force.

The generalized forces $Q_1, Q_2, \ldots Q_r$, corresponding to any set of forces acting on a system are defined as the coefficients of the differentials $\delta q_i, \delta q_2, \ldots \delta q_r$, which specify an arbitrary displacement of the system, in the expression for the work done by the forces during the arbitrary displacement. Thus, if δW denotes the work,

$$\delta W = Q_1 \delta q_1 + Q_2 \delta q_2 + \ldots\ldots Q_r \delta q_r \tag{12}$$

is the defining equation for Q_i. In matrix notation (12) is

$$\delta W = \delta q' Q \tag{13}$$

where $\delta q'$ is the row matrix with the displacement differentials δq_i as elements, and Q is the column matrix with the generalized forces Q_i as elements. Equation (13) will be used in Section VII, to obtain the generalized forces Q_i from the air-forces on a system.

IV. Inertia Forces

The inertia forces are expressed in Lagrange's equations (11) by the term $-\frac{d}{dt}\left(\frac{\delta T}{\delta \dot{q}_i}\right)$. To evaluate this term an expression is required for the kinetic energy of the system in terms of $\dot{q}_i$.

The kinetic energy is given by the following definite integral, which is extended over the whole system:

$$T = \frac{1}{2}\iiint (\dot{u}_1^2 + \dot{u}_2^2 + \dot{u}_3^2)\,\rho\, dx_1\, dx_2\, dx_3, \tag{14}$$

where ρ = mass density at the point x_1, x_2, x_3,

$$\dot{u}_i = \frac{\delta u_i}{\delta t}.$$

Expression (14) merely states that the total kinetic energy of the system is the sum of the kinetic energies of each element of mass $\rho dx_1\, dx_2\, dx_3$. Equations (2) are used to write (14) in terms of the generalized displacements. Differentiating (2) with respect to t according to the rules of matrix differentiation,

$$\begin{aligned} \dot{u}_1 &= \dot{q}'\phi_1 = \phi_1'\dot{q}, \\ \dot{u}_2 &= \dot{q}'\phi_2 = \phi_2'\dot{q}, \\ \dot{u}_3 &= \dot{q}'\phi_3 = \phi_3'\dot{q}, \end{aligned} \tag{15}$$

And

$$\begin{aligned} \dot{u}_1^2 &= \dot{q}'\phi_1\phi_1'\dot{q}, \\ \dot{u}_2^2 &= \dot{q}'\phi_2\phi_2'\dot{q}, \\ \dot{u}_3^2 &= \dot{q}'\phi_3\phi_3'\dot{q}. \end{aligned} \tag{16}$$

In (16) $\dot{q}$ and $\dot{q}'$ are respectively, column and row matrices, with $\dot{q}_i$ as elements, and the matrix products $\phi_i\phi_i'$ form square matrices of the following type:

$$\begin{bmatrix} \phi_{i1}\phi_{i1} & \phi_{i1}\phi_{i2} & \ldots & \phi_{i1}\phi_{ir} \\ \phi_{i2}\phi_{i1} & \phi_{i2}\phi_{i2} & \ldots & \phi_{i2}\phi_{ir} \\ \ldots & \ldots & \ldots & \ldots \\ \phi_{ir}\phi_{i1} & \phi_{ir}\phi_{i2} & \ldots & \phi_{ir}\phi_{ir} \end{bmatrix}$$

Substituting for $\dot{u}_1^2 + \dot{u}_2^2 + \dot{u}_3^2$ in (14) from (16), the expression for the kinetic energy becomes:

$$T = \frac{1}{2}\dot{q}'\iiint \rho\,(\phi_1\phi_1' + \phi_2\phi_2' + \phi_3\phi_3')\,dx_1dx_2dx_3\,\dot{q} \tag{17}$$

The matrices $\dot{q}'$ and $\dot{q}$ are taken outside the integral because they are not functions of the variables of integration. Since the elements of the matrix integrand $\rho\,(\phi_1\phi_1' + \phi_2\phi_2' + \phi_3\phi_3')$ are known functions of x_1, x_2, x_3, and since the triple integral has definite limits, the integration can be performed and results in a square symmetrical matrix with constant elements. Consideration of the dimensions of the integrand of (17) shows that all the elements of the matrix integral have the dimensions (force $\times$ length $\times$ (time)^2) of a moment of inertia; the integral of (17), therefore, can be expressed as follows:

$$\iiint \rho\,(\phi_1\phi_1' + \phi_2\phi_2' + \phi_3\phi_3')\,dx_1dx_2dx_3 = I_r a, \tag{18}$$

where I_r is a reference moment of inertia, and a is a square matrix with constant, dimensionless elements. Substitution of (18) into (17) gives the result:

$$T = \frac{1}{2}\dot{q}' I_r a\,\dot{q} \tag{19}$$

Expression (19) for the kinetic energy T is a quadratic in $\dot{q}_i$, the coefficients of which are defined by the symmetrical matrix $I_r a$; the matrix $I_r a$, in fact, describes the inertia characteristics of the system.

The terms $-\frac{d}{dt}\left(\frac{\delta T}{\delta \dot{q}_i}\right)$ of the dynamic equations (11) can be obtained from (19) by matrix differentiation; thus,

$$\begin{aligned} -\frac{d}{dt}\left(\frac{\delta T}{\delta \dot{q}_i}\right) &= -\frac{1}{2} I_r \frac{d}{dt}\left(\Sigma a_{ij}\dot{q}_j + \Sigma \dot{q}_j a_{ji}\right) = \\ -I_r \frac{d}{dt}\Sigma a_{ij}\dot{q}_j &= -I_r \Sigma a_{ij}\ddot{q}_j, \end{aligned} \tag{20}$$

where a_{ij} is the "i"th element in the "j"th column of a. The second equality in (20) results from the fact that $a_{ij} = a_{ji}$ for the symmetrical matrix a. The column of all the terms $-\frac{d}{dt}\left(\frac{\delta T}{\delta \dot{q}_i}\right)$ in the full set of equations (11) is, therefore,

$$-I_r a\,\ddot{q}, \tag{21}$$

where $\ddot{q}$ is the column matrix with $\ddot{q}_i = \frac{\delta^2 q_i}{\delta t^2}$ as elements.

It has been shown that the inertia characteristics of a system are described by the matrix $I_r a$, which is given by the definite integral (18). It has also been shown how the inertia terms in Lagrange's equations are obtained from this matrix. For illustration, the evaluation of $I_r a$ from (18) will be described for a wing-aileron system for which the displacements can be specified, as in the example of Section II, by equations of the type (4), (see Fig. 2).

The displacement components are first expressed in the

form (2) by substituting the expressions for h, α, β, from (4) into (3); thus,

$$\begin{aligned} u_1 &= 0, \\ u_2 &= q' [x_3\phi_a] = + [x_3\phi_a'] q \qquad (22) \\ u_3 &= - q' [\phi_h + x_2\phi_a] = - [\phi_h' + x_2\phi_a'] q \quad \text{(for the wing proper)}, \\ u_3 &= - q' [\phi_h + x_2\phi_a + (x_2-d)\phi_\beta] = - [\phi_h' + x_2\phi_a' + (x_2-d)\phi_\beta'] q \quad \text{(for the aileron)}, \end{aligned}$$

By comparing (22) with (2) the generalized coordinate components ϕ_1, ϕ_2, ϕ_3, are readily expressed in terms of the components ϕ_h, ϕ_a, ϕ_β, and x_2, x_3. With ϕ_1, ϕ_2, ϕ_3, the integral (18) can be set up; because of the two separate functions u_3 for wing and aileron this integral must be broken up into two parts. These integrals are:

$$\begin{aligned} & {}_W\!\iiint \rho\, [\phi_h\phi_h' + x_2 (\phi_a\phi_h' + \phi_h\phi_a') + (x_2^2 + x_3^2)\, \phi_a\phi_a']\, dx_1\, dx_2\, dx_3 \\ & + {}_A\!\iiint \rho\, [(x_2-d) (\phi_h\phi_\beta' + \phi_\beta\phi_h') + x_2 (x_2-d) (\phi_a\phi_\beta' + \phi_\beta\phi_a') + (x_2-d)^2 \phi_\beta\phi_\beta']\, dx_1\, dx_2\, dx_3 = I_r a. \end{aligned} \qquad (23)$$

In (23) and following the integral sign ${}_W\!\iiint$ or ${}_W\!\iint$ denotes integration over the entire wing-aileron system, while ${}_A\!\iiint$ or ${}_A\!\iint$ denotes integration over only the aileron. Since ϕ_h, ϕ_a, ϕ_β, are functions only of x_1, the integration with respect to x_2 and x_2 can be carried out immediately and expressed in terms of the following section mass distribution properties:

${}_W\!\iint \rho\, dx_2 dx_3 = m$ = mass of wing-aileron, per unit length of span;

${}_W\!\iint \rho x_2\, dx_2 dx_3 = m x_a$ = moment of wing-aileron mass about reference axis, per unit length of span;

${}_W\!\iint \rho\, (x_2^2 + x_3^2)\, dx_2 dx_3 = m r_a^2$ = moment of inertia of wing-aileron mass about reference axis, per unit length of span;

${}_A\!\iint \rho\, (x_2 - d)\, dx_2 dx_3 = m_\beta x_\beta$ = moment of aileron mass about aileron hinge, per unit length of span;

${}_A\!\iint \rho\, (x_2 - d)^2 dx_2 dx_3 = m_\beta r_\beta^2$ = moment of inertia of aileron mass about hinge, per unit length of span;

${}_A\!\iint \rho x_2 (x_2 - d)\, dx_2 dx_3 = m_\beta (r_\beta^2 + d x_\beta)$ = sum of mass moment of inertia of aileron and product of moment of aileron about hinge line by distance between hinge and reference axis, per unit length of span.

With the notation adopted above for the double integrals (23) becomes:

$$\int_0^s [m\phi_h\phi_h' + m x_a (\phi_a\phi_h' + \phi_h\phi_a') + m r_a^2 \phi_a\phi_a' + m_\beta x_\beta (\phi_h\phi_\beta' + \phi_\beta\phi_h') + m_\beta (r_\beta^2 + x_\beta d) (\phi_a\phi_\beta' + \phi_\beta\phi_a') + m_\beta r_\beta^2 \phi_\beta\phi_\beta']\, dx_1 = I_r a \qquad (24)$$

The evaluation of each element of the inertia matrix $I_r a$ from (24) involves only the computation of a series of definite simple integrals.

V. Elastic Forces

The elastic forces are expressed by the term $-\frac{\delta V}{\delta q_i}$ in Lagrange's equations of motion. An expression for the potential energy V in terms of the generalized displacements q_i is necessary to evaluate this term.

[8] See "Theory of Elasticity," by S. Timoshenko, McGraw-Hill Book Co., 1934.

The potential, or elastic strain energy in any elastic system can be expressed in terms of the strains; in reference (8) it is given by the following definite integral extended over the whole system:

$$V = \frac{1}{2}\iiint [(\psi + 2G)\, (\epsilon_1^2 + \epsilon_2^2 + \epsilon_3^2) + 2\psi\, (\epsilon_1\epsilon_2 + \epsilon_2\epsilon_3 + \epsilon_3\epsilon_1) + G\, (\gamma_{12}^2 + \gamma_{23}^2 + \gamma_{13}^2)]\, dx_1 dx_2 dx_3, \qquad (25)$$

where ϵ_1 = longitudinal strain parallel to x_1 axis $= \frac{\delta u_1}{\delta x_1}$,

ϵ_2 = longitudinal strain parallel to x_2 axis $= \frac{\delta u_2}{\delta x_2}$,

ϵ_3 = longitudinal strain parallel to x_3 axis $= \frac{\delta u_3}{\delta x_3}$,

γ_{12} = shear strain parallel to x_1x_2 plane $= \frac{\delta u_1}{\delta x_2} + \frac{\delta u_2}{\delta x_1}$,

γ_{13} = shear strain parallel to x_1x_3 plane $= \frac{\delta u_1}{\delta x_3} + \frac{\delta u_3}{\delta x_1}$,

γ_{23} = shear strain parallel to x_2x_3 plane $= \frac{\delta u_2}{\delta x_3} + \frac{\delta u_*}{\delta x_2}$,

ν = poisson's ratio,

E = Young's elastic modulus,

$G = \frac{E}{2(1+\nu)}$ = shear modulus,

$\psi = \frac{E}{(1+\nu)(1-2\nu)}$.

The strains have just been expressed in terms of the displacement derivatives; these, in turn, can be expressed in terms of the generalized displacements and coordinates by (2). Expression (25) for the potential energy, written in terms of the generalized displacements and coordinates, is:

$$\begin{aligned} V = \frac{1}{2} q' \Big\{ \iiint \Big[& (\psi + 2G) \Big(\frac{\delta\phi_1}{\delta x_1} \frac{\delta\phi_1'}{\delta x_1} + \frac{\delta\phi_2}{\delta x_2} \frac{\delta\phi_2'}{\delta x_2} + \frac{\delta\phi_3}{\delta x_3} \frac{\delta\phi_3'}{\delta x_3} \Big) + \psi \Big(\frac{\delta\phi_1}{\delta x_1} \frac{\delta\phi_2'}{\delta x_2} + \frac{\delta\phi_2}{\delta x_2} \frac{\delta\phi_1'}{\delta x_1} + \frac{\delta\phi_2}{\delta x_2} \frac{\delta\phi_3}{\delta x_3} \\ & + \frac{\delta\phi_3}{\delta x_3} \frac{\delta\phi_2'}{\delta x_2} + \frac{\delta\phi_3}{\delta x_3} \frac{\delta\phi_1'}{\delta x_1} + \frac{\delta\phi_1}{\delta x_1} \frac{\delta\phi_3'}{\delta x_3} \Big) \\ & + G \Big(\frac{\delta\phi_1}{\delta x_2} \frac{\delta\phi_1'}{\delta x_2} + \frac{\delta\phi_2}{\delta x_1} \frac{\delta\phi_2'}{\delta x_1} + \frac{\delta\phi_1}{\delta x_3} \frac{\delta\phi_1'}{\delta x_3} + \frac{\delta\phi_3}{\delta x_1} \frac{\delta\phi_3}{\delta x_1} \\ & + \frac{\delta\phi_2}{\delta x_3} \frac{\delta\phi_2'}{\delta x_3} + \frac{\delta\phi_3}{\delta x_2} \frac{\delta\phi_3'}{\delta x_2} + \frac{\delta\phi_2}{\delta x_1} \frac{\delta\phi_1'}{\delta x_2} + \frac{\delta\phi_1}{\delta x_2} \frac{\delta\phi_2'}{\delta x_1} \\ & + \frac{\delta\phi_1}{\delta x_3} \frac{\delta\phi_3'}{\delta x_1} + \frac{\delta\phi_3}{\delta x_1} \frac{\delta\phi_1'}{\delta x_3} + \frac{\delta\phi_2}{\delta x_3} \frac{\delta\phi_3'}{\delta x_2} \\ & + \frac{\delta\phi_3}{\delta x_2} \frac{\delta\phi_2'}{\delta x_3} \Big) \Big] dx_1\, dx_2\, dx_3 \Big\} q \end{aligned} \qquad (26)$$

where the integrand is a square matrix made up of the sum of square matrices of the form $\frac{\delta\phi_i}{\delta x_j} \frac{\delta\phi_k'}{\delta x_1}$ multiplied by scalars. The integration in (26) can be carried out since the elements of the matrices ϕ are known functions of x_1, x_2, x_3. The following expression for the potential energy results from the integration:

$$V = \frac{1}{2} q' K_r f q, \qquad (27)$$

where $K_r f$ is the square symmetrical matrix with constant elements, which are the definite integrals of the elements of the matrix integrand of (26) over the whole system. Consideration of the dimensions of the quantities in (26)

shows that each of the elements of $K_r f$ has the dimensions (length $\times$ force) of an elastic stiffness; the elements are all referred to a reference stiffness K_r which may be considered a scalar multiplier of the dimensionless matrix f. The coefficients of the quadratic expression (27) in q_i are defined by the matrix $K_r f$ which describes the elastic properties of the system.

The typical term $-\dfrac{\delta V}{\delta q_i}$ in the dynamic equations (11) is obtained by differentiation of (27) with respect to q_i,

$$-\frac{\delta V}{\delta q_i} = -\frac{1}{2} K_r\,(\Sigma f_{ij} q_j + \Sigma q_j f_{ji}) = -K_r f_{ij} q_j, \tag{28}$$

where f_{ij} is the "i"th element in the "j"th column of f. The column of the terms $-\dfrac{\delta V}{\delta q_i}$ appearing in the full set of equations (11) is:

$$-K_r f q \tag{29}$$

Before taking up an example of the evaluation of the elastic matrix $K_r f$ it should be pointed out that for the great majority of structures the use of the very lengthy integral (26) is not necessary. The example chosen later will illustrate this point. The general expression (26) for the potential energy of any elastic system has been included more for demonstrating the complete generality of the form (27) than for actual use.

As an example, the evaluation of the matrix $K_r f$ will be described for the wing-aileron system of the example in Section II, for which the displacements can be specified by (4) and the elastic forces, by (5), (see Fig. 2). Instead of the general expression (25) for the potential energy it will be simpler to use the expressions for the potential energies of bending and twisting of a beam, and for the yielding of elastic supports; thus:

$$V = \frac{1}{2}\left\{\int_0^s \left[EI\left(\frac{\delta^2 h}{\delta x_1^2}\right)^2 + GJ\left(\frac{\delta \alpha}{\delta x_1}\right)^2\right] dx_1 + K_h\left(\frac{\delta h}{\delta x_1}\right)^2_{x_1 = 0} + K_\beta(\beta)^2_{x_1 = x_A}\right\}. \tag{30}$$

Substituting for h, α, β, from (4) the potential energy can be expressed in terms of the generalized displacements as follows:

$$V = \frac{1}{2} q' \left\{\int_0^s \left[EI\,\frac{\delta^2\phi_h}{\delta x_1^2}\,\frac{\delta^2\phi_h'}{\delta x_1^2} + GJ\,\frac{\delta\phi_\alpha}{\delta x_1}\,\frac{\delta\phi_\alpha'}{\delta x_1}\right] dx_1 + K_h\left(\frac{\delta\phi_h}{\delta x_1}\right)_{x_1 = 0} \times \left(\frac{\delta\phi_h'}{\delta x_1}\right)_{x_1 = 0} + K_\beta(\phi_\beta)_{x_1 = x_A} \times (\phi_\beta')_{x_1 = x_A}\right\} q = \frac{1}{2} q' K_r f q \tag{31}$$

The matrix $K_r f$ is the quantity within the curved brackets. Each of its elements is the sum of a simple definite integral and two simple products; the evaluation of the matrix $K_r f$ is, therefore, straightforward.

VI. Equations of Motion with No Air-Forces

Lagrange's equations of motion in the absence of air-forces ($Q_i = 0$) can be set up using the inertia and elastic terms (21) and (29); in matrix notation the equations are:

$$-I_r a\ddot{q} - K_r f q = 0. \tag{32}$$

The solutions of these equations represents the free motions of a linear conservative system, and correspond approximately to the modes and frequencies observed in still-air vibration tests. A study of these solutions will indicate the use of still-air vibration tests in flutter investigations.

The solutions of equations (32) of a linear conservative system are discussed in detail in References 5, 6, and 7. Those of Reference 7, in matrix notation, are particularly concise and correspondingly intelligible; for that reason, they will be used for the present discussion. The reader is referred to Chapter 9 of Reference 7 for detailed derivations; here the solution will simply be re-stated. The complete solution of (32) is represented by

$$q = zM(t)\xi + zM(-t)\eta,$$

(Reference 7, Chapter 9, Section 10) (33)

where ξ and η are two columns each containing r arbitrary constants of integration, z is a square matrix with constant elements, and $M(t)$ is the diagonal matrix:

$$\begin{bmatrix} e^{\lambda_1 t} & 0 & \cdots & 0 \\ 0 & e^{\lambda_2 t} & \cdots & 0 \\ \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & \cdots & e^{\lambda_r t} \end{bmatrix}$$

The quantities $\lambda_1, \lambda_2, \ldots \lambda_r$ are the roots of the determinental equation,

$$|\,I_r a\lambda^2 + K_r f\,| = 0 \tag{34}$$

When equation (34) represents a linear conservative system with the displacements measured from a position of stable equilibrium, the roots are all pairs of imaginaries and are proportional to the natural frequencies of the system; thus, if $\omega_1, \omega_2, \ldots \omega_r$ are the natural frequencies,

$$i\omega_1 = \pm\lambda_1\,;\quad i\omega_2 = \pm\lambda_2\,;\quad \ldots\ldots\quad i\omega_r = \pm\lambda_r, \tag{35}$$

where $i = \sqrt{-1}$.

The complete solution (33) is the sum of r independent solutions each having the form,

$$q = \xi_j z_j e^{\lambda_j t} + \eta_j z_j e^{-\lambda_j t},\ (j = 1, 2 \ldots r), \tag{36}$$

where z_j represents the "j"th column of z, and ξ_j, η_j are respectively the "j"th elements of the columns ξ, η. Since, from (35),

$$\lambda_j = i\omega_j,$$

and

$$e^{\lambda_j t} = e^{i\omega_j t} = \cos\omega_j t + i\sin\omega_j t,$$

$$e^{-\lambda_j t} = e^{-i\omega_j t} = \cos\omega_j t - i\sin\omega_j t,$$

it is possible to rewrite (36) as

$$q = \zeta_j z_j \cos\omega_j t + \mu_j z_j \sin\omega_j t \tag{37}$$

where ζ_j and μ_j are arbitrary constants derived from ξ_j, η_j. Each of the parts (37) of the complete solution evidently represents a "resonant" or normal mode of vibration; the frequency ω_j is the resonant frequency and the column z_j describes the mode.

A new set of generalized displacements p can be defined in terms of q by the linear set of equations

$$q = zp \tag{38}$$

where z is the square matrix of the complete solution (33). Equation (32) written in terms of p transforms into

$$I_r\,[z'az]\,\ddot{p} + K_r\,[z'fz]\,p = 0, \tag{39}$$

(Reference 7, Chapter 9, Section 3)

or

$$I_r A\ddot{p} + K_r F p = 0,$$

where z' is the transposed matrix of z. The matrices

$z'az = A$ and $z'fz = F$ are diagonal (Reference 7, Chapter 9, Section 10); therefore the expanded form of (39) is the following set of independent equations in each generalized displacement p_i,

$$I_r A_{ii} \ddot{p}_i + K_r F_{ii} p_i = 0 \quad (i = 1, 2, \ldots r) \tag{40}$$

It is evident from (38) that each of the new generalized displacements p_i represents a normal mode of the system; the independent of equations (40) shows that each normal mode operates as a single degree of freedom. The normal modes p_i are those ordinarily observed in still-air vibration tests.

Ground Vibration Tests – As mentioned previously, there is a full correspondence between the solutions of the equations of motion with air-forces absent and the frequencies and modes observed in still-air vibration tests; such tests, therefore, will reflect the mass and elastic characteristics of a system. However, since the equations representing the actual flutter must include the air-forces, the correspondence between vibration test results and flutter characteristics is incomplete. There are, nevertheless, important uses for still-air vibration test results which will be described in following paragraphs.

There is a considerable amount of work involved in obtaining the matrices $I_r a$ and $K_r f$ for a particular system; not only is there the actual evaluation of the elements of these matrices, but also the choice of adequate generalized coordinates. A comparison of the resonant modes and frequencies observed in still-air vibration tests with the normal modes and frequencies of the solution (33) will provide an experimental check of the work thus far.

It is also possible to obtain values for the internal friction damping of structures from still-air vibration tests. Theodorsen, in Reference 4, has indicated that the internal friction forces of a structure are approximately proportional to amplitude and independent of frequency. A friction force which is proportional to amplitude and independent of frequency can be included in the normal form (40) of the dynamic equations by adding an imaginary or out-of-phase component to the stiffness term $K_r F_{ii}$. Actually, the existence of the independent normal forms (40) with friction forces included in the dynamic equations depends upon an assumption concerning the form of the "dissipative" function of Reference 5 from which the friction forces can be derived. Such an assumption will be made; the effects and justification of it are discussed in Reference 9. With this assumption the typical equation of the normal set of dynamic equations with friction included is:

$$I_r A_{ii} \ddot{p}_i + K_r F^o_{ii} p_i = 0 \tag{41}$$

where $F^o_{ii} = F_{ii}(1 + g_{ii})$, and g_{ii} is the friction or damping factor for the mode p_i. Methods of obtaining the damping factors g_{ii} from vibration tests are described in Reference 9.

The equations of motion in the original generalized displacements q_i, with the friction forces included, may be obtained by transforming the equations typified by (41) to those in q_i. From (38),

$$p = z^{-1}q, \tag{42}$$

where $[z^{-1}]$ is the reciprocal matrix of z. (See Reference 7 for the concept of "reciprocal matrix.") The transformation results in

$$- I_r a \ddot{q} - K_r f^o q = 0, \tag{43}$$

where $f^o = [z^{-1}]' F^o [z^{-1}]$, and F^o is the diagonal matrix with the complex stiffnesses F^o_{ii} of (41) as elements. Equation (43) is the counter-part of (32) with the internal friction forces included.

VII. Air-Forces

The air-forces entering an ordinary flutter theory are the increments of air-force caused by deformations of the surface of a system in a uniform airstream. The "static" air-forces which act on the undeformed system, which are ordinarily used in airplane performance and strength calculations, do not enter into the flutter phenomenon; the static air-forces can be considered to be continuously nullified by the elastic forces of the system in its equilibrium state. Only the air-force increments, or "dynamic" air-forces resulting from the unsteady flow caused by changing deformations of the surface occur in the equations of flutter. The determination of the dynamic air-forces is made so complicated by the unsteady flow condition that only a few solutions for particular surfaces are available; some of these solutions will be enumerated in the next paragraph.

Theodorsen, in Reference 2, has derived expressions for the air-forces for two-dimensional flow over a rigid airfoil with an aileron which is executing any oscillation with constant amplitude that consists of vertical movement, rotation of the airfoil as a whole, and rotation of the aileron about its hinge. Equivalent expressions have been derived subsequently by Cicala, Kassner and Fingado, and Kussner. Jones, in Reference 10, has derived approximate expressions for the lift on an oscillating rigid airfoil with a finite aspect ratio. Cicala has described an approximate method in Reference 11 for calculating the air-forces for three-dimensional flow over a flexible airfoil of finite span in any constant-amplitude oscillation; the computations are, however, extremely laborious. Possio, in Reference 12, has obtained the forces on a rigid oscillating airfoil for two-dimensional flow of a compressible fluid. Flutter speeds can be predicted from the air-forces given by the solutions just mentioned, which are all for constant-amplitude oscillations, since the oscillations which would occur at the flutter speed[13] would be of constant amplitude.

The air-forces are introduced into the equations of motion by the generalized force term Q. Since a general expression for the air-forces is not available, the method of expressing them in the form of generalized forces will be illustrated by an example: Generalized forces corresponding to the air-forces given in Reference 2 will be obtained for the wing-aileron system described in Section II and shown in Fig. 2. The airflow over airplane wings and tail surfaces having the usual aspect ratios is approximately two-dimensional; therefore, within certain limits of error, the flutter of these surfaces may be treated by interpreting the air-forces of Reference 2 as section forces.

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[9] See NACA Technical Note No. 751, 1940: "Damping Formuias and Experimental Values of Damping in Flutter Models," by Robert P. Coleman.

[10] See NACA Technical Report No. 681, 1940· "The Unsteady Lift of a Wing of Finite Aspect Ratio," by Robert T. Jones.

[11] See NACA Technical Memorandum No. 887, 1939, translated from *L'Aerotecnica,* Vol. 18, No. 4, April, 1938: "Comparison of Theory with Experiment in the Phenomenon of Wing Flutter," by P. Cicala.

[12] See *L'Aerotecnica,* Vol. 18, No. 4, April, 1938: "L'azione Aerodinamica sul Profilo Oscillante in un Fluido Compressible a Velocita Iposonora," by Camillo Possio.

[13] The flutter speed is defined as the speed at which the total damping of a system under the action of air-forces (and the inertia, elastic, and friction forces) changes from positive to negative; at this speed the damping is zero so that sustained oscillations would occur.

Within these limits of error the generalized air-force expressions derived in the following would be applicable for any conventional airplane wing or tail surface.

The most general motions for which the air-forces of Reference 2 are applicable can be expressed in terms of the deflections h, α, β (see Fig. 2); the air-forces in terms of these deflections, from Reference 2, are:

$$\begin{aligned} P &= -\pi\rho\omega^2 b^2 (A_{ch}\, h + bA_{c\alpha}\,\alpha + bA_{c\beta}\,\beta), \\ M_\alpha &= -\pi\rho\omega^2 b^3 (A_{\alpha h}\, h + bA_{\alpha\alpha}\,\alpha + bA_{\alpha\beta}\,\beta), \\ M_\beta &= -\pi\rho\omega^2 b^3 (A_{bh}\, h + bA_{b\alpha}\,\alpha + bA_{b\beta}\,\beta), \end{aligned} \tag{44}$$

where P = downward air-force, per unit length of span,

M_α = stalling air-moment about reference point on whole airfoil, per unit length of span,

M_β = stalling air-moment on aileron about aileron hinge, per unit length of span,

b = semi-chord of wing at current section,

ρ = mass density of air,

ω = frequency of oscillation,

$A_{ij} = R_{ij} + I_{ij}\sqrt{-1}$, where R_{ij} and I_{ij} have the values given by the expressions on page 12 of Reference 2, with $\frac{1}{\kappa} = 0$, and the terms $\Omega_\alpha X$, $\Omega_h X$, $\Omega_\beta X$, omitted.

For a particular section the quantities A_{ij} are functions of the parameter,

$$k = \frac{\omega b}{v},$$

where v is the air-speed of the system. The air-forces over a whole system are characterized by the ratio $\frac{\omega}{v}$, or by the dimensionless parameter,

$$k_o = \frac{\omega b_o}{v}, \tag{45}$$

where b_o is a reference length. The parameter k for each section is given in terms of k_o by

$$k = k_o\theta, \text{ where } \theta = \frac{b}{b_o} \tag{46}$$

The problem of determining the generalized forces from the air-forces given by (44) will now be considered. Since the generalized forces are defined by the work expressions (12) or (13), the work done by the air-forces P, M_α, M_β, during the arbitrary deflection δh, $\delta\alpha$, $\delta\beta$ will be used as a starting point. The expression for this work is:

$$\delta W = \int_0^s (P\delta h + M_\alpha \delta\alpha + M_\beta \delta_\beta)\, dx_1,$$

or, expressing the integrand as a matrix product:

$$\delta W = \int_0^s \begin{bmatrix} \frac{\delta h}{b_o}, & \delta\alpha, & \delta\beta \end{bmatrix} \begin{bmatrix} Pb_o \\ M_\alpha \\ M_\beta \end{bmatrix} dx_1 \tag{47}$$

The column of the force components in (47) can be expressed from (44) as the product of a square and a column matrix; thus,

$$\begin{bmatrix} Pb_o \\ M_\alpha \\ M_\beta \end{bmatrix} = -\pi\rho\omega^2 b_o^4 \Lambda \begin{bmatrix} \frac{h}{b_o} \\ \alpha \\ \beta \end{bmatrix}, \tag{48}$$

where Λ is the square matrix,

$$\begin{bmatrix} \theta^2 A_{ch} & \theta^3 A_{c\alpha} & \theta^3 A_{c\beta} \\ \theta^3 A_{\alpha h} & \theta^4 A_{\alpha\alpha} & \theta^4 A_{\alpha\beta} \\ \theta^3 A_{bh} & \theta^4 A_{b\alpha} & \theta^4 A_{b\beta} \end{bmatrix}; \quad \theta = \frac{b}{b_o} \tag{49}$$

The column of deflections of (48) can be written in terms of the generalized displacements and coordinates by expressing (4) as the following matrix product:

$$\begin{bmatrix} \frac{h}{b_o} \\ \alpha \\ \beta \end{bmatrix} = \phi_{h\alpha\beta}' q, \tag{50}$$

where q is the column matrix with the r generalized displacements q_i as elements, and $\phi_{h\alpha\beta}'$ is the three row rectangular matrix,

$$\begin{bmatrix} \phi_{h1}/b_o & \phi_{h2}/b_o & \dots & \phi_{hr}/b_o \\ \phi_{\alpha1} & \phi_{\alpha2} & \dots & \phi_{\alpha r} \\ \phi_{\beta1} & \phi_{\beta2} & \dots & \phi_{\beta r} \end{bmatrix} = \phi_{h\alpha\beta} \tag{51}$$

Similarly, the row matrix $[\delta h/b_o, \delta\alpha, \delta\beta]$ of the arbitrary deflections can be represented by the matrix product,

$$[\delta_h/b_o,\ \delta\alpha,\ \delta\beta] = \delta q'\, \phi_{h\alpha\beta} \tag{52}$$

where $\delta q'$ is the row matrix of arbitrary generalized displacements δq_i, and $\phi_{h\alpha\beta}$ is the transposed three column matrix of $\phi_{h\alpha\beta}'$ in (51). The elements of the matrices $\phi_{h\alpha\beta}$ and $\phi_{h\alpha\beta}'$ are dimensionless.

Substituting (50) into (48) and (48) and (52) into (47), the work δW is expressed,

$$\delta W = \delta q' \left[-\pi\rho\omega^2 b_o^4 \int_0^s \phi_{h\alpha\beta} \Lambda \phi_{h\alpha\beta}'\, dx_1 \right] q.$$

The matrix integrand $\phi_{h\alpha\beta}\Lambda\phi_{h\alpha\beta}'$ is a square matrix of order r the elements of which, for a particular value of k_o, are all known functions of x_1. To make the integral dimensionless the variable of integration will be changed from x_1 to (x_1/s); thus;

$$\delta W = \delta q' \left[-\pi\rho\omega^2 b_o^4 s \int_0^1 \phi_{h\alpha\beta} \Lambda \phi_{h\alpha\beta}'\, d\left(\frac{x_1}{s}\right) \right] q \tag{53}$$

The integration in (53) can be carried out and will result in the following expression for the work:

$$\delta W = \delta q' \left[-\pi\rho\omega^2 b_o^4 sc \right] q\ ; \quad c = \int_0^1 \phi_{h\alpha\beta} \Lambda \phi_{h\alpha\beta}'\, d\left(\frac{x_1}{s}\right), \tag{54}$$

where c is a square matrix of order r, the elements of which are complex dimensionless functions of k_o; and $-\pi\rho\omega^2 b_o^4 s$ is a scalar dimensional multiplier. Comparing (54) with (13) the column of generalized air-forces Q for the complete set of dynamic equations is seen to be,

$$Q = -\pi\rho\omega^2 b_o^4 scq. \tag{55}$$

The matrix $-\pi\rho\omega^2 b_o^4 s\, c$ describes the dynamic air-forces acting on the system during an oscillation of frequency ω, and c is, for a particular system, a function only of k_o.

VIII. Equations with Air-Forces Included

The solution of the equations of motion with the air-forces of Reference 2 will be considered in this section; solutions of the type described should be applicable for wing or tail-surface flutter of conventional airplanes. Substituting (21), (29), and (55) into (11), the matrix form of the equations of motion is

$$-I_r a\ddot{q} - (K_r f + \pi\rho\omega^2 b_o^4 sc)\, q = 0 \tag{56}$$

In (56) q and $\ddot{q} = \frac{\delta^2 q}{\delta t^2}$ are matrix columns of the generalized displacements and their second time derivatives; a, f, and c, are square matrices with dimensionless elements; and I_r, K_r, $\pi\rho\omega^2 b_o^4 s$ are dimensional scalar multipliers.

In seeking a solution of equations (56) consideration

must first be given the fact that the air-force term $-\pi\rho\omega^2 b_0{}^4 sc$ is valid only for an oscillation of constant amplitude. The motion implied by (56), therefore, must be a sustained oscillation, and can be represented by

$$q = ye^{i\omega t} \tag{57}$$

where y is a column of constants. From (57)

$$\ddot{q} = -\omega^2 ye^{i\omega t} \tag{58}$$

Substituting for q and $\ddot{q}$ in (56) from (57) and (58), and dividing through by $\pi\rho\omega^2 b_0{}^4 s\, e^{i\omega t}$, the equation of motion becomes

$$[-\Gamma a + c + \Upsilon f]\, y = 0 \tag{59}$$

$$\text{where } \Gamma = \frac{I_r}{\pi\rho b_0{}^4 s}$$

$$\Upsilon = \frac{K_r}{\pi\rho b_0{}^4 s\omega^2} = \left(\frac{\omega_r}{\omega}\right)^2$$

$$\omega_r{}^2 = \frac{K_r}{\pi\rho b_0{}^4 s} = \text{reference frequency.}$$

Equation (59) is the equation of motion of a system at a flutter speed[14]; the determinental equation which is the condition for the existence of a flutter speed is

$$|-\Gamma a + c + \Upsilon f| = 0 \tag{60}$$

For a particular system k_0 (see equation (45)) and Υ are the only variables in (60); the values of these two variables appropriate to a flutter speed are obtained from the two conditions that the real and the imaginary parts of the determinant (60) are zero. The flutter speed v and the flutter frequency ω can be computed from k_0 and Υ by the relations

$$\begin{aligned} \omega &= \omega_r/\sqrt{\Upsilon}\,, \\ v &= \omega b_0/k_0 = \omega_r b_0/k_0\sqrt{\Upsilon} \end{aligned} \tag{61}$$

The complex elements of the matrix c are such complicated functions of k_0 that the direct solution for k_0 just indicated involves a very laborious trial-and-error process. A more direct method of solution, which is particularly suited to the usual problem of finding the effect of changes in design on the flutter speed, is to assign successive values to k_0 and to treat one or more of the elements of the matrices a or f as variables. Any change in stiffness or shift of weight in the system can be expressed in terms of its effect on the elements of these matrices. The expanded form of (60) is linear in each matrix element and is a polynomial in Υ; direct solutions for Υ and the variable matrix elements from the expanded polynomial equations can usually be devised. A series of such solutions for several values of k_0 will, through the use of (61), yield points on a plot of the flutter speed with some variable design feature of a system.

In some cases, particularly when there are a number of generalized coordinates q, it may be advantageous to re-write (56) in terms of the generalized displacements p which are given by (38). Equation (56) transforms to

$$-A\ddot{p} - (K_r F + \pi\rho\omega^2 b_0{}^4 sC)\, p = 0, \tag{62}$$

where (see equation (39)) A and F are diagonal matrices, and C is the matrix $z'cz$. Setting

$$q = we^{i\omega t} \tag{63}$$

the transformed form of (59) is

$$[-\Gamma A + \Upsilon F + C]\, w = 0. \tag{64}$$

The expanded form of equations (59) or (64) is a set of linear simultaneous equations[15] in the elements y_i or w_i of the columns y or w; the expanded form of (64) is:

$$\begin{array}{lllll} (-\Gamma A_{11} + \Upsilon F_{11} + C_{11})w_1 & + C_{12}w_2 + \ldots\ldots & & + C_{1r}w_r = 0 \\ C_{21}\, w_1 & + (-\Gamma A_{22} + \Upsilon F_{22} + C_{22})\, w_2 + \ldots\ldots & & + C_{2r}w_r = 0 \\ \ldots\ldots & \ldots\ldots & \ldots\ldots & \ldots\ldots \\ C_{r1}\, w_1 & C_{r2}w_2 + & \ldots\ldots + (-\Gamma A_{rr} + \Upsilon F_{rr} + C_{rr})w_r & = 0 \end{array} \tag{65}$$

In the form (65) the variable Υ occurs only in the single coefficient of each equation which lies on the principle diagonal; this form is particularly suitable for determining by inspection the effects of a change in Υ on the set of equations. For this reason, it usually becomes apparent upon inspection if, as sometimes happens, one or more groups of equations (65) become effectively independent of the others for certain ranges of k_0 and Υ. In this event the transformed form of the determinental equation (60),

$$|-\Gamma A + \Upsilon F + C| = 0 \tag{66}$$

can be split up into several lower order determinental equations which are much easier to handle. Displacement forms which have little or no effect on flutter can be eliminated from the computations in this way; and, if groups of displacement forms tend to act together to the exclusion of the others, these groups may be treated separately.

■ Concluding Remarks on Applications

Equations have been derived which can be used to determine the flutter speed of the wing or tail surfaces of any conventional airplane. Since the process by which these equations have been obtained is perfectly general (within the limits of a linear theory), the equations can be corrected or extended when more accurate or more general air-force solutions become available. There is the possibility, in using this approach to the flutter problem, of considerable labor saving by standardization of the work; a development along this line in applications to the aircraft flutter problem, which the author anticipates, is briefly outlined in the following paragraph:

A standard set of generalized coordinates will be devised which will serve to describe adequately the configuration of any conventional airplane. The difficult step of choosing adequate generalized coordinates would then be accomplished once and for all. Definite integrals of the type (24), (31), and (53), will be carried out in terms of parameters by which simple standard types of wing plan-forms and mass and stiffness distributions have been expressed. This preliminary work will make for the rapid determination of the matrices a, c and f of (59). With the standard generalized coordinates, these matrices, in addition to forming the flutter equation (59), will provide a very complete basis of comparison of airplanes.

[14] See the definition of flutter speed given in Footnote 13.
[15] See Reference 7, Chapter I, Section 4, example (vi).

DISCUSSION

Points Out Several Unanswered Questions

—Newman A. Hall

Vought-Sikorsky Aircraft, Division of United Aircraft Corp.

MR. LORING has performed an excellent service in applying to the flutter problem the principles of generalized coordinates. In the discussion of dynamical systems this is, of course, a well-established means of getting at essential elements of the system and, for example, as is shown in the paper, of separating clearly the normal modes.

The matrix notation has been used skillfully to carry through the general development. The necessity, however, of the use of column and row matrices suggests the possibility of the introduction of tensors. The mathematical idea in matrix and tensor notation is very similar. Where, however, column matrices or first order tensors occur together with functions of them, the tensors have often proved more compact and have given a better indication of the nature of the argument. For example, the discussion of the inertia characteristics of the system in this paper is considerably simplified by a tensor notation. If, as is suggested, a basic theory is to be introduced, it would seem wise to be sure that the most efficient mathematical notation is in use.

There are several questions raised by this paper and admittedly left unanswered. The proper choice of generalized coordinates in any extensive use of this theory is a problem of the first rank. Will the theory be able to serve as a guide in this or will independent methods of investigation be required? The author passes over the question of dynamic coupling also, which may prove important in the cumulative effect of small factors. It will probably be possible to adjust the theory to correlate with further experimental and theoretical results on the nature of the air forces. However, how far is the usefulness of the theory dependent upon the knowledge of these forces?

The mathematical elegance shown here is very valuable, and a compact and clearly organized development of this type will serve as a useful guide in further investigations.

Replies to Points Raised

– S. J. Loring
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THE choice of generalized coordinates is a problem of first rank; it is, in fact, the step which makes possible this type of general analysis. The conditions for the choice of satisfactory generalized coordinates are stated in paragraph 4 of Section II of my paper; these conditions themselves are usually sufficient for making a choice of the coordinates. A knowledge of the general shape of the components of the first few normal modes is required; I have found that the components of the normal modes of conventional airplanes can be obtained by inspection with the help of the fact that the normal modes satisfy the conditions of restraint. The example in Section II illustrates this process of obtaining generalized coordinates. If cases arise in which this method does not give results, investigations can be made using any generalized coordinates which satisfy the conditions of restraint by the methods of Section VI. The adequacy of the generalized coordinates chosen can be checked by still-air vibration tests (Section VI). Also, it is pointed out in Section IX that the step of choosing generalized coordinates for a whole class of structures can be done once for all.

The effects of dynamic coupling are, of course, included in the equations of motion and will be reflected in any of the solutions. It is beyond the scope of the paper to go farther than developing the equations of motion and indicating methods of solution, so that the question of dynamic coupling as such is not discussed.

In regard to Mr. Hall's suggestion to replace the matrix notation by tensor notation, there are several points that must be considered. One advantage of the matrix notation over the tensor notation is that the former does lead directly to the determinantal equation for flutter; matrix notation also helps to keep clearly before the analyst the linear character of the equations of motion with which he is dealing. Since the numerical computation is by far the most time consuming part of an actual flutter speed computation, even if tensor notation did shorten the notations, as Mr. Hall suggests, it is doubtful that this would shorten the whole problem of flutter analysis enough to outweigh the advantages of matrix notation just mentioned.

The air forces which have been introduced in Sections VII and VIII are admittedly only approximate as applied to actual surfaces. However, the success that has already been achieved in using these air forces in predicting flutter speeds of wings for which the simpler theories are adequate is good justification of their use in most cases. As mentioned in the text, the theory can be used equally well with any more precise (linear) air forces which become available in the future.

Development of Vehicle Specifications

WHAT are the steps to be taken – and in what order – to reach a correct analysis of a transport operation, one that will provide the basic information for properly specifying equipment?

First of all, a careful time study of the proposed operation itself must be made. The nature of the commodity, its weight and bulk and the quantity to be moved – either per trip, per hour, per day, or otherwise. The frequency of the trips must also be definitely ascertained in order to determine the required cycle of the operation.

Without this preliminary time study of the work to be accomplished there is nothing tangible on which to base the study of the equipment that must be used in order to accomplish the work. This will definitely indicate the type and size of the body which, of course, must be definitely ascertained before any consideration can be given to the chassis required.

An accurate check as to the nature of the roads to be traveled, the hills to be climbed, the traffic conditions, and the number of stops and starts, with the estimated time lost in loading and unloading is of vital importance in connection with the cycle of the operation and its performance ability requirements.

The next step is a check-up and summarization of the restrictions imposed upon dimensions and weights of various types of equipment by the various state laws through which the operation must proceed. It frequently occurs that the logical equipment (from the standpoint of lowest cost per unit hauled, or per mile traveled) is not permitted to operate because of these restrictions. Authentic state law interpretations graphically illustrated are available from the National Highway Users Conference or direct from the various states themselves so that no error should be made.

Next should come an investigation of the income that will be derived from the operation or its cost as a for-hire job, so that the estimated cost after it has been developed, can be compared in order to determine whether it will be a profitable operation.

The foregoing may or may not be listed in order of most importance and being in a sense intangibles and very much interrelated may be subdivided and qualified again and again.

Having in mind the commodity to be transported, its nature and its quantity in terms of weight and bulk, the operation cycle requirements and each of the foregoing intangibles, a direct index is thus provided for the type of body required, its general dimensions and its weight.

The thus-indicated combined weight of payload and body equipment thereby definitely indicates the structural strength or carrying capacity required in the chassis by which it must be supported and transported, either truck, semi-trailer or other combination.

Excerpt from the paper: "Motor-Vehicle Load Distribution Factors," by Fred B. Lautzenhiser, chief transport engineer, International Harvester Co., presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 2, 1941.

MECHANICAL MINDS for MOTOR CARS

by HAROLD E. CHURCHILL
The Studebaker Corp.

IN the discussion of automatic devices in motor cars presented in this paper, special emphasis is placed on fully automatic and semi-automatic clutches, drives, and transmissions.

Two methods of attack are being used to eliminate the clutch pedal: power operation of the mechanical clutch and the fluid coupling. Three fundamental types of control for vacuum-operated clutches are discussed:

1. **Position or follow-up type valving.**
2. **Balanced or pressure-sensitive type.**
3. **Balanced cushion with variable bleed.**

In a review of the merits and demerits of the fluid coupling, fuel economy and the shock loads on the driving mechanism are stressed. An appraisal of the operating principles and characteristics of various fully and semi-automatic transmissions compares the following: overdrive; four-speed semi-automatic transmission with fluid coupling; three-speed fully automatic transmission; and four-speed fully automatic transmission with fluid coupling.

In his conclusion the author predicts that sliding gears will not be used in the final version of a generally satisfactory automatic gear box; that the clutch pedal and driver operation of the clutch will soon be eliminated in all but the lowest-price cars; and that the final design of automatic transmission must be torque- and speed-responsive and should have at least four gear ratios or their equivalent.

THE use of automatic devices has been one of the mediums through which we have attained today's standards of safety, economy, convenience, and comfort, as found in the modern automobile. Such devices assume the responsibility for the completion of a routine duty which was formerly an obligation of the operator, and relieve him of the manual and mental effort. Thus, they simplify driving and, by the repetition of a properly predetermined cycle of events to meet a given demand most effectively, they permit more economical operation of the vehicle.

Such devices, of necessity, must complicate rather than simplify the mechanism of the car. Manufacturing problems are increased. Trained specialists are required for the service techniques involved. However, these problems are more than offset by the convenience and pleasure afforded the owner.

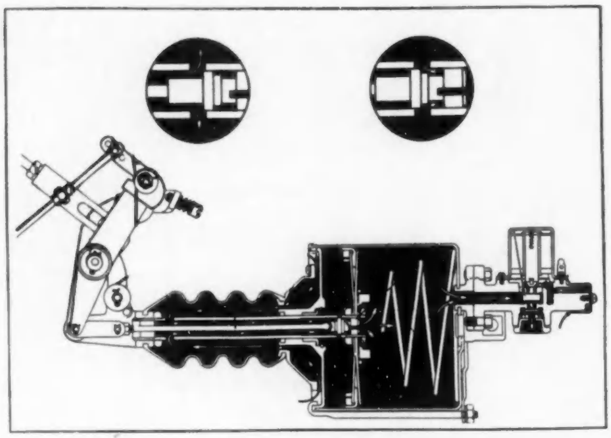
There are two general classifications, namely: full-automatic and semi-automatic, into which all such units fit. A physical or chemical reaction, while the automobile is in operation, is the guiding genius directing the fully automatic device in its operation and may be considered its brain. The semi-automatic device requires the direction and coordination of the operator. All units under the foregoing classifications employ mechanical, electrical, pneumatic, or hydraulic processes, alone or in combination, to execute a command as directed. Every automobile produced in 1941 includes one or more such devices.

Automatic devices in the preceding classifications are today accepted as component parts of the design of engines, electrical systems, and suspensions of automobiles.

Thermostats and water distribution control and maintain uniform cylinder-block temperatures. Thermostatic manifold heat valves, automatic chokes, vacuum economizers and accelerating pumps are component parts of the induction system to aid in starting, to improve economy at road load, and to increase power at full throttle. Vacuum spark advance and a centrifugally operated distributor advance permit the engine to be operated with a high spark advance at part throttle but retard the spark at low manifold depressions. Automatic drains in the intake manifold permit raw fuel to drain from the manifold without admitting air. Tappets are made to turn to reduce wear, and self-adjusting to reduce noise. Bearings are protected by constant oil pressure maintained by the relief valve. Even air conditioning is provided to maintain the health of the engine.

The electrical system is protected by current and voltage regulators, thermostatic cutouts, and voltage compensators. Automatically operated lights are provided in the engine compartment and body interior.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 5, 1941.]



■ Fig. 1 – Position or follow-up type valving for vacuum-operated clutch

Automatic valves, of the thermostatic and inertia type, make adjustments in shock absorbers, to compensate for temperature and road conditions, without effort on the part of the operator.

These and many more are commonplace items in the make-up of the modern automobile. They contribute to the continued improvement in performance and comfort made by the industry in its product. However, it is not within the scope of this paper to discuss all of them in detail.

The medium of transmitting and the method of controlling power from the engine to the driving wheels have occupied the attention of engineers since the inception of the first automobile. This problem now occupies the center of the stage. It is therefore the author's aim in this paper to discuss some of the recent developments in this field.

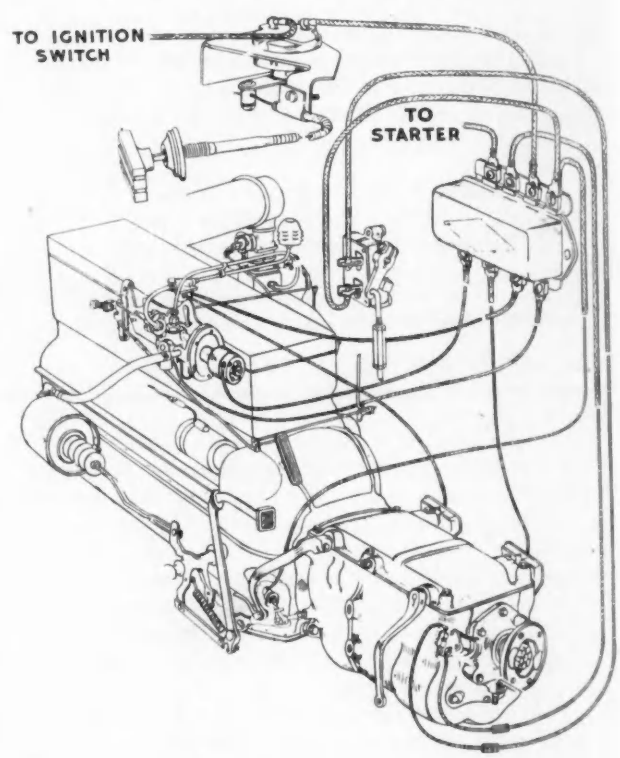
The introduction of a fully automatic transmission has not only focused the attention of the industry on transmission development but, because of the elimination of the clutch pedal, it has presented it with a problem of operation of the primary clutch mechanism. Two methods of attack, namely, the fluid coupling and power operation of the mechanical clutch, have been used to date.

Semi-centrifugal clutches have been used as a means to reduce clutch pedal pressure. By utilizing centrifugal force, acting on counterweighted release fingers, to increase plate pressure with increase in engine speed, very light pedal pressures may be obtained at engine idle by using light pressure-plate springs. Of course, this advantage is nullified at high engine speeds because the combination of spring pressure and centrifugal force must be overcome to effect a release. Centrifugal types of clutches have the inherent disadvantage that they are dependent only on engine speed for engagement and disengagement. Attempts to overcome this disadvantage by the addition of vacuum or electrical control have not eliminated this objection. Although primary clutches, actuated by hydraulic pressure, have been used with considerable success experimentally, they present many problems difficult of solution for a satisfactory production design. Any satisfactory design would seem to be too costly and complex to justify its use for the sole purpose of operating a primary clutch. Pneumatic controls, using manifold vacuum as a source of power for actuating the clutch, have been used with continued success and are found on several 1941 model cars as optional equipment.

Vacuum-operated clutches available today use three fundamental types of control as follows:

(a) Position or follow-up type valving (Fig. 1).
(b) Balanced or pressure-sensitive type (Fig. 2).
(c) Balanced cushion with variable bleed (Fig. 3).

The follow-up type of control employs a power cylinder with the control valves incorporated in the piston rod, and connected through a toggle mechanism to the accelerator treadle. The piston rod is connected to the clutch release shaft through the same toggle lever system in such a manner that motion of the piston rod and valve rod are superimposed. Initial movement of the accelerator moves the valve rod rearward, closing the cylinder to atmosphere and opening the ports so that a pressure balance is obtained on either side of the piston. Since the piston rod moves faster than the valve rod, the piston rod ports overtake the valve plunger and seal off the vacuum ports. The resulting "follow-up" action takes place throughout the engaging stroke. By means of an adjustable cam in the control linkage, the clutch is engaged rapidly to the "cushion point." If movement on the accelerator pedal is not maintained until full engagement is secured, the piston-rod valve will come to its "lap" position and check engagement at this point.



■ Fig. 2 – Balanced or pressure-sensitive control of vacuum-operated clutch

In this type of control, clutch engagement is regulated only by the movement of the accelerator. Since the rate of clutch engagement, once the cushion point has been reached, varies with the gear ratio in which engagement is made, this type of control would appear to require con-

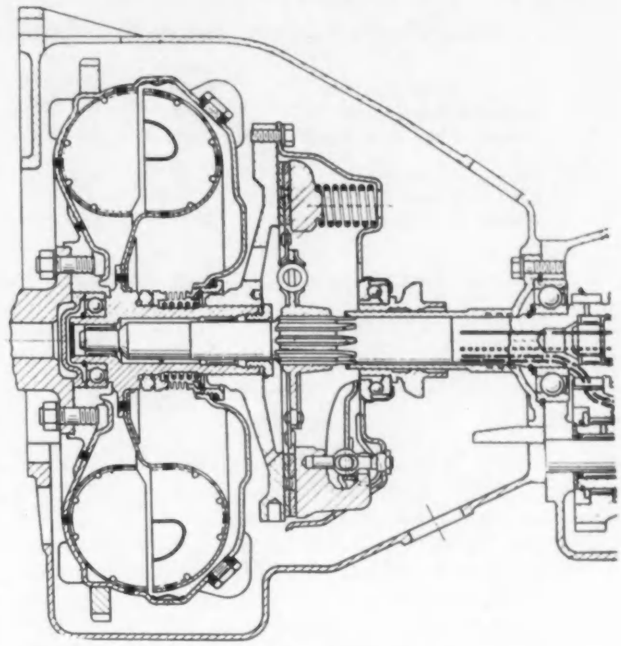
siderable finesse on the part of the driver to secure smooth engagements in all gear ratios. Of course, this design is not intended to eliminate the clutch pedal but only reduce the number of manual clutch operations required.

The balanced or pressure-sensitive type valving is being used in a clutch control system which permits the driver to shift gears in the normal manner without using the clutch pedal. However, the clutch pedal is retained as a part of the system to assist smooth engagement during "warm-up" and for cold starting.

In this design the rate of clutch engagement is controlled entirely by the rate of air bleed into the vacuum cylinder and is independent of manifold vacuum. Therefore, clutch engagement is not affected by variations in manifold depression.

In this design clutch engagement is made by an air bleed into the cylinder. A primary regulator valve interconnected with the accelerator pedal and throttle opening provides a rate of air bleed selected to satisfy normal engagements. The control of air bleed through the primary valve is augmented by solenoid-operated valves for abnormal conditions such as encountered in starting in first, second or reverse gear, and also in shifting from third to second at relatively high car speeds. The entire system referred to is rendered inoperative above approximately 17 mph to prevent free-wheeling above this speed. Provision is made to overrule the governor, so that shifts may be made out of direct gear above the governed speed. A lockout switch is also provided to secure down-hill braking in second gear.

The automatic clutch system in which this type of control is being used requires a complex arrangement of automatically operated controls for the sole purpose of eliminating foot operation of the clutch. The installation, when properly serviced, operates satisfactorily. It has been the author's experience that extreme caution must be exercised in design, production, and service, to insure owner satisfaction with any automatic device requiring such a multiplicity of closely coordinated controls.

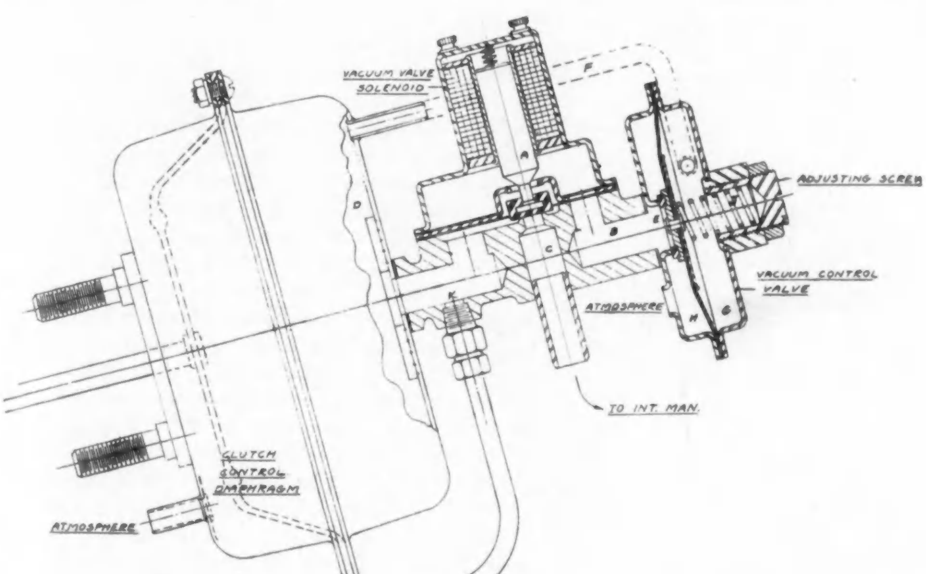


■ Fig. 4 – Fluid coupling

The third basic type, balanced adjustable control with variable bleed, is the simplest form of vacuum clutch control available and appears to meet the necessary requirements. This system employs a power cylinder and control valve as an integral unit. It requires the use of a governor switch, throttle switch, and "overruling" switch to complete the installation.

The cylinder incorporates a diaphragm-type piston with the solenoid valve and regulator valve directly attached thereto. The solenoid valve is energized and opened to manifold vacuum with the accelerator pedal in the idle position, below any established engine speed at which the governor maintains the electrical circuit to energize the solenoid valve. Manifold vacuum admitted into the power cylinder is also admitted to the regulator valve in opposition to a predetermined spring load. The spring load is always less than the total force exerted by full manifold vacuum but, since the regulator diaphragm is vacuum suspended when the clutch is in the released position, the regulator valve is closed.

As in other types of installation, the first movement of the accelerator pedal breaks the energizing circuit on the solenoid valve. When the



■ Fig. 3 – Balanced cushion with variable bleed control of vacuum-operated clutch

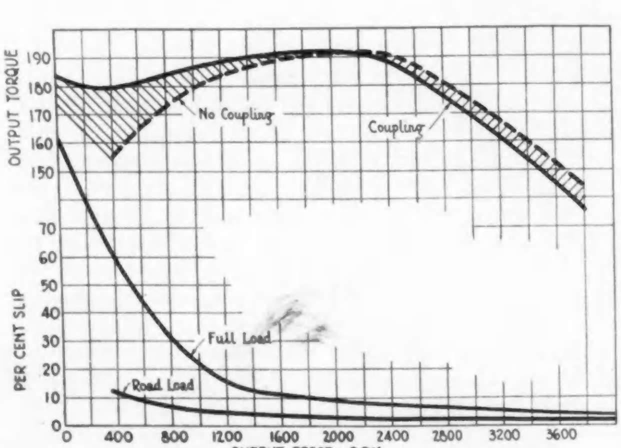
Table 1 – Results of Economy Tests on Cars Equipped with Various Combinations of Transmission and Coupling

Car	Transmission	Axle Ratio	Coupling	Gasoline Used, % Traffic	Gasoline Used, % Country	Miles Per Gallon Traffic	Miles Per Gallon Country	Miles Per Stop Traffic	Miles Per Stop Country
A	Automatic Two-Speed	3.54	Yes	120	104	8.8	16.5	....	....
A	Overdrive 2nd Gear Start	4.55	No	100	100	10.6	17.3	....	....
B	Automatic Four-Speed	3.42	Yes	100	100	8.7	15.0	0.10	8.5
C	High and Low Ranges, Automatic Start in Low Range	3.54	Yes	102	101	8.6	14.9	0.11	8.5
D	Conventional	4.3	Yes	132	125	8.1	13.0	0.15	8.9
A	Overdrive 2nd Gear Start	4.55	No	100	100	10.6	15.6	0.15	8.1
E	Overdrive 3rd Gear Start	4.55	Yes	119	108	10.4	19.1	0.11	8.8
E	Overdrive 2nd Gear Start	4.55	No	100	100	12.3	20.9	0.11	9.1

solenoid valve shuts off manifold vacuum, it also automatically admits atmosphere to the power cylinder and momentarily unbalances the regulator diaphragm, thus permitting the regulator valve to open but, as soon as the vacuum drops sufficiently to permit the spring load to overcome the force exerted by the remaining vacuum on the regulator diaphragm, the regulator diaphragm closes. Such automatic regulation permits quick engagement of the clutch to the cushion point and further final engagement is secured by bleeding atmosphere into the power diaphragm through a bleed valve incorporated in the throttle switch. The bleed-valve orifice increases in area with throttle opening, thus providing fast engagement for full-throttle acceleration and vice versa. Since this system incorporates a governor switch to prevent free-wheeling above predetermined speeds, means is provided in an overruling switch in the shift mechanism for operating the clutch to shift gears above the governed speeds.

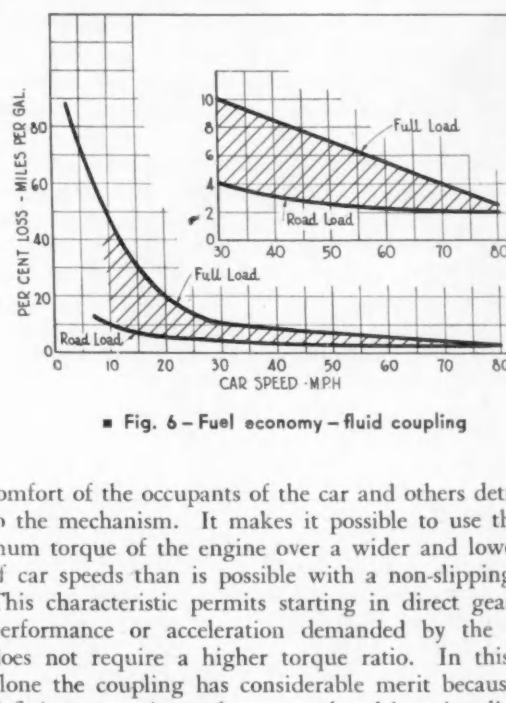
The cushion point is the only adjustment necessary in this system and is made by varying the spring load on the regulator diaphragm. Once set, this adjustment needs no further attention. Since this system satisfies all conditions of clutch operation with certain types of transmission, the clutch pedal may be dispensed with.

Recently the fluid coupling (Fig. 4) has come into prominence as a means of attack on the problem of clutch-pedal consciousness. By inference in advertising, the belief has grown in the public mind that fluid drive eliminates the necessity for manual clutch operation. All American cars using sliding-gear transmissions are equipped with a mechanical clutch.

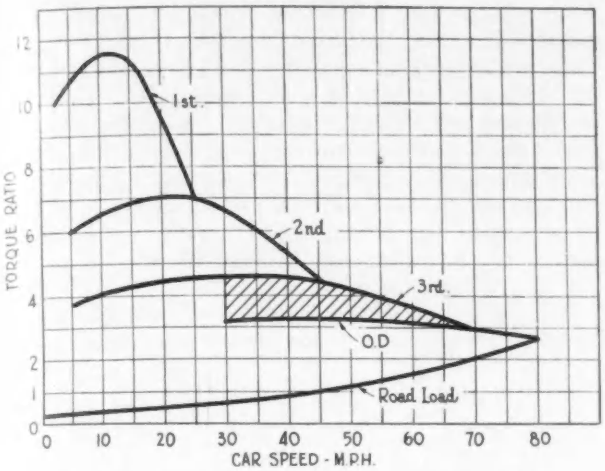


■ Fig. 5 – Slip and output torque – fluid coupling

There has been considerable discussion of the merits of the coupling. Shock loads on the driving mechanism are reduced. It is a means of damping torsional disturbances in the drive line, some of which are disturbing to the comfort of the occupants of the car and others detrimental to the mechanism. It makes it possible to use the maximum torque of the engine over a wider and lower range of car speeds than is possible with a non-slipping clutch. This characteristic permits starting in direct gear, if the performance or acceleration demanded by the operator does not require a higher torque ratio. In this respect alone the coupling has considerable merit because it is a definite convenience factor to the driver in eliminating some gear shifting. However, there is a loss of torque available at speeds above maximum engine torque because of slippage. Because of the inherent characteristic of the coupling which permits it to transmit torque with 100% slip, even at engine idle, the mechanical clutch must be disengaged or the brakes applied, to prevent creep when the car is stopped in gear. Slippage occurs at all car speeds and varies with the power transmitted. Fig. 5 shows the torque increase at low speed ratios and the percent slip variation between full and road loads over the entire speed range of a typical coupling installation on a 190 ft-lb engine. Horsepower loss is directly proportional to the slippage and is directly reflected in higher fuel consumption per mile per gallon. Fig. 6 shows the effect of the coupling on increased fuel consumption in miles per gallon



■ Fig. 6 – Fuel economy – fluid coupling



■ Fig. 8 – Performance characteristics of the overdrive

at road and full load for the same coupling and engine used in Fig. 5. These economy data are calculated from horsepower and pounds per brake horsepower hour fuel consumption data taken at full and road loads on the same engine. The loss in miles per gallon at road load is not quite proportional to the percent slip because, for a given output speed at road load with the coupling, the throttle opening must be slightly greater to compensate for the horsepower loss in the coupling. This difference, however, is very slight in pounds per brake horsepower hour.

Table 1 gives the results of economy data obtained in comparable road tests of cars equipped with various combinations of transmission and coupling-equipped cars. Comparative road tests have been made on various combinations of transmissions and couplings to obtain actual economy data as shown in Table 1. The cars were operated in pairs over the same test route under identical traffic and cruising conditions. All starts were made in the gear which the average owner would use for such normal operating conditions. In all tests, except those data pertaining to cars B and C, a car without coupling was considered as a base for comparison of 100%. Cars D and A, although of different makes, were approximately the same size, weight, and pounds per horsepower, so the results are fairly comparable. It will be noted from the data that the cars made approximately the same number of stops. All coupling-equipped cars showed a marked increase in gasoline consumption. All cars used from 50 to 100% more gasoline in city traffic than in country driving.

Following are some of the difficulties the author has observed on tests of coupling installations:

(a) Possible damage to the rear main bearing because of high coupling temperature.

(b) Oil fumes entering the interior of the car, should the coupling "blow off."

(c) Increased temperatures at the toe and floor boards requiring better insulation.

(d) Seal leakage due to high temperature drawing of the seal spring producing lowered unit pressure on the seal ring.

(e) Ventilation of the clutch housing and baffling for adequate dirt exclusion.

The following objections may be noted by the car owner:

(a) Increase in fuel consumption.

(b) Higher engine speeds for a given axle ratio.

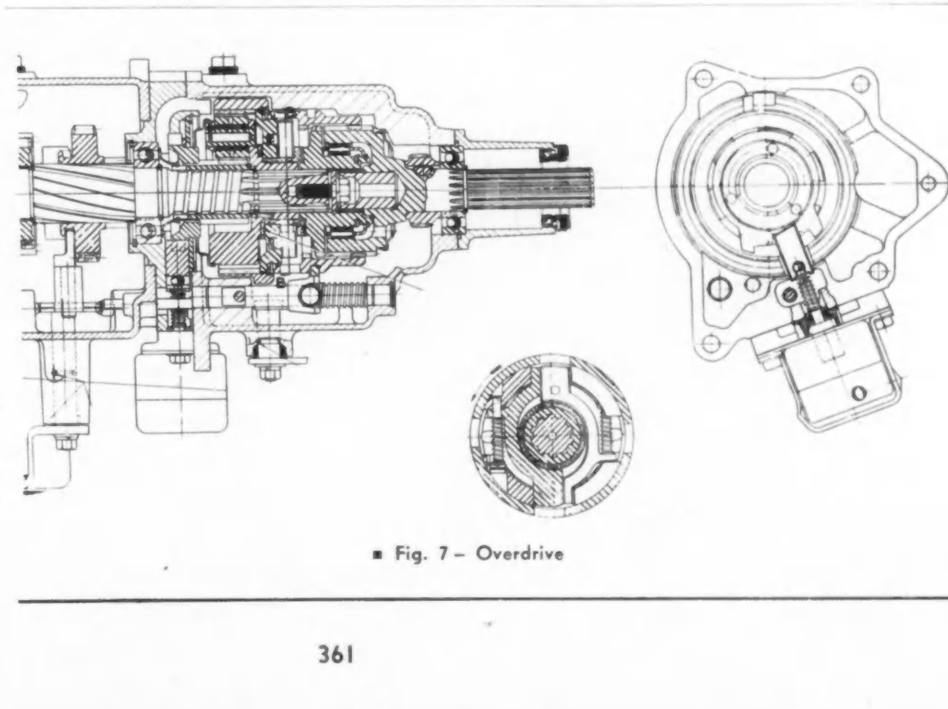
(c) Higher towing speeds required to start car, especially with cold oil.

(d) Car cannot be moved with starter in case of an emergency.

(f) Increased time required for automatic shift in slide-gear transmissions because of necessity for delaying throttle return to idle, to prevent engine stalling when the car is stopped in gear.

Most of the objections just noted would be eliminated by using the mechanical clutch when the car is stopped in gear. However, the convenience of being able to drive without using the clutch pedal should not be denied the owner of a fluid-coupling-equipped car just because of engineering problems inherent in the design.

In recent years more consideration has been given to transmission development than to any other item on the car. The public has been made transmission-conscious because these developments have produced tangible improvements in convenience, comfort, and economy of operation. The shift lever through the floor has been antiquated by remote-control shifting mechanisms, usually located on the steering column. This new location makes the shift more accessible and tends to reduce noise telegraphed into the driver's compartment. It also provides more usable and more comfortable passenger space in the front seat. One manufacturer has reduced the effort and



■ Fig. 7 – Overdrive

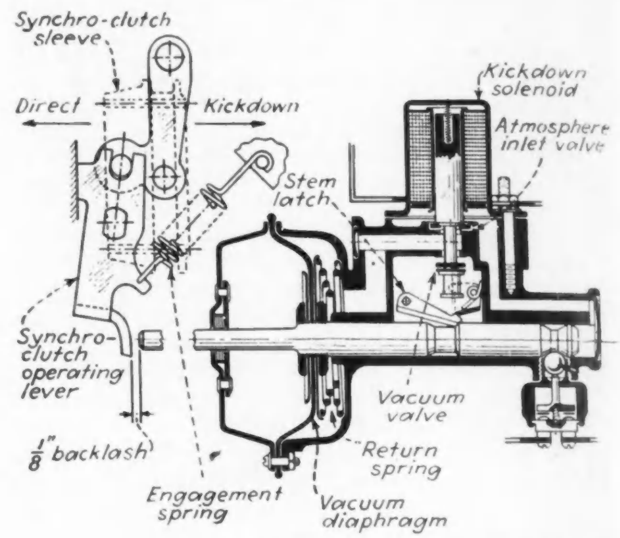
travel required to shift gears by supplying a power shift that is controlled by the shifter lever as standard equipment. Several others offer the same system as optional equipment. Other power devices to augment the standard shift are available but none are now in use. Notable among these is the "Electric Hand" system of remote-controlled selective shifting.

Various designs of automatic transmissions which supply certain pieces in the jig-saw puzzle of satisfactory performance are available today. All the necessary characteristics are available in one design or another, but most of them are lacking in some detail. In other words, the tools are ready and waiting but we don't know how best to use them.

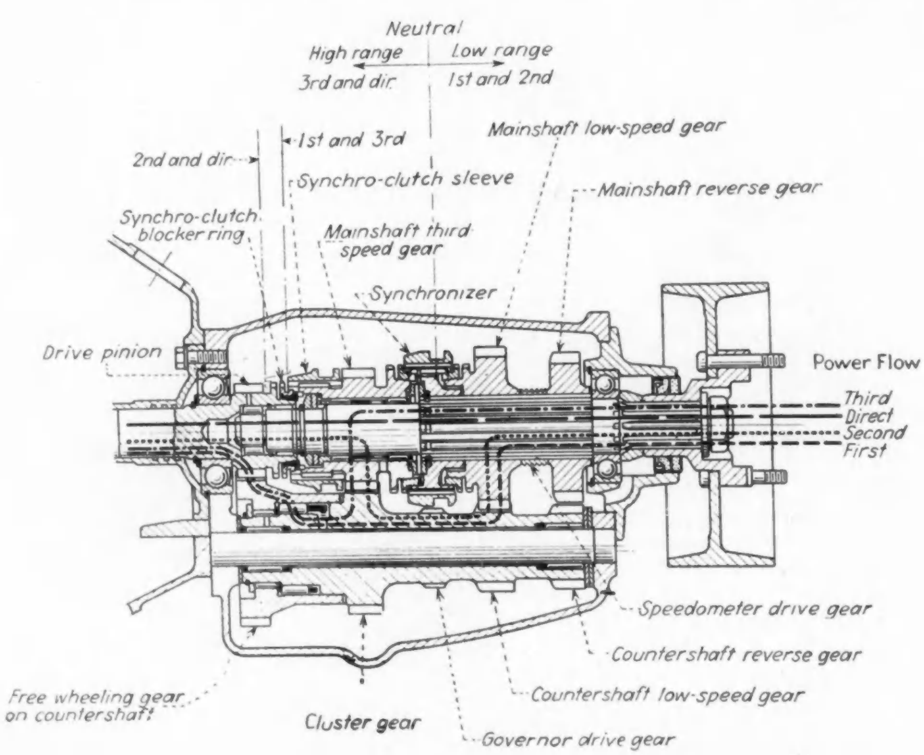
The results of the tremendous amount of development work to date have produced three basic types of automatic and semi-automatic controls available in production. Their principles of operation are quite well known, but an analysis and comparison of their performance characteristics produces some interesting conclusions. A three-speed automatic gear box is included in this discussion for comparison with the three types now in production.

The overdrive (known as Transmission A, shown in Fig. 7) is the modern forerunner of the automatic transmission. Its performance characteristics are indicated in Fig. 8. Manual selection of gears provides an 11.5:1 ratio in first and 7:1 in second gears for starting and grade ability. A 4.5:1 ratio, manually selected for traffic operation, may be changed automatically to 3.3:1 above 30 mph for cruising and economy. If the demand for a higher ratio than that provided in overdrive is desired, it may be obtained by a manual action of the operator. The area in which this semi-automatic selection of torque ratio occurs between the limits of 3.3:1 and 4.5:1 is indicated by the shaded area in Fig. 8. Therefore, it will be observed that only a small percentage of the total maximum performance available is obtained in this type of transmission without manually shifting gears. However, the largest percentage of a car's mileage life is obtained within the speed range covered by the overdrive. Automatic shifting in the overdrive range does provide a good combination of acceleration and economy at the will of the operator, without the necessity for manually shifting gears to obtain it. Propeller shaft speeds are considerably higher than in the other types under discussion because of the 4.5:1 direct ratio.

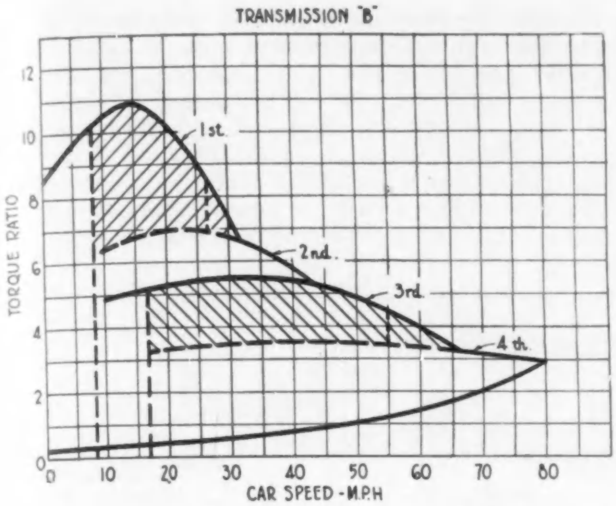
Transmission B (Fig. 9) employs approximate overall ratios of 11:1 in first, 7:1 in second, 5.5:1 in third, and



■ Fig. 10 – Vacuum-operated mechanism employed in Transmission B for gear shifting



■ Fig. 9 – Semi-automatic Transmission B



■ Fig. 12 – Composite of high and low-range characteristics of Transmission B shown in Fig. 11

3.5:1 in fourth gears. These ratios are grouped into two manually selective ranges. The low range comprises first and second gear ratios for maximum acceleration, starting, and climbing. High range includes third and fourth gears for traffic and cruising.

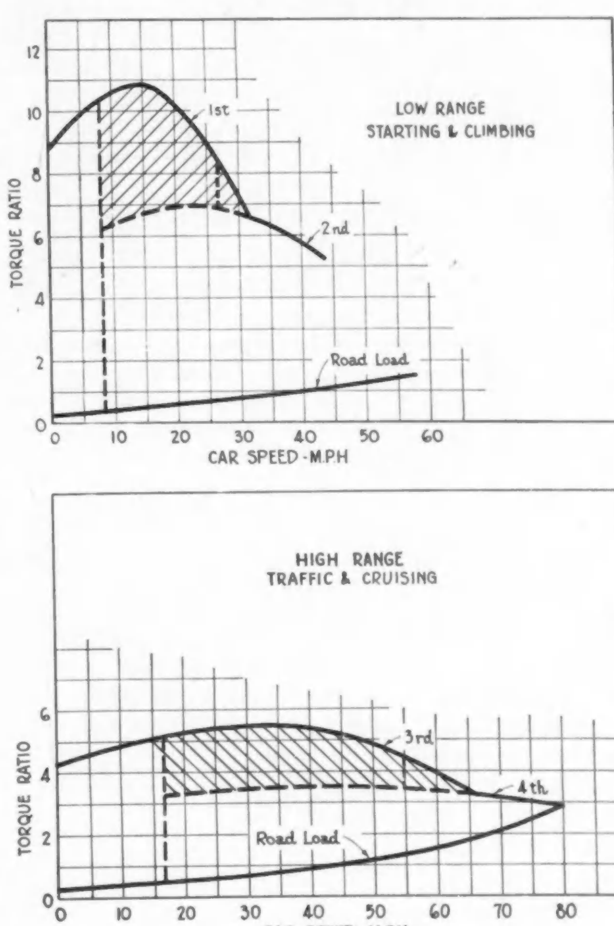
The speed limits within which automatic shifting may be accomplished are governor-controlled. Within the speed range defined by governor control, "up" or "down" shifting is effected at the driver's option by releasing or depressing the accelerator. Actual automatic shifting is done by a vacuum-operated mechanism (Fig. 10).

In the manually selected low range, the car is always started in the 11:1 ratio. Through the governor control mechanism, an automatic shift to second gear may be made by releasing the accelerator at any speed above approximately 8 mph. Should torque demand necessitate a shift back to first gear, this can be accomplished by depressing the throttle to the fully open position. However, provision is made to prevent kickdown at a car speed where such a maneuver would give an excessive engine speed but no appreciable increase in torque available to drive. The gear box returns automatically to first gear below the governed speed, regardless of throttle position. The sectional portion of the low-range chart in Fig. 11 represents the area in which automatic shifting may be accomplished.

In the manually selected high range, starting is always done in third gear (5.5:1 ratio) and by governor control an automatic shift to direct (3.5:1) can be made by releasing the accelerator at any car speed above approximately 17 mph. Should torque demand, such as in heavy traffic, occur above the governed speeds, the transmission can be shifted back to third gear at the will of the operator by depressing the accelerator. Provision is also made in this range to prevent kickdown at a high car speed where no appreciable increase in torque would occur, but where excessive engine speeds would result because of the 5.5:1 third gear ratio. The sectioned portion of the high-range chart in Fig. 11 represents the area in which automatic shifting may be accomplished. The engine speed differential between third and fourth gears is, in the author's opinion, objectionable at higher car speeds. However, the 5.5:1 ratio third gear, in combination with the fluid coupling and the range of automatic operation between third and fourth gears, provides satisfactory performance for most normal operating conditions, except for actual starting torque or low-speed acceleration such as in heavy traffic. To secure this range of operation requires only one manual shift.

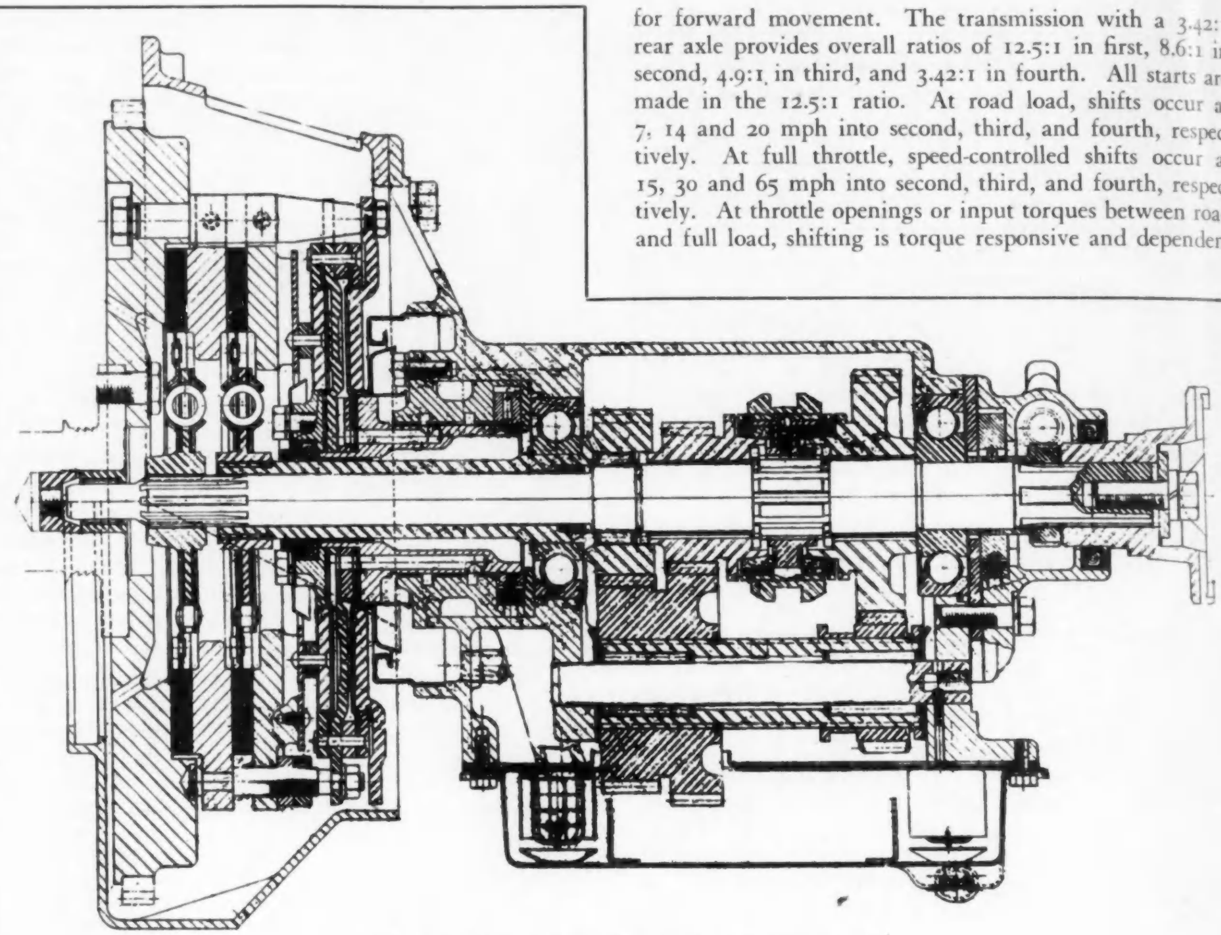
Fig. 12 is a composite of the high- and low-range characteristics shown in Fig. 11. Comparison of the sectioned areas of automatic operation with that in Fig. 8, shows that the operation of Transmission B automatically covers considerably more of the total performance available than does the automatic overdrive. The 3.5:1 direct ratio is also desirable because of the reduction of propeller shaft speeds and accompanying roughness due to unbalance.

The performance characteristics of a three-speed fully automatic Transmission C (Fig. 13) are indicated in Fig. 14. One manual shift is required for all forward speeds. The gear box, in conjunction with a 4.8:1 rear axle, provides overall ratios of 11:1 in first, 4.8:1 in second, and 3.5:1 in overdrive. All starts must be made in the 11:1 ratio. Shifting is accomplished by hydraulic pressure and the gear box is speed and torque responsive. At road-load throttle, shifts occur at 10 mph from first to second, and at 25 mph from second, or direct to overdrive. The transmission is speed-controlled to shift from first to second at not greater than 25 mph. This is necessary because of excessive engine speeds occurring above 25



■ Fig. 11 – Low-range and high-range performance charts – Transmission B

for forward movement. The transmission with a 3.42:1 rear axle provides overall ratios of 12.5:1 in first, 8.6:1 in second, 4.9:1 in third, and 3.42:1 in fourth. All starts are made in the 12.5:1 ratio. At road load, shifts occur at 7, 14 and 20 mph into second, third, and fourth, respectively. At full throttle, speed-controlled shifts occur at 15, 30 and 65 mph into second, third, and fourth, respectively. At throttle openings or input torques between road and full load, shifting is torque responsive and dependent



■ Fig. 13 – Three-speed fully automatic Transmission C

mph in the 11:1 ratio, first gear. At greater than road-load throttle, shifting is done at car speeds proportional to the torque demand.

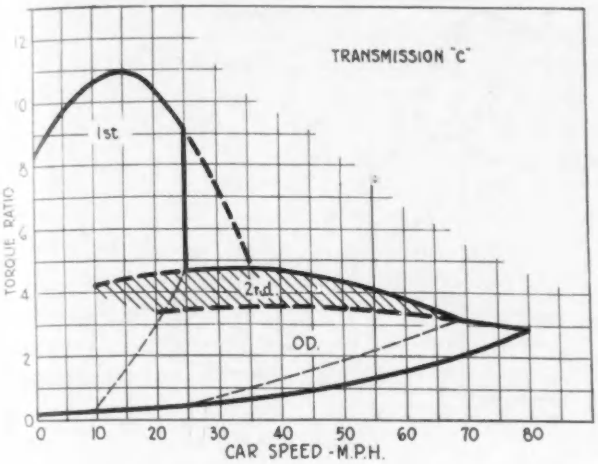
Once the shift is completed into overdrive the transmission will operate at full throttle in this ratio down to car speeds of 20 mph. However, provision is made to shift automatically from overdrive to second at the will of the driver. This is done by depressing the throttle to the wide-open position. The same operation is required on the other gear boxes herein discussed. Performance thus acquired is indicated in Fig. 15.

This transmission does not have acceptable performance because no higher than a 4.8:1 overall ratio is available above 25 mph. Its cruising performance is good because of the low overdrive ratio. The operation of this transmission gives conclusive evidence that any automatic transmission, to provide acceptable all-around performance, must have at least four gear ratios or their equivalent.

Transmission D (Fig. 16) utilizes constant-mesh planetary gearing controlled by hydraulic pressure actuating band and multiple-disc clutches to provide four forward speeds, automatically shifted, which are torque- and speed-responsive. The fluid coupling, an integral part of the transmission, transmits only a proportion of the torque input dependent on the gear ratio, and thus slippage is minimized. No clutch pedal or mechanical clutch is necessary.

The operator is required to make only one manual shift upon throttle opening. An attempt has been made to show graphically the performance areas covered in each ratio. These are represented by the shaded areas in Fig. 17. As in A, B, and C, once the transmission has progressively shifted to fourth gear, it may be shifted back to third gear at any speed (see Fig. 18) between 10 and 60 mph. This permits the driver to meet at will a torque

■ Fig. 14 – Performance characteristics of three-speed fully automatic Transmission C



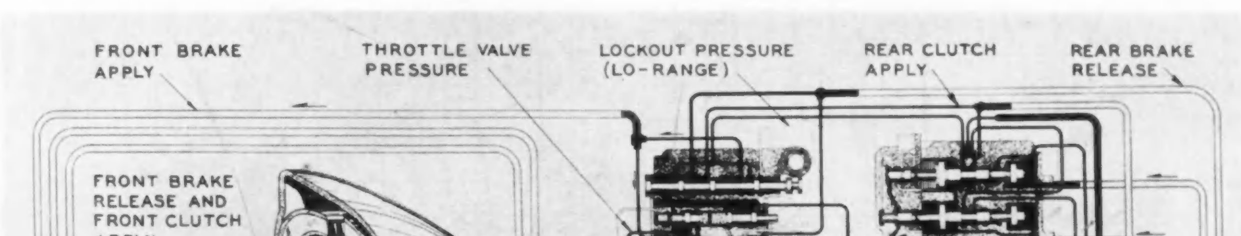
■ Fig. 15 – Performance acquired by shifting from overdrive to second on Transmission C

demand not fulfilled by fourth gear by simply depressing the accelerator to the full throttle position. Provision is also made for locking out third and fourth gears below 42 mph to provide engine braking for descending hills.

The performance characteristics of this transmission seem to cover the performance range more adequately than any other similar automatic shifting device now available and with the least amount of manual and mental effort required by the driver. Ratio changes at more than road-load throttle are often accompanied by a noticeable lurch due to the control of overlap in shifting. The 3.42:1 direct ratio is desirable because of propeller-shaft speeds, and the spread between second, third, and fourth gear ratios seems to be about the optimum.

Experience has shown that, with automatically shifted ratios higher than 1.4 or 1.5:1 between third and fourth gears, objectionable engine noise and general disturbance is produced due to the pronounced engine speed differential when shifts are made at car speeds in excess of 45 to 50 mph. The relationship of engine speed to car speed in the various gears of the transmissions just discussed are shown in Fig. 19 for Transmission A; Fig. 20 for B; Fig. 21 for the three-speed automatic Transmission C; and Fig. 22 for Transmission D. The shaded areas in these charts also indicate the regions in which automatic shifting is accomplished. As previously mentioned, Fig. 20 shows the effect on engine speeds of too wide a spread in gear ratio between third and fourth gears. Fig. 21 also indicates that the same condition prevails between first and second in this transmission. Such a condition not only produces objectionable noise, due to the sudden change, but has a profound effect on gasoline and oil economy.

Automatic or semi-automatic control of gear ratios available in the transmissions discussed operates the majority

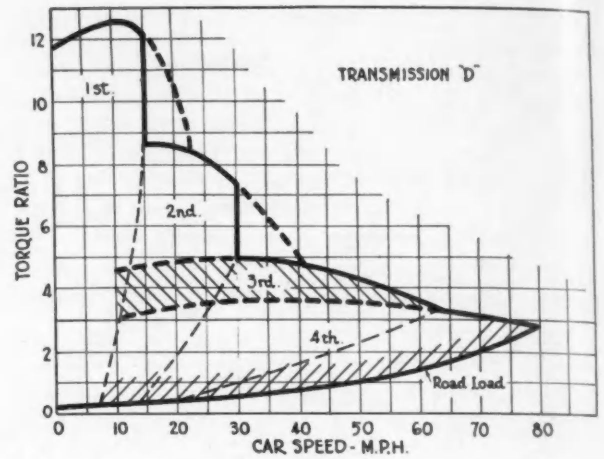


■ Fig. 16 – Four-speed fully automatic Transmission D

Oldsmobile

of the time without requiring the attention or conscious effort of the operator, but there are conditions of driving encountered in which their automatic operation is distracting and objectionable. One particular condition may occur in heavy traffic or in climbing curved, mountainous roads. Where car speeds are low and only slightly more than road load is required, the velocity and torque increment to effect a shift is small. When traffic or road conditions require constant change of throttle position at these low speeds, the continual up and down shift encountered is a distinct annoyance to the driver. For lack of better terminology, the author has called this condition "gear-change frequency."

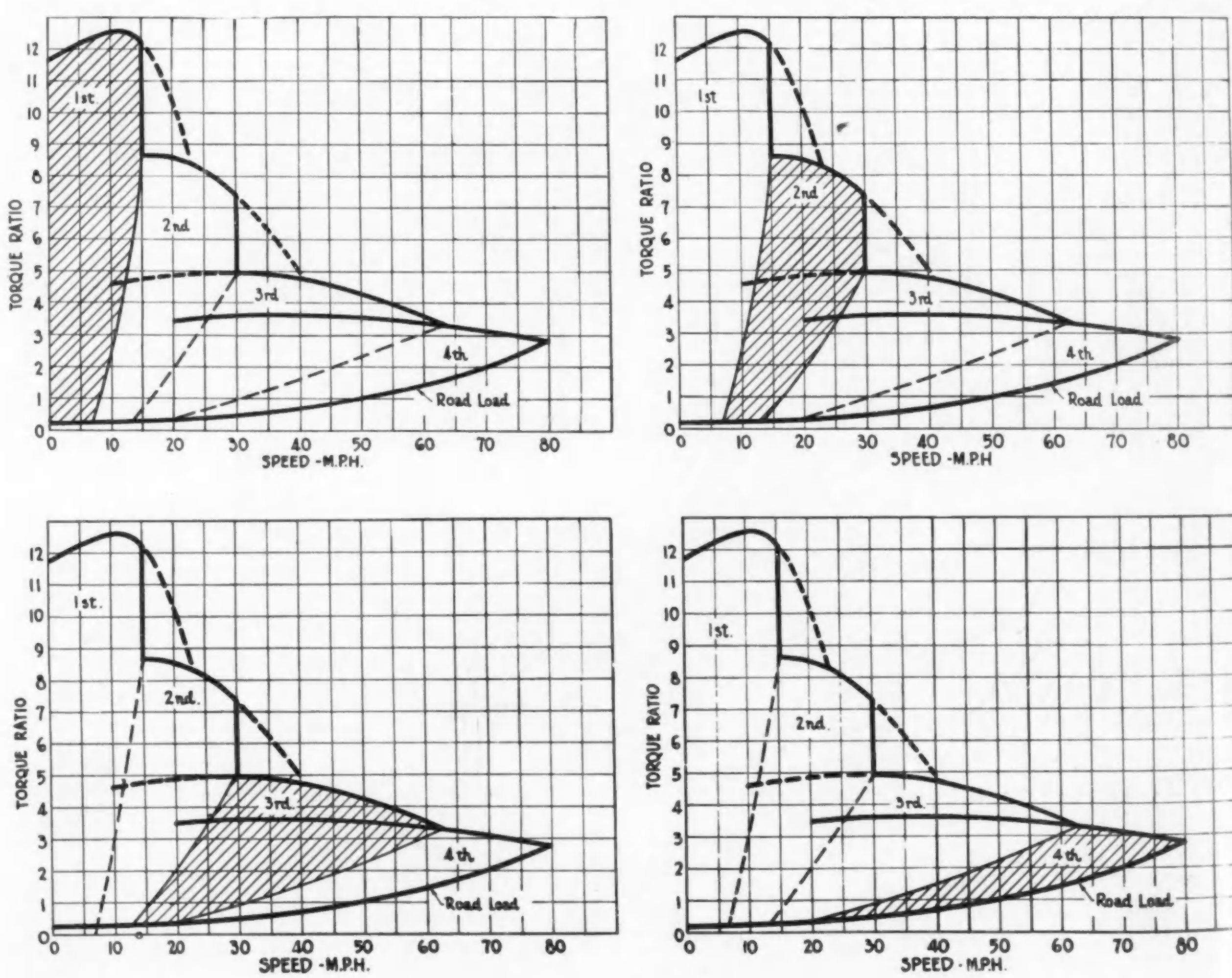
To obtain data on this type of operation, three cars equipped with automatic transmissions were run in direct comparison with an automatic-overdrive-equipped car having transmission performance characteristics indicated in Fig. 8. The results of these tests are shown in Fig. 23. Traffic and road conditions were comparable. Transmission D started in first gear made 17 shifts between third and fourth and one shift to second in negotiating the test route of approximately two miles. Transmission E, a three-speed box automatically shifted between second and third gears, made 16 shift cycles in traversing the course. Transmission C, a three-speed fully automatic, made 7 shifts down to first gear and three shifts up to overdrive,



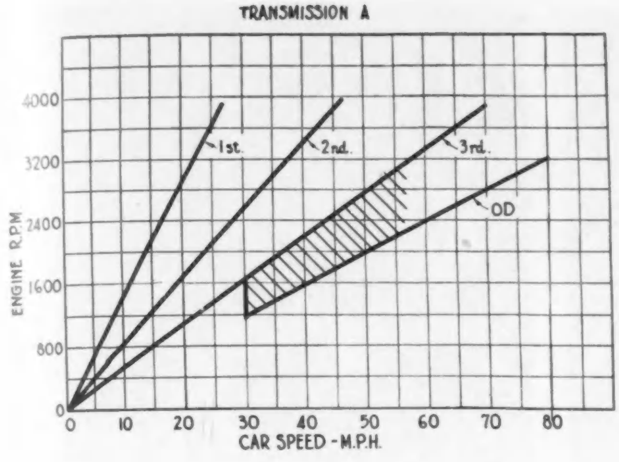
■ Fig. 18 – Performance acquired by shifting from fourth to third on Transmission D

while a standard automatic overdrive transmission car equipped with fluid coupling, negotiated the course in the same time as the others without requiring a single shift.

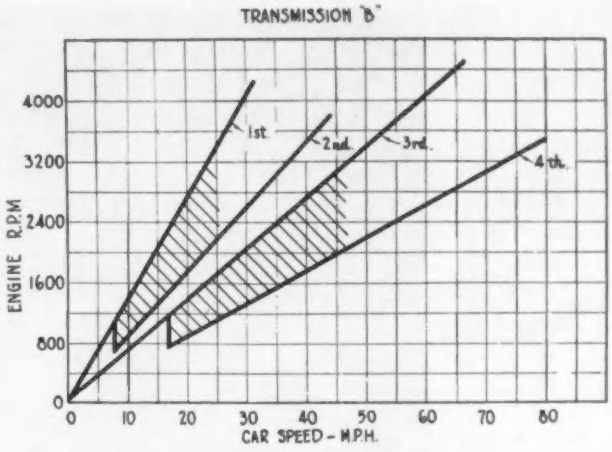
The ratios and car speeds given in this discussion are not exact, but sufficiently close to the actual figures that



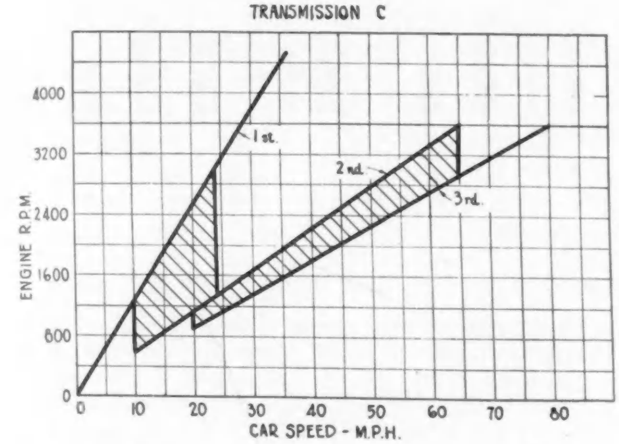
■ Fig. 17 – Performance areas covered in each ratio of Transmission D



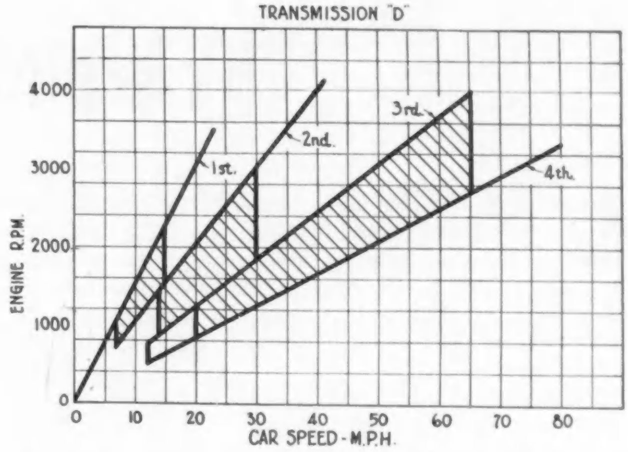
■ Fig. 19 – Relationship of engine speed to car speed in various gears of Transmission A



■ Fig. 20 – Relationship of engine speed to car speed in various gears of Transmission B



■ Fig. 21 – Relationship of engine speed to car speed in various gears of Transmission C



■ Fig. 22 – Relationship of engine speed to car speed in various gears of Transmission D

the slight error has no effect on the conclusions to be drawn from this discussion.

The author realizes that the scope of the problem of automatic operation of clutches and transmissions is too broad for specific analysis within the limited text of this paper but believes there are certain general conclusions to be drawn and predictions to be made from the generalities presented. They are as follows:

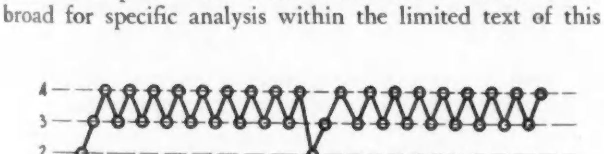
(a) The clutch pedal and manual operation of the clutch will soon be eliminated on automobiles in which price consideration permits the installation of automatic transmissions.

(b) Elimination of the clutch pedal will cause public demand and acceptance of power braking, operated by a foot treadle and requiring low pressure and short travel for definite control of the braking system.

(c) Automatic transmissions require the use of at least four gear ratios or their equivalent to provide satisfactory performance. It is the author's opinion that sliding gears will not be used in the final version of a generally satisfactory automatic gear box. The transmission must be torque- and speed-responsive.

(d) Only a choice of direction, selected from a neutral position, and manual operation of the accelerator should be required for the operation of an automatic transmission.

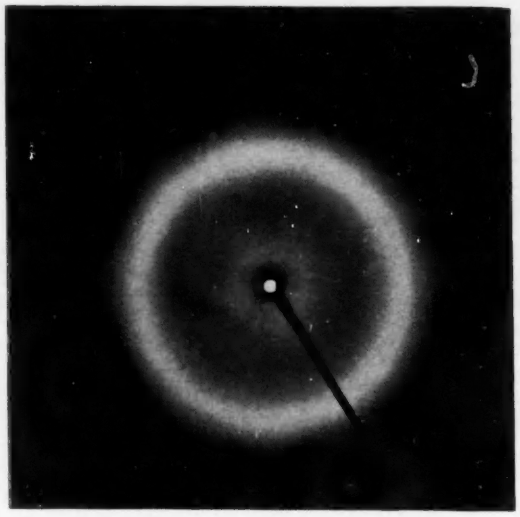
(e) Kinetic energy of a circulating fluid, in combination with planetary gearing automatically controlled by mechanical means, may provide the best means for producing a satisfactory method of transmitting power from the engine to the driving wheels of an automobile.

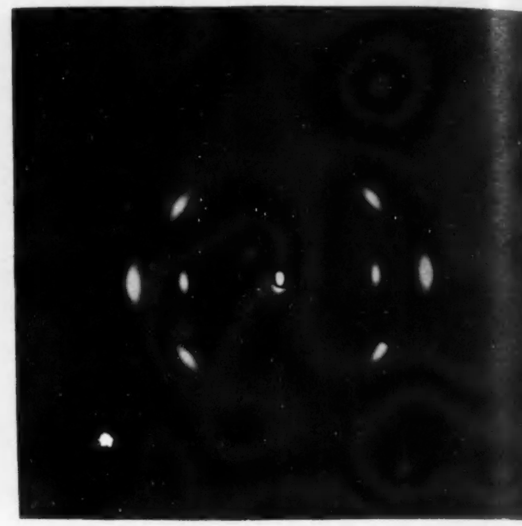


■ Fig. 23 – Gear-change frequencies – Transmissions A, C, D, and E

Fig. 1 (left) – Rubber unstretched

Fig. 2 (right) – Rubber stretched





Properties of Some

AS early as 1860, C. G. Williams[1], as a result of the destructive distillation of rubber, isolated isoprene and observed that, soon after its preparation, it was polymerized into a body having increased viscosity. Bouchardat[2], in 1879, found that isoprene could be converted into a rubber-like solid by treating isoprene with fuming hydrochloric acid. This may be said to be one of the first synthetic rubbers. However, Tilden[3, 4] was the first to observe the possibility of producing, from other types of raw materials, a synthetic material having the properties usually associated with natural rubber. He was the first to prepare isoprene from turpentine and to polymerize it to rubber-like materials. Later, in Germany, Hofmann[5] and Harries[6] carried out an intensive study of the polymerization of isoprene. It is also well known that during the First World War a synthetic rubber was prepared by

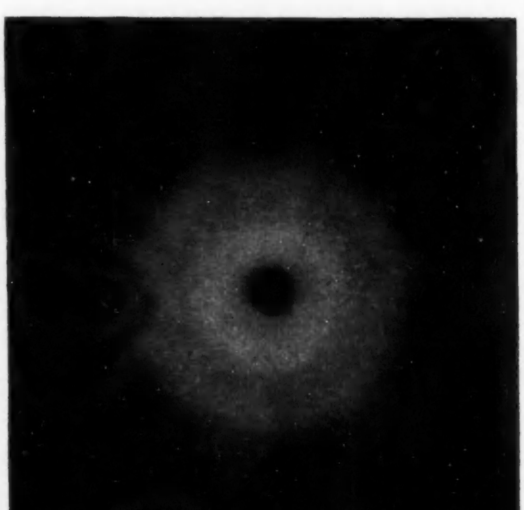


Fig. 5 – Neoprene unstretched

IN this paper a brief review of the development of synthetic rubbers is first given. The difficulty encountered in substituting the synthetic for natural rubber is discussed, and it is pointed out that, from the consideration of their molecular structure, one should not expect the two rubbers to be interchangeable in every way, but that special handling technique undoubtedly will have to be developed.

Results of certain vulcanizable synthetic rubbers in two typical rubber formulas are compared with one another and with natural rubber.

A new dynamic test is described, and the results of these same rubbers in the same formulas as used for the compounding tests are given.

Data on the dynamic modulus, internal friction, resilience and heat build-up are also presented.

It also is shown that, with one particular type of synthetic rubber, a relatively low loading of carbon black is necessary in order to give physical properties which approach those of natural rubber when measured by the same test.

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 9, 1941.]

[1] See Proceedings of the Royal Society (London), Vol. X, 1860, pp. 516-519: "On Isoprene and Caoutchine," by C. G. Williams.

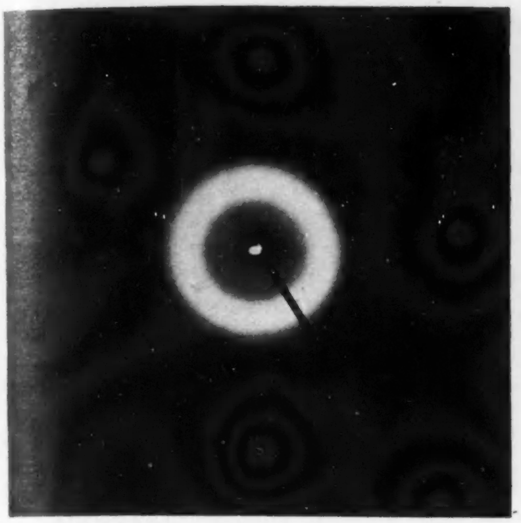
[2] See Comptes Rendus, Vol. 89, 1879, pp. 1117-1120: "Action des Hydracides sur l'Isoprene; Reproduction du Caoutchouc," by G. Bouchardat.

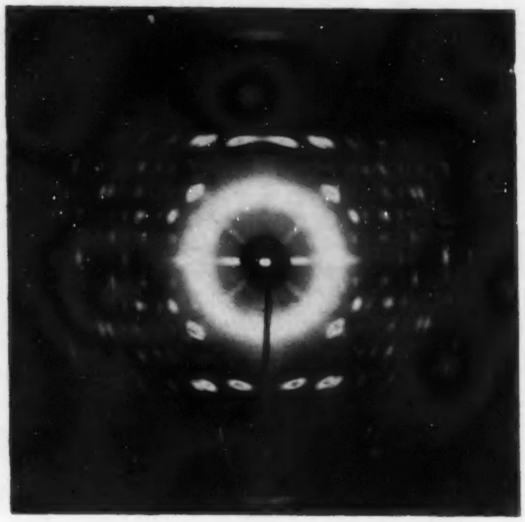
[3] See *Journal of the Chemical Society*, Vol. 45, 1884, pp. 410-420: "On the Composition of Terpenes by Heat," by W. A. Tilden.

[4] See *Chemical News*, Vol. 65, 1892, p. 265, by W. A. Tilden.

[5] F. Hofmann, German Patent No. 250,690 (1909).

[6] See *Chemiker Zeitung*, Vol. 34, 1910, p. 315: Lecture at Vienna, by C. Harries.





▪ Fig. 3 (left) — Vistanex unstretched

▪ Fig. 4 (right) — Vistanex stretched

SYNTHETIC RUBBERS

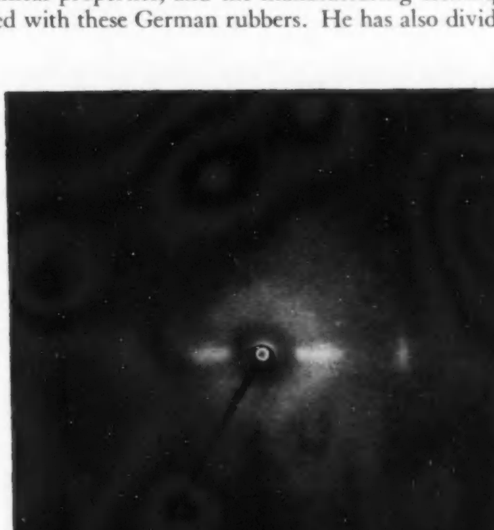
by L. B. SEBRELL and R. P. DINSMORE

The Goodyear Tire & Rubber Co.

polymerizing dimethyl butadiene at 30 C for a considerable period of time[7].

It has been estimated that the total production of artificial rubber during the period 1914-18 in Germany was 2300 tons. Subsequent to the World War, interest in synthetic rubber was not particularly acute until the discovery by Patrick[8] in 1920 of the plastic called Thiokol. This product, as is well known, is a condensation of an aliphatic dichloride with a polysulfide and is more of the nature of a sulfur plastic. In the early part of 1931, the Du Pont Co., based on the experiments of Nieuwland[9, 10], and further developed through the work of Carothers and his associates[11-14] introduced the material now commonly known as Neoprene, this material being based upon the polymerization of 2-chloro-butadiene. At about the same time, in Germany, the I. G. Farbenindustrie began extensive experiments to improve upon the type of rubber which had been used in that country in the preceding war and, as a result of this work, several processes were developed which were covered in patents to Tschunker, Bock and Konrad[15, 16]. These rubbers were based on the copolymerization of butadiene and acrylic nitrile or butadiene and styrene.

A very excellent discussion of all of the synthetic rubbers, together with a review of their compositions and properties, has been put out by Wood[17]. Stöcklin[18] also has set forth in considerable detail the structure, physical and chemical properties, and the manufacturing technique to be used with these German rubbers. He has also divided



▪ Fig. 6 — Neoprene stretched

[7] See p. 214 "Technologie der Kautschukwaren," by K. Gottlob, Vierweg u. Sohn, Braunschweig, 1925.

[8] J. C. Patrick, U.S. Patent 1,890,191 (1932).

[9] J. A. Nieuwland, U.S. Patent 1,811,959 (1931).

[10] See the *Journal of the American Chemical Society*, Vol. 53, 1931, pp. 4197-4202: "The Controlled Polymerization of Acetylene," by Nieuwland, Calcott, Downing, and Carter.

[11] See the *Journal of the American Chemical Society*, Vol. 54, 1932, pp. 4066-4070: "Addition of Hydrogen Chloride to Vinylacetylene," by Carothers, Berchet, and Collins.

[12] See the *Journal of the American Chemical Society*, Vol. 53, 1931, pp. 4203-4225: "Chloroprene and a New Synthetic Rubber," by Carothers, Williams, Collins, and Kirby.

[13] See the *Journal of the American Chemical Society*, Vol. 54, 1932, pp. 4071-4076: "Homologs of Chloroprene and Their Polymers," by Carothers and Coffman.

[14] See the *Journal of the American Chemical Society*, Vol. 55, 1933, pp. 789-795: "The Polymerization of Bromoprene," by Carothers, Kirby, and Collins.

[15] E. Tschunker and W. Bock, U.S. Patent 1,938,731 (1933); German Patent 570,980 (1933).

[16] E. Konrad and E. Tschunker, U.S. Patent 1,973,000 (1934); German Patent 658,172 (1938).

[17] See Bureau of Standards Circular C-427, June, 1940, by L. A. Wood.

[18] See Transactions of the Institution of the Rubber Industry, June, 1939, pp. 51-75: "Buna — A Review and Discussion of Present Development," by P. Stöcklin.

the highly polymerized rubber-like materials into two general classes:

1. Synthetics capable of vulcanization:
   a. Butadiene co-polymers:
   b. Reaction products of dihalogenated aliphatic compounds with sodium polysulfide.
2. Synthetics incapable of vulcanization:
   a. Polymerized isobutylenes (similar to Vistanex) of polymerized acrylic acid esters (similar to Plexigum and like materials).
   b. Products having rubber-like properties in combination with certain plasticizing agents (as, for example, Koroseal).

■ Scope of Paper

In the present paper, discussion will be limited to those types of synthetic rubbers falling in the first classification according to Stöcklin, just mentioned. It is proposed to select representative types of synthetic rubbers from this class, and to point out the difference in structure between some of these synthetic rubbers and natural rubber, as revealed by X-ray analysis. These same types of rubber have been compounded into two main types of formulas: (1) A formula containing approximately 23 volumes of hard carbon black and giving a rubber stock of the type usually used in tire treads; (2) A formula containing 50 volumes of soft carbon black and being, in general, representative of the formula which would be used for mechanical goods. No attempt was made to bring out the best properties which may be obtained with any of the individual rubbers but to show, in general, by applying the same formula to all of them, their general characteristics. Finally, in conclusion, a new dynamic test for evaluating various synthetic rubbers in comparison with natural rubber is described, and the results obtained by this test with the various synthetic rubbers will be given. The information derived from such dynamic tests represents the type of information which an engineer would wish to know if he were to design an engine mounting or spring suspension utilizing any of these types of synthetic rubber.

■ Manufacturing Problems Using Synthetics

Before proceeding to a discussion of the data to be produced in this paper, a word concerning the difficulties to be encountered in handling and processing these synthetic rubbers in the factory should be in order. It is true that synthetic rubbers possess elasticity and resilience but, as will be seen later, they differ very considerably in their molecular makeup; and with this information, it ought not to be expected that they will process in the rubber factory in exactly the same way as has been done for many years with natural rubber.

It is, of course, only natural that the rubber technologist should wish to have a synthetic rubber which can be substituted with as little difficulty as possible for the natural product through the various stages of milling, extruding, calendering, and forming into the various articles which may be required, such as tires, various types of engine mountings, gaskets, and so on.

Each synthetic rubber constitutes a different chemical compound. They are similar only in certain superficial properties. In regard to some of their properties, as, for example, resistance to solvents such as gasoline, benzene, and lubricating oil, some of the synthetic rubbers are very much better than natural rubber. Some, when properly fabricated into a tire, have been found to show superior abrasion resistance to that of natural rubber. In other properties, such as resistance to very low temperatures and tackiness, they are sometimes quite deficient.

We should look upon synthetic rubber, not as a material which can be universally substituted for the natural product, but as a material having special properties which, when properly handled, will give improved results as compared with natural rubber. On the other hand, we should look upon it as a material whose development and perfection will liberate us from the threat of embargo of natural rubber during the time of national emergency and as a guarantee that the price of the natural product will never again reach the peaks which have characterized it in the past.

With these thoughts in mind, we may proceed with the study of some of the characteristic types of synthetic rubber.

■ X-Ray Structure of Synthetic Rubber

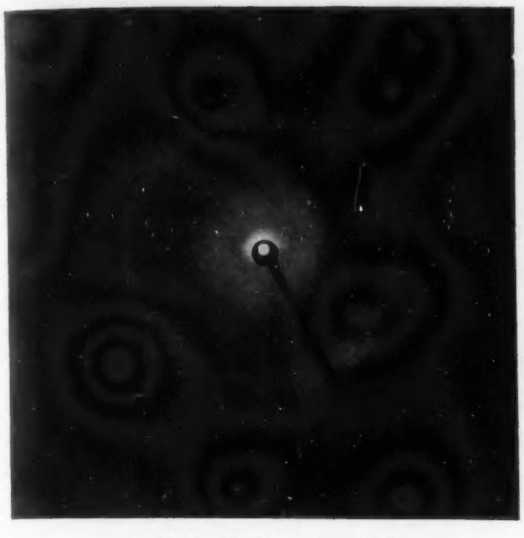
In presenting a series of X-ray diagrams for the various types of synthetic rubber in comparison with natural rubber, in both the stretched and the unstretched condition, it is our purpose to bring out the fact that the molecular structure of the synthetic rubbers is entirely different from that of natural rubber. It is also proposed to review briefly the theories which have been advanced, based on the X-ray analysis of rubber, to account for the elasticity of natural rubber and to advance the possible reason for the variation shown by the X-ray diagrams of synthetic rubber.

At the present time, from the most general point of view, the molecular structure of a rubber-like material is envisaged as a sort of brush-heap structure of entangled long-chain molecules[19]. X-ray diffraction patterns show that, for some rubber-like materials, notable regularities of structure sometimes occur in the tangle of long-chain molecules. It is now realized that these regularities are not essential for rubber-like behavior. Nevertheless, their observation and study are important because they afford a unique opportunity for studying the molecular structure of the chains and the molecular rearrangements which occur upon the application of stress.

Under ordinary circumstances, the X-ray diffraction pattern for natural rubber consists of a broad halo similar to that obtained for liquids. This halo is shown in Fig. 1. Upon stretching, sharp diffraction spots appear as shown in Fig. 2. This phenomenon was first observed by Katz[20]. Since then it has been the subject of many investigations. For further details and the present status of the work, a review article may be consulted[21].

A pattern such as that shown in Fig. 2 indicates the presence of small, ordered crystalline regions. The crystallites are aligned in the direction of the stretching. The explanation of the appearance of these crystalline regions is somewhat involved in conjecture. What happens, apparently, is that, under the action of the applied stress, relatively short lengths of adjacent long chain molecules are straightened and approximately aligned and positioned with respect to each other. As a result of these favorable

[19] See p. 205: "Elasticity, Plasticity, and Structure of Matter," by R. Houwink, Cambridge Press, 1937.
[20] See *Naturwissenschaften*, Vol. 13, 1925, p. 410, by J. R. Katz.
[21] See *Chemical Reviews*, Vol. 26, 1940, pp. 203-226: "Contribution of X-Ray Research to Knowledge of Rubber," by S. D. Gehman.



■ **Fig. 7 – Thiokol unstretched**

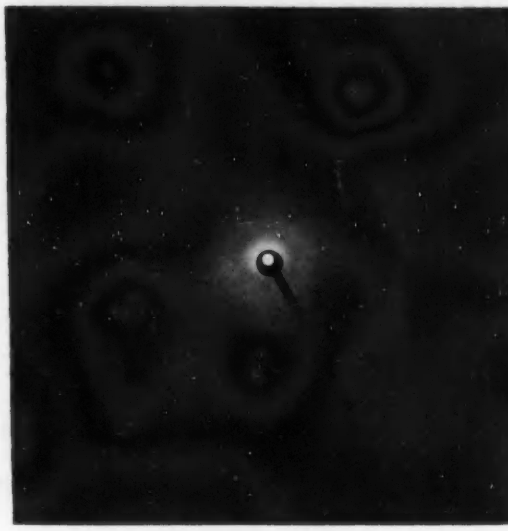


Fig. 8 – Thiokol stretched

circumstances, a spontaneous crystallization can then occur due to intermolecular forces which are sufficiently strong and regular to result in a crystal lattice. A single long chain molecule is thought of as traversing a number of the ordered regions or crystallites[22].

The alignment of the crystallites in the direction of stretching is somewhat analogous to the changes in structure which occur upon the cold working of metals[23]. One of the fundamental differences lies in the fact that crystal grains exist in the metal before the working and are brought into alignment by the working. In the case of rubber, the crystallites are not only aligned, but are brought into existence by the stretching. The formation of the crystallites in stretched rubber is responsible for some characteristic properties of rubber which are analogous to strain-hardening of metals. Thus, stiffening occurs at the higher elongations and the stress-strain curve is concave toward the stress axis. There is a reduction in creep or plastic flow at higher elongations. It is possible to stretch rubber so slowly that crystallites are not formed. Under such circumstances, the tensile strength is greatly reduced[24].

The molecularly ordered state brought about by the stretching of rubber is unstable when the stress is released. Thermal agitation then quickly results in a dissolution of the crystallites and a return of the molecules to the more probable random arrangement. This is accomplished by a rapid retraction and the return of the amorphous, relatively unordered structure[25].

For rubber-like materials which do not form crystallites upon stretching, we can be reasonably certain that a similar straightening and alignment of long-chain molecules occurs upon stretching, with consequences similar to those in the case of natural rubber, although an actual crystal lattice is not formed. There must be some aspect to a rubber-like structure which prevents excessive slipping of the long-chain molecules upon the application of stress. Otherwise, deformation would be of a plastic nature. This necessary rigidity can be introduced into the structure in a number of ways. The formation of crystallites, offering points of anchorage for the long-chain molecules, is only one possibility. Primary valence cross-linkages between the long-chain molecules appears to be the effective means in the case of many synthetic rubbers. In the case of vulcanized rubber, both mechanisms occur, and the relative effects of crystallite formation and cross-linkage in affecting the properties of vulcanized rubber is an interesting subject for experimental investigation[26]. In still other cases, the secondary valence forces between the long-chain molecules may hinder plastic flow to a sufficient extent to give rise to high elasticity.

The formation of crystallites upon stretching does occur in the case of several synthetic rubbers, proving that this characteristic is not necessarily related to the botanical origin of natural rubber. Patterns for unstretched and stretched Vistanex are shown in Figs. 3 and 4. Such patterns were first reported by Brill and Halle[27]. The difference in structure indicated by the patterns is truly remarkable. Neoprene was the first synthetic rubber to exhibit crystallinity upon stretching[12]. The degree of crystallinity, as judged by the sharpness and intensity of the X-ray diffraction spots, appears to be less than in the case of natural rubber or Vistanex. Figs. 5 and 6 are patterns for unstretched and stretched Neoprene, respectively. Some varieties of Thiokol give crystalline fiber diagrams upon stretching[28, 29]. In Figs. 7 and 8 are shown the patterns for a commercial, vulcanized Thiokol stock, unstretched and stretched. Here the halo is possibly sharp enough to indicate some rudimentary crystallization. There is no evidence of orientation, however.

The formation of crystallites upon stretching, such as occurs for stretched rubber, Vistanex, and Neoprene, is apparently possible only when a uniform chemical structure exists in the long-chain molecules and when there are

[22] See "Der Aufbau der Hochpolymeren Organischen Naturstoffe," by K. H. Meyer and H. Mark, Leipzig, 1930.

[23] See ASTM Symposium on Radiography and X-Ray Diffraction Methods, 1937, pp. 302-323: "Application of X-Ray Methods to Problems of Cold Work, Preferred Orientation, and Recrystallization," by J. T. Norton.

[24] See *Kautschuk*, Vol. 14, 1938, pp. 77-79: "The Behavior of Raw Rubber When Stretched Isothermally," by H. Hintenberger and W. Neumann.

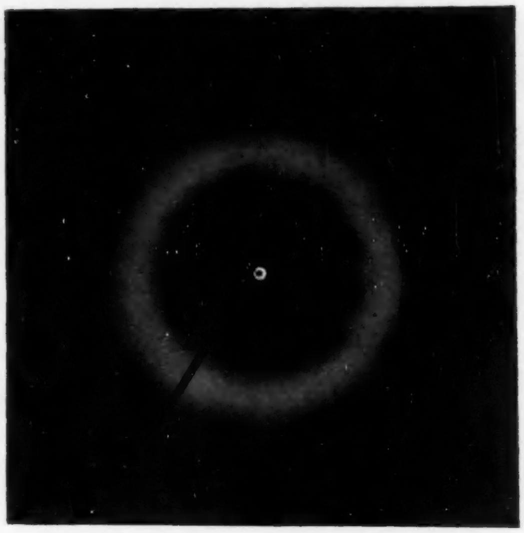
[25] See *Chemical Reviews*, Vol. 25, 1939, pp. 121-135: "Elasticity of Long-Chain Compounds as a Statistical Effect," by H. Mark.

[26] See *Journal of Applied Physics*, Vol. 12, January, 1941, pp. 23-34: "An X-Ray Study of the Proportion of Crystalline and Amorphous Contents in Stretched Rubber," by J. E. Field.

[27] See *Naturwissenschaften*, Vol. 26, 1938, pp. 12-13: "The Rubber-Like Behavior of an Artificial Substance (Oppanol) in X-Ray Light," by R. Brill and F. Halle.

[28] See Transactions of the Faraday Society, Vol. 32, 1936, pp. 77-96: "X-Ray Spectography of Polymers," by J. R. Katz.

[29] See *Chemical Reviews*, Vol. 26, 1940, pp. 143-167: "Investigation of Synthetic Linear Polymers by X-Rays," by C. S. Fuller.



■ Fig. 9 – Buna S unstretched

■ Fig. 10 – Buna S stretched

few, if any, primary valence cross-linkages between the chains. The chemical formulas usually ascribed to these chain molecules are indicated below:

$$\text{Rubber} \quad -CH_2-\overset{\displaystyle CH_3}{\overset{|}{C}}=CH-CH_2-CH_2-\overset{\displaystyle CH_3}{\overset{|}{C}}=CH-CH_2-$$

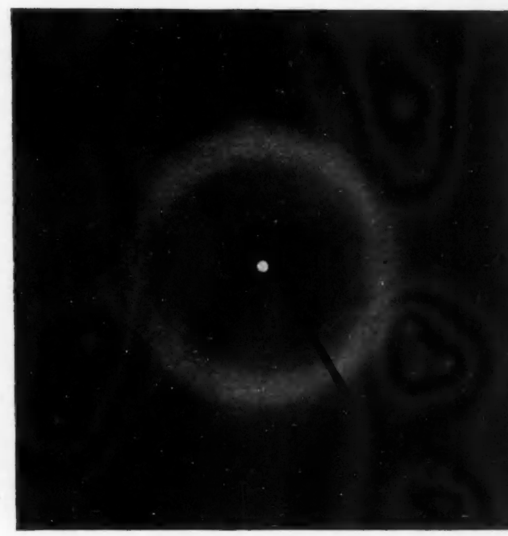
$$\text{Neoprene} \quad -CH_2-\overset{\displaystyle Cl}{\overset{|}{C}}=CH-CH_2-CH_2-\overset{\displaystyle Cl}{\overset{|}{C}}=CH-CH_2-$$

$$\text{Vistanex} \quad -CH_2-\underset{\displaystyle CH_2}{\underset{|}{\overset{\displaystyle CH_3}{\overset{|}{C}}}}-CH_2-\underset{\displaystyle CH_3}{\underset{|}{\overset{\displaystyle CH_3}{\overset{|}{C}}}}-CH_2-\underset{\displaystyle CH_3}{\underset{|}{\overset{\displaystyle CH_3}{\overset{|}{C}}}}-CH_2-\underset{\displaystyle CH_3}{\underset{|}{\overset{\displaystyle CH_3}{\overset{|}{C}}}}-$$

The crystallization which occurs upon stretching is evidence for the orderly arrangement of the methyl groups and chlorine atoms along the chains.

In the case of the polymerization of butadiene, amorphous products have always been reported[28]. As confirmed by the insolubility of the products, what evidently happens here is an extensive cross-linking of the chains. That is, instead of securing long chains (similar to those for rubber, but with the methyl groups replaced by hydrogen atoms), polymerization will proceed only until a chain of limited length is formed, and then a cross-linkage to a neighboring chain will occur at one of the double bonds. The uncontrolled character of these primary valence cross-linkages between the long-chain molecules is presumably responsible for so much irregularity in structure that the formation of a crystal lattice upon stretching is not possible. A number of synthetic rubbers are products of the copolymerization of butadiene and some other monomer or monomers. In such cases, it is most likely that, in addition to cross-linking, the monomers enter the chains in a random fashion so that the possibility of crystallization is entirely precluded.

Patterns for Buna S and Buna N are shown in Figs. 9, 10, 11, and 12. The halo persists even at the highest elongation obtainable. In the case of a Buna N tread stock, a splitting of the halo into two arcs was observed at higher elongations. This can be interpreted as being due to a high degree of alignment of the chain molecules in the direction of stretching without the formation of a three-dimensional lattice. The X-ray diagram for Chemigum, Figs. 13 and 14, indicates the existence of an amorphous structure.



■ Fig. 11 – Buna N unstretched

The formation of crystallites upon stretching, such as occurs for natural rubber, is thus not a necessary characteristic for a rubber-like material, as has been explained previously. It occurs only under favorable conditions of regularity in the long-chain molecules. For the amorphous synthetic rubbers, the information which can be obtained from the X-ray patterns is very much limited. The X-ray diffraction patterns show simply a liquid structure. They do not reveal the molecular basis of their rubber-like elasticity, that is, their capabilities of large extensions and retractions. This is now thought to reside in the straightening and alignment of the long-chain molecules by stretching and the destruction of this relatively ordered arrangement by thermal agitation upon release of the stress. Since the ordered arrangement produced by stress is not perfect enough to result in a crystal lattice, it does not become apparent in the X-ray diffraction patterns.

Only where crystallization occurs can the X-ray diffraction patterns be of much use in understanding the structure. A fundamental limitation of the X-ray diffraction method is, even then, that it gives information on the relative geometrical positions of the molecules, but none directly on the magnitude of the molecular forces involved.

From these considerations, it is apparent that the structure of the synthetic rubbers is at least entirely different from that of natural rubber and that they may owe their rubber-like properties to the operation of a somewhat different mechanism.

Compounding Properties

In this study, attention, as already indicated in the beginning of the paper, will be limited to those synthetic rubbers which are capable of vulcanization. Specifically these are: the Neoprene types, the Buna types, Thiokol, Chemigum, and the Hycar synthetic rubber manufactured by Goodrich. The compounding properties of Neoprene and Thiokol are generally quite well understood. Chemigum and Hycar are new types of synthetic rubber, having in general somewhat the properties of the Buna rubbers. Since Chemigum is a product of The Goodyear Tire & Rubber Co. and is made in three different grades of hardness, the compounding results in the general type of formulas already discussed, are given. All of these rubbers are compared as to their several properties with natural rubber.

From a processing standpoint, the relative workability is indicated in some measure by the extrusion plasticities of the synthetics themselves as shown in Fig. 15. These values are cc per min at 200 psi pressure at 92 C. This chart shows that, of the entire list, only Chemigum I and Neoprene I are really soft and workable of themselves. The remainder of the list is very tough and shows little tendency to knit together or to flow smoothly when placed on a mill. In many cases, notably Chemigums II and III, Hycar OR and Buna N, not a great deal can be done in the way of further softening except by the addition of large quantities of liquid softeners. Severe milling or the use of peptizing agents does not effect any appreciable improvement in the handling of these rubbers. Neoprene GN and Buna S, on the other hand, can be softened by the use of peptizing agents to practically any desired stage. Thiokol can be softened by the addition of small amounts of certain rubber accelerators. By far, the most serious aspect of the successful use of these rubbers is their difficult processability.

With this picture of the condition of these rubbers, themselves, we can proceed to a discussion of their properties in practical compounds. For this purpose we have selected two compounds, one of which represents a tread type and the other a mechanical-goods type of formulation. The essential difference between the two is one of loading, the tread type being loaded with 23 volumes of gas black and the mechanical goods type being loaded with 50 volumes of a semi-reinforcing soft black commonly used in this type of formulation.

The tread type will be discussed first, and on Fig. 16 are shown the tensile strengths of the series. All of the synthetics, with the exception of Thiokol, show excellent to fair tensile values in comparison to natural rubber, indi-

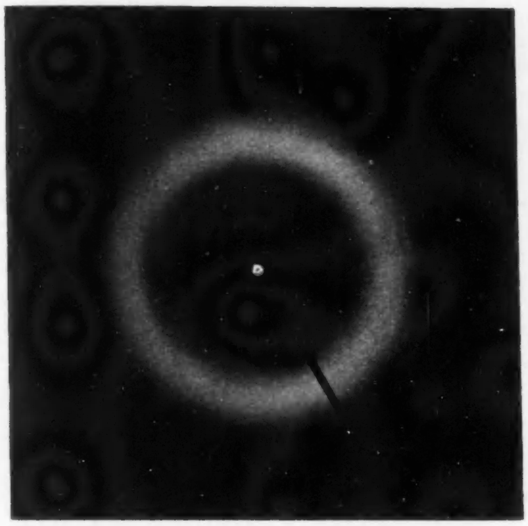


Fig. 12 – Buna N stretched

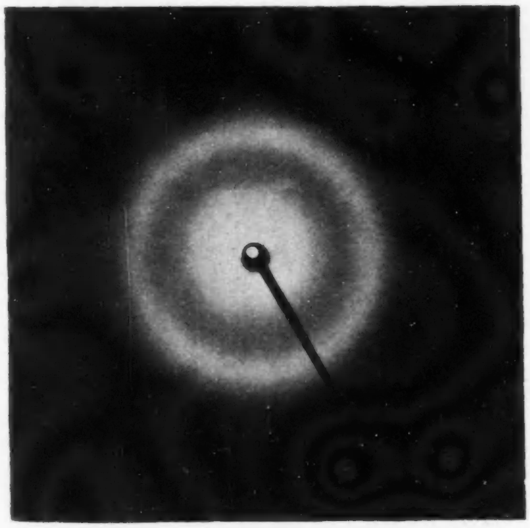


Fig. 13 – Chemigum unstretched

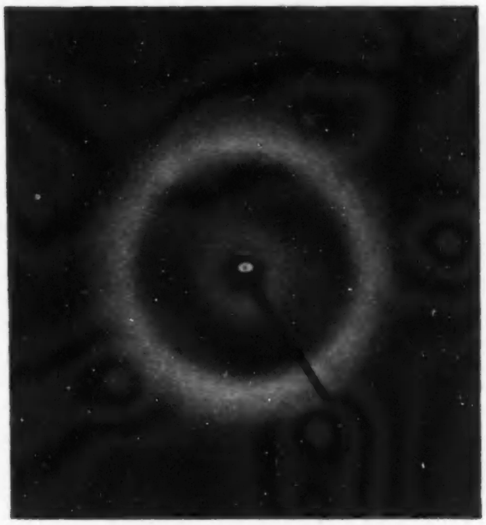


Fig. 14 – Chemigum stretched

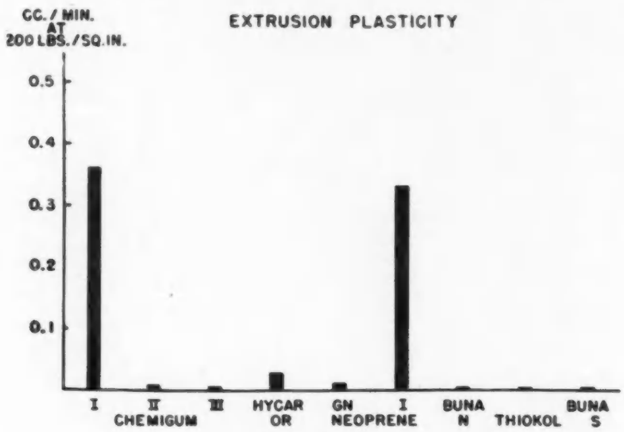


Fig. 15

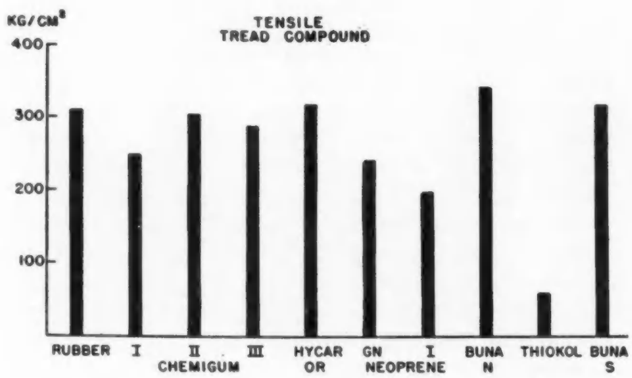


Fig. 16

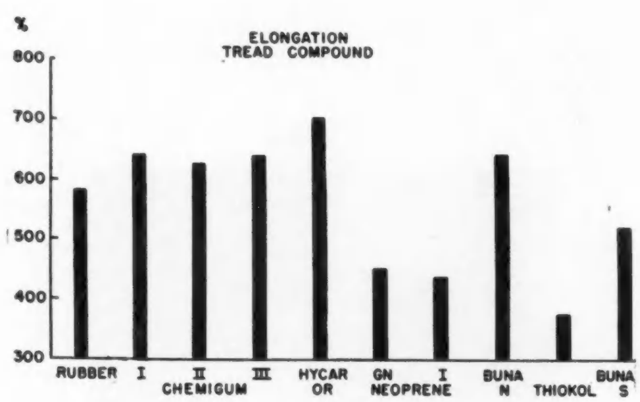


Fig. 17

cating good quality in this respect. Elongations at break corresponding to these tensile values are shown in Fig. 17, and here again it may be concluded that practically all of the series, with the exception of Thiokol, show sufficient elongation to cause no great concern. Neoprene GN can probably be softened to give greater elongation than was obtained in this case.

Fig. 18 shows the modulus at 300% elongation. This value generally reflects the stiffness or resistance to deflection, and where large variations are involved, is reflected in the durometer hardness shown in Fig. 19. Aside from the Neoprenes, these values also compare favorably with natural rubber. The Neoprene GN could probably be further softened to bring it more in line with the rest.

Fig. 20 shows rebound values, and here becomes apparent one of the deficiencies of synthetic rubber. Rebound is expressed as the percent energy returned to a standard pendulum falling from a given height. Rebound is a measure of energy absorption and is a fair index of the tendency for heat generation and build-up where the compound is subjected to deflection cycles at a rapid rate. No one of the synthetics is the equal of natural rubber in this respect, and small differences quickly become apparent where service is severe. It is not without considerable development effort that the synthetics have been made as good as here shown, and for many uses this is satisfactory and adequate.

Resistance to flexing is shown in Fig. 21. The values are minutes to failure for a special type of test piece stretched to 60% elongation and returned to 0% elongation and given a pseudo compression of 45%. The compression is not actually that since the piece is free to bend upward through this portion of the cycle. Here again the synthetic rubbers are not the equal of natural rubber, but definite progress has been made in adapting them to uses where this property is a particular requisite. These flexing tests were made on the standard Goodyear flexing machine. The effect of permanent set on the synthetic rubber test pieces has not, as yet, been definitely determined, and it may be that, if the set is appreciable, it will affect the values given.

Fig. 22 shows the per cent volume increase when the series of compounds is immersed in a standard motor gasoline for 12 days. Rubber, as would be expected, is very bad. Chemigums I, II, and III are progressively better, Chemigum III being very good, as is also the Hycar.

Grasselli abrasion loss, shown in Fig. 23, while not a reliable index of service abrasion, indicates such a degree of superiority for the synthetics over natural rubber that there is a reasonable justification to expect a fairly good performance from the synthetics. This has been proved in a number of cases where the synthetics actually showed a slight superiority to natural rubber in road-wear abrasion measurements of tire treads.

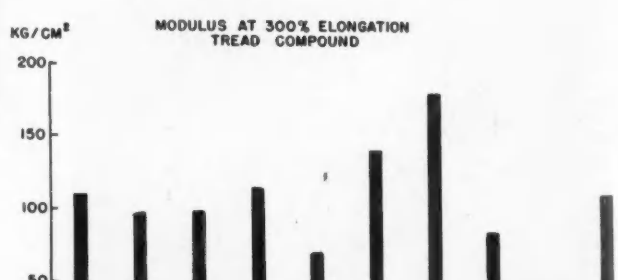


Fig. 18

The tensiles and elongations of the mechanical-goods type of formulation are shown in Figs. 24 and 25, respectively. Here again many of the synthetics show equal or better quality than natural rubber. Shore hardness, shown on Fig. 26, indicates these compounds to be harder than might generally be used for this purpose, but they can be softened readily within reasonable limits by the addition of the proper softeners. The ability of a synthetic to take a given loading with a minimum increase in hardness is an important characteristic. It was for this reason that this compound was not complicated by the use of softeners.

The swelling of this type of compound in a standard gasoline is shown in Fig. 27. Rubber is very poor. Buna S, which is not recommended for this purpose, is next in order. Neoprene I is considerably better than GN and compares with Chemigum I. Chemigum II and Buna N are comparable and Chemigum III, Hycar OR and Thiokol are the best of the series.

The swelling values for benzol are shown in Fig. 28. Chemigum III and Hycar are the best of this series.

Summarizing our observations in a tire-tread type of formulation, it may be concluded that, with the exception of Thiokol, they all may be used for tire treads where the service requirements are not too severe. For severe service, Chemigums I and II and Neoprene GN are to be recommended, but they cannot at present be expected to equal natural rubber in the severest type of service. For other applications, where a relatively high-quality stock such as a high-quality mechanical goods stock is desired, these synthetics have distinct possibilities and offer the added feature of superior resistance to swelling when exposed to gasoline and other solvents.

The problems involved in the compounding of these various types of synthetic rubbers have already been discussed.

Dynamic Tests

In the description of a new dynamic test and the data which we have obtained on various synthetic rubbers in this test, it is our purpose to supply sufficient information on the physical characteristics of the various synthetic rubbers as would allow an engineer properly to design certain types of vibration and automotive equipment in which he might be interested.

In the case of synthetic rubber, it is especially important to have tests which can evaluate the elastic properties of

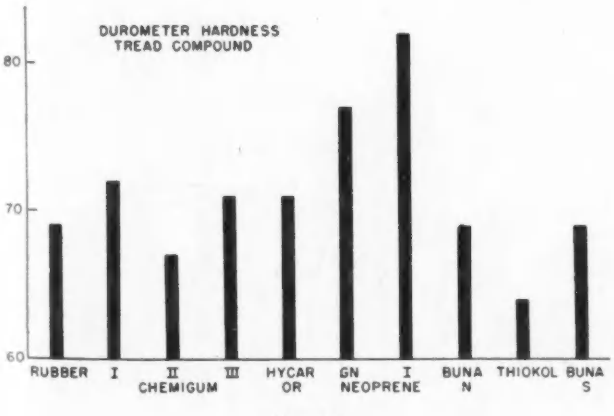


Fig. 19

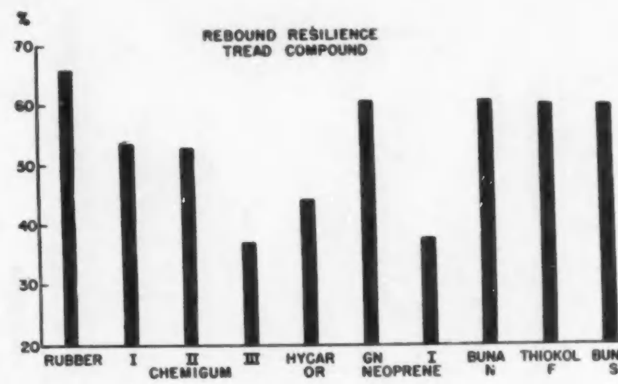


Fig. 20

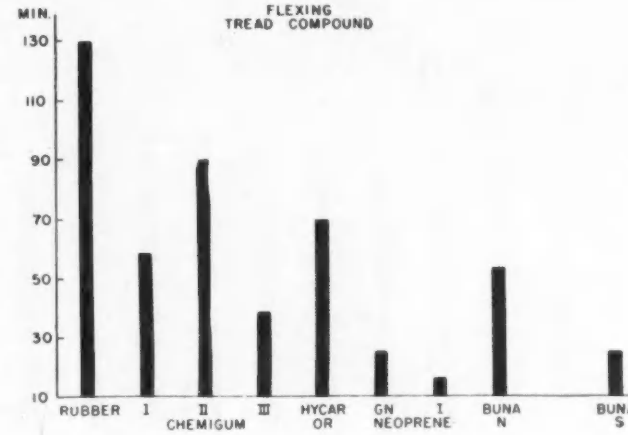


Fig. 21

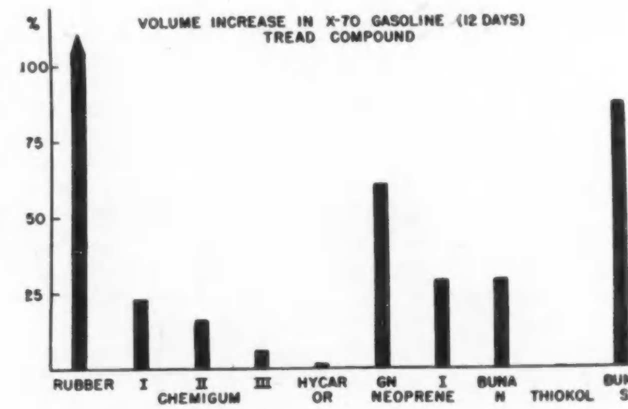


Fig. 22

the stocks in a range of low deformations corresponding to those encountered in most applications rather than to rely on the indications from ultimate values of tensile strength and elongation entirely.

This point is illustrated by some data published by Rohde[30].

Hysteresis loops, secured in the normal manner with a tensile testing machine, indicated that, after ten cycles, the

[30] See *Kautschuk*, Vol. 15, 1939, pp. 64-68; *Rubber Chemistry and Technology*, 1939, pp. 799-804: "Characterization of the Plastic-Elastic State," by E. Rohde.

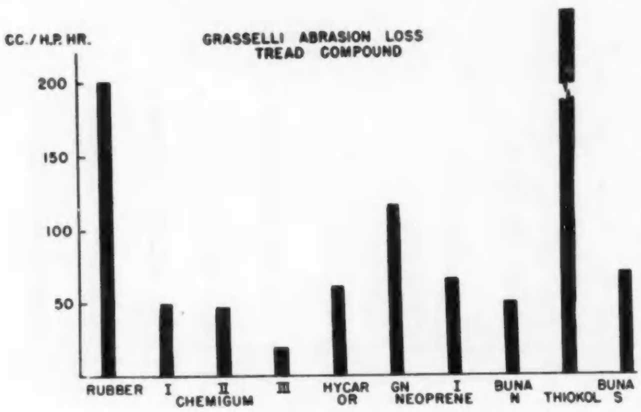


Fig. 23

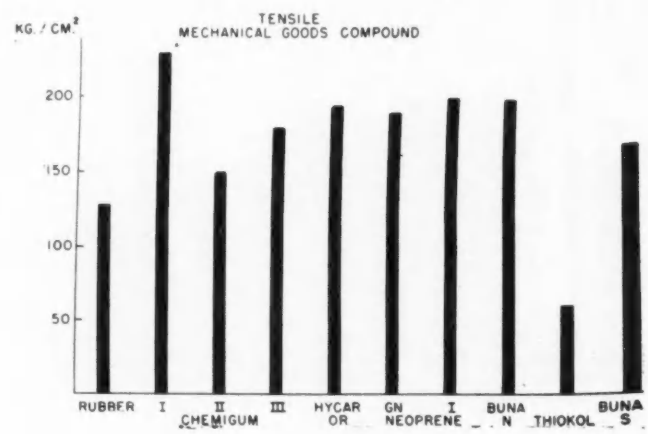


Fig. 24

energy loss for a natural rubber tread stock was about six times as great as for a Buna S tread. In actual tests on tires, the Buna S tread, however, did not run at a lower temperature when compared with the natural rubber tread.

Hysteresis loops were then obtained for various ranges of elongation. It was found that, for the tread stocks tested, natural rubber was superior in elasticity and work capacity for tensile stresses below 35 kg per sq cm. Only above this stress was Buna S superior. Consequently, this advantage did not show up when the stocks were employed as tire treads.

In the same vein, many of the apparently contradictory test results for synthetic rubbers will probably, at some future time, find a logical explanation. Wherever there is such an interaction of plastic and elastic properties as occurs for the synthetic rubbers, it becomes necessary in testing to define closely many variables which otherwise might not be so important. These include the time rate of the deformation, the character of the deformation, the range of stresses and elongations, and the temperature.

Several dynamic tests have been described and applied to synthetic rubber stocks[31, 32, 33, 34]. In the Goodyear research laboratories, a modification of the method of Naunton and Waring has been developed by R. B. Stambaugh and S. D. Gehman and used extensively for measurements of dynamic properties of synthetic rubber stocks. Fig. 29 is a photograph of the apparatus. As in the apparatus of Naunton and Waring, two opposed rubber test pieces are set into forced vibration by means of an alternating current which passes through a coil in a strong magnetic field. The resonance frequency for the system, at which relatively large amplitudes of vibration occur, depends upon the stiffness of the rubber and the mass of the vibrating system.

The procedures used in the test differ considerably from those described by Naunton and Waring. The vibrator is mounted horizontally in order to obtain the same compression in the two test pieces used. The test pieces are vibrated under a fixed static compression of 8%. The measurements are made at a constant frequency of 60 cycles per sec, which is in contrast to the higher and variable frequencies used by Naunton and Waring. The use of a constant frequency is made possible by tuning the system to resonance by variation of the vibrating mass. Enough power is supplied to the system to secure amplitudes which can be read directly by means of a traveling

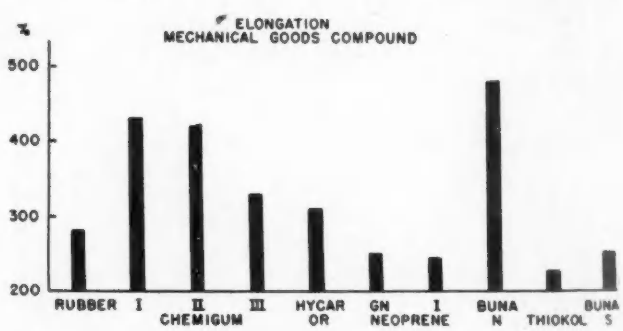


Fig. 25

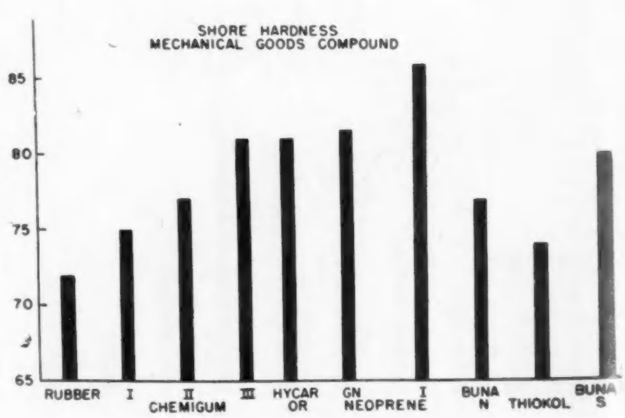


Fig. 26

[31] See ASME Transactions, Vol. 62, 1940, pp. 469-474; also ASTM Proceedings, Vol. 39, 1939, p. 1180: "Neoprene as Spring Material," by F. L. Yerzley.

[32] See Proceedings of the Rubber Technology Conference, Institution of the Rubber Industry, 1938, p. 821: "Dynamic Evaluation of Damping and Durability of Rubber Compounds," by H. Roelig.

[33] See Proceedings of the Rubber Technology Conference, Institution of the Rubber Industry, 1938, pp. 805-820: "Fatigue in Rubber, Part II," by W. J. S. Naunton and J. R. S. Waring.

[34] See Transactions of the Institution of the Rubber Industry, Vol. 14, 1939, pp. 340-364: "Fatigue in Rubber III – Fatigue and Reenforcement," by W. J. S. Naunton and J. R. S. Waring.

microscope. Naunton and Waring employed a vibration pick-up and amplifier for this purpose.

The alternating driving force for the vibration is determined by the a-c current through the coil. The coil is calibrated by ascertaining the force necessary to hold it in the equilibrium position for various d-c currents. A straight-line current-force calibration is obtained.

The equation of motion for the system is,

$$m\frac{d^2x}{dt^2} + b\frac{dx}{dt} \times sx = F\cos pt \qquad (1)$$

m = vibrating mass.

b = proportionality factor between velocity and frictional force.

s = spring constant of system.

F = maximum value of driving force.

p = angular frequency = $2\pi \times$ cycles per sec.

t = time.

x = displacement.

In the calculations, all these quantities are expressed in cgs units.

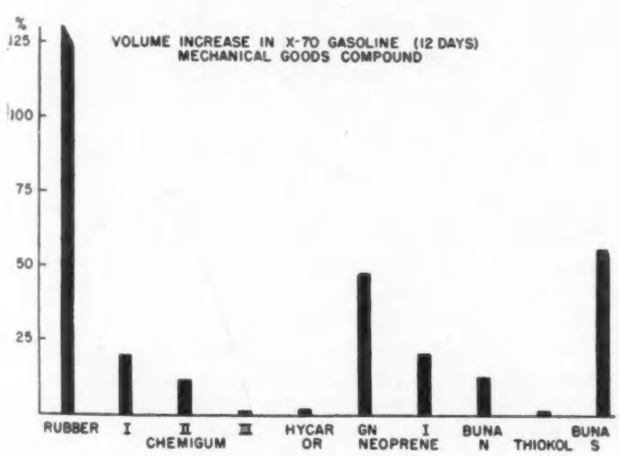


Fig. 27

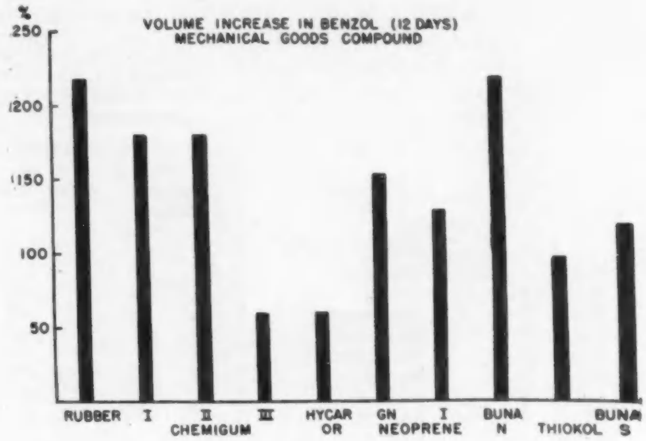


Fig. 28

The solution of the above equation gives,

$$x = \frac{F\cos(pt-\theta)}{\sqrt{(s-mp^2)^2+b^2p^2}}\ ;\ \tan\theta = \frac{bp}{s-mp^2} \qquad (2)$$

At resonance, it can be assumed that,

$$s = mp^2 \qquad (3)$$

Hence, the dynamic stiffness is determined by the vibrating mass, p being constant and equal to $120\ \pi$.

Also, at resonance, to a sufficient degree of accuracy,

$$X_{res.} = \frac{F}{bp} \qquad (4)$$

The amplitude at resonance, $X_{res.}$ is measured, and F, the driving force, is known from the driving current. Hence, b can be calculated.

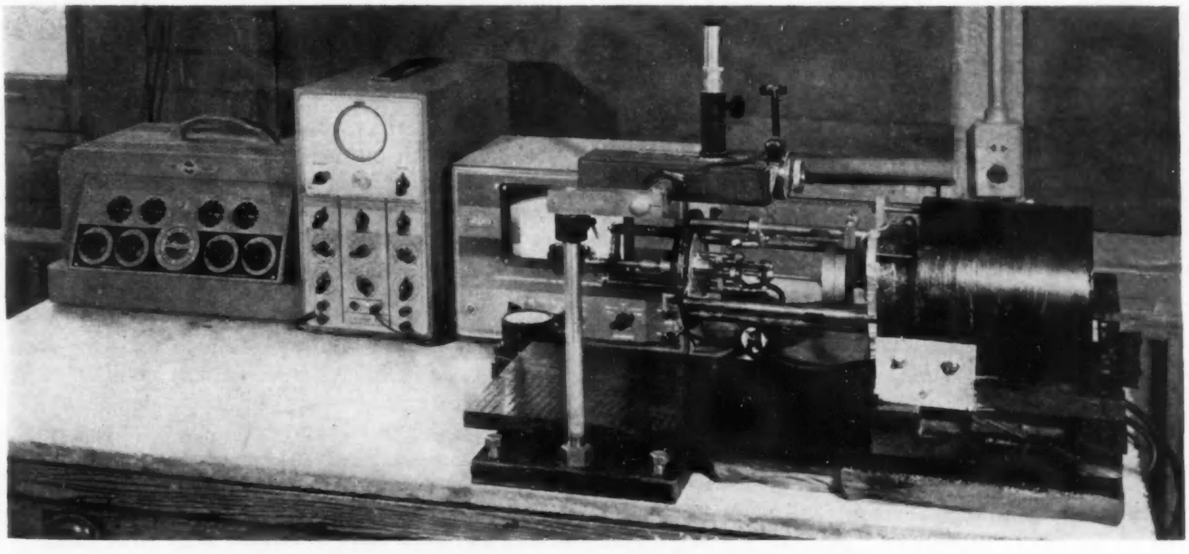
From the value of s, the dynamic modulus can be calculated from the formula,

$$E = \frac{sh}{2A} \qquad (5)$$

E = dynamic modulus

A = area of end of test piece

h = height of test piece.



■ Fig. 29 — Goodyear apparatus for measuring dynamic properties of synthetic rubber stocks

Table 1 – Results of Dynamic Tests

Test pieces: ½ in. high x ½ in. diameter.
Frequency: 60 cycles per sec.
Static load: 8% compression.
Driving force: 1.7 x 10^6 dynes (Current: 2.5 amp)

Compound No.		Type of Rubber	Type of Compound	Amplitude at resonance (mm)	Dynamic modulus kg/sq cm	Internal friction, cgs units x 10^{-3}	Resilence, %	H_f (f = 1.73 kg)	H_x (x=0.15 mm)
T-18		Natural rubber	Tread	0.900	76	25	45	100	100
R-337	P-735	Chemigum I	Tread	0.315	148	72	31	64	140
R-340	P-12	Chemigum II	Tread	0.170	286	132	33	33	240
R-340	P-13	Chemigum III	Tread	0.042	378	544	3	35	440
R-339	P-19	Neoprene I	Tread	0.068	350	334	10	36	380
R-339	P-18	Neoprene GN	Tread	0.325	203	69	42	40	159
R-340	P-10	Thiokol	Tread	0.337	106	40	40	78	60
R-340	P-21	Buna N	Tread	0.364	159	62	39	53	135
R-340	P-23	Buna S	Tread	0.448	148	51	44	53	123
R-337	P-736	Hycar	Tread	0.260	143	87	23	73	144
R-340	P-15	Natural rubber	Mech. Goods	0.338	151	67	34	60	126
R-339	P-23	Chemigum I	Mech. Goods	0.148	303	152	30	32	262
R-340	P-17	Chemigum II	Mech. Goods	0.128	318	177	26	32	291
R-340	P-18	Chemigum III	Mech. Goods	0.023	671	1000	1	20	810
R-339	P-22	Neoprene I	Mech. Goods	0.046	642	368	25	16	580
R-339	P-21	Neoprene GN	Mech. Goods	0.150	315	150	32	30	264
R-340	P-20	Thiokol	Mech. Goods	0.171	251	132	28	40	223
R-340	P-19	Buna N	Mech. Goods	0.158	295	143	31	33	252
R-340	P-22	Buna S	Mech. Goods	0.125	356	181	29	27	310
R-340	P-14	Hycar	Mech. Goods	0.038	399	602	3	34	462
R-200	D-105	Chemigum II	20 parts gas	0.690	86	33	40	96	96
R-200	D-106	Chemigum II	30 parts black	0.353	132	64	31	72	105
R-200	D-107	Chemigum II	40 parts black	0.165	207	137	20	53	203

The coefficient of normal viscosity discussed in vibration theory[35] is used as a measure of the internal friction. This is given by the equation

$$\eta = b \frac{h}{2A} \tag{6}$$

η = coefficient of normal viscosity.

The other letters have been previously defined.

Naunton and Waring used for the resilience the ratio calculated for the amplitudes of two successive free vibrations of the system. The vibrational energy, however, is proportional to the square of the amplitude[35], and we have taken this as a measure of the resilience.

From vibration theory (see reference in the preceding paragraph), this ratio is given by

$$R = 100\, e^{-\frac{2\pi b}{mp}} = 100\, e^{-2\pi p \frac{\eta}{E}} \tag{7}$$

The calculations from the experimental results are very simple. For a sample size of ½ in. high by ½ in. in diameter, constant frequency of 60 cycles per sec, constants can be calculated for equations (3), (4), (5), (6) and (7) to give

$$E = 0.0725\, m \tag{8}$$

m is the mass of the vibrating system in grams for resonance at 60 cycles.

E is the dynamic modulus in $\frac{\text{kg}}{\text{cm}^2}$

$$\eta = \frac{902}{X_{res.}} i \tag{9}$$

η is in cgs units; i is the rms current in amperes, and $X_{res.}$ the amplitude at resonance in cm. The constant depends upon the force calibration of the coil.

$$R = 100\, e^{-2.42 \times 10^{-3} \frac{\eta}{E}} \tag{10}$$

Formulas for calculating the relative heat generation when comparing stocks for the same cyclic deflection and for the same cyclic force have been worked out by Naunton and Waring[34]. They are,

$$H_f \propto \frac{(100 - R)\, F^2}{E} \tag{11}$$

for the same force F, and

$$H_x \propto (100 - R)\, EX^2 \tag{12}$$

for the same deflection, X.

Table 1 gives some data secured from the test for a variety of synthetic rubber compounds.

It is found that values of dynamic resilience from the test correlate rather closely with the per cent rebound as determined by the rebound pendulum. Fig. 30 shows a plot of per cent rebound against per cent dynamic resilience for a wide variety of synthetic rubber stocks. It will be noted that, for these stocks, the resilience varies over a range from 3% to 48%, whereas the rebound has a narrower range, 28% to 65%. In general, the dynamic resilience as determined from the test gives lower values and a larger spread between results for similar stocks than does the pendulum rebound test.

The calculated relative heat generation rates for the same cyclic force, given in the next-to-the-last column of Table 1, show an advantage for the synthetic rubber stocks as compared to natural rubber, due principally to the higher values of the dynamic modulus for the synthetic

[35] See ASME Transactions, Vol. 51, 1929, pp. 227-236: "Vibration Damping Including the Case of Solid Friction," by A. L. Kimball.

rubber stocks. For the same force, this results in much smaller deflections and hence low heat generation.

The calculated relative heat generation rates for the same amplitude of vibration are given in the last column of Table 1. The calculation of H_x is complicated by the fact that the dynamic modulus and resilience depend somewhat on the amplitude, as was noted by Naunton and Waring[33]. Hence, values of E and R from Table 1 cannot be used directly in Equation (12), but must be corrected to the same value of the amplitude. The value of the amplitude used for the comparison of the stocks was 0.15 mm. The natural rubber tread stock was taken as the standard. At this amplitude, E had a value of 129 kg/sq cm and R a value of 37%. Corrections were made, when necessary, to the values of E and R for the other stocks to allow for differences in amplitude at which E and R had been determined.

The data for the last three compounds in Table 1 show the controlling influence of the gas-black loading on the dynamic properties. In this case only about half of the loading of gas black was required to give the synthetic rubber stock the same dynamic modulus as the natural rubber tread stock. For this loading, the resilience is nearly as high as for the natural rubber tread stock. For the same loading of gas black, however, the synthetic rubber gave a much stiffer and less resilient tread stock than natural rubber.

The foregoing data on the relative effect of increased carbon black loading show that, with a low loading, the properties of natural rubber may be closely approximated and, for uses where high resilience and low internal friction, with resulting low heat build-up, are required, it undoubtedly would be most feasible to reduce the carbon-black loading as indicated. It would, of course, be more expensive to use this type of compounding in a mechanical-goods part than the type which has always been used with rubber. However, this fact illustrates very forcefully the fact that synthetic rubber cannot be substituted for natural rubber on a quantitative basis nor on an equal-cost basis without examining carefully the physical properties which are to result from such a procedure.

Summary

(1) The difficulties in substituting synthetic rubber for natural rubber have been pointed out.

CORRELATION OF DYNAMIC RESILIENCE WITH PENDULUM REBOUND

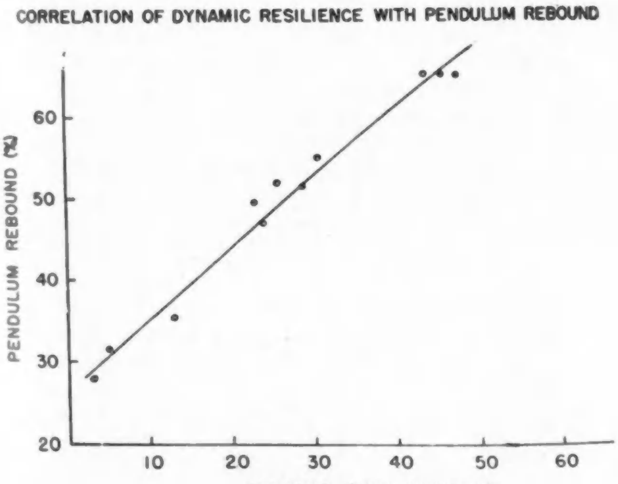


Fig. 30

(2) The differences between natural and synthetic rubber, as shown by X-ray analysis, have been discussed. The exact molecular mechanism by which synthetic rubbers exhibit elasticity has not yet been definitely established, but a proposal to account for it has been advanced.

(3) Typical compounding results for several vulcanizable synthetic rubbers in two different formulas have been given.

(4) The results of a new dynamic test on the various synthetic rubbers under discussion and the two different formulas used have been given and their possible use in design has been suggested.

Acknowledgment

The authors wish to acknowledge the aid and cooperation of M. J. DeFrance, S. D. Gehman, and J. E. Field of the Research Laboratories of The Goodyear Tire & Rubber Co., in the preparation of this paper.

BRAKE-DRUM MATERIALS

CAST iron, whilst possibly the best of cheap brake-drum materials, is still an indeterminate. Cast-iron brake drums of practically identical chemical composition may not possess the same structural make-up or the same physical properties. One batch may exhibit coarse graphite and some ferrite in a base structure of pearlite; another batch made from the same mixture and the same analysis may present an open pearlitic or even a closely pearlitic base structure with a different form of graphite according to the degree of metallurgical control operating in any particular foundry and according to their methods of pouring and molding.

Mostly all brake-drum irons made with alloys, particularly those containing chromium, contain some free cementite, and each structure responds very differently to scoring, thermal checking, and ultimate service life.

The primary considerations in obtaining a good brake-drum material are:

1. To select the correct structure in combination with the best mechanical properties for the particular service conditions to be encountered.

2. Uniformity – that is, every brake drum must have the same structure regardless of design or dimension.

In the absence of these, fancy compositions of irons, the indiscreet use of alloys, the claim for high strength tests and the loose use of metallographic terms are apt to be more misleading than informative.

For example, brake drums on a logging trailer in the mountainous regions of the Northwest call for an exceedingly hard wearing iron. The service is so severe on the long mountain grades that a tank of water is mounted on the trailer for the purpose of cooling off the drums.

Such service and treatment produce both severe wear and premature thermal cracking.

Thus it will be seen that an iron successful in one kind of service may fail in another.

Excerpts from the paper: "Brake Drums – Gray Cast Iron and Meehanite," by Oliver Smalley, president, Meehanite Research Institute of America, presented at a meeting of the Pittsburgh Section of the Society, April 22, 1941.

The M.I.T.-Wright Brothers and its

by JOHN R. MARKHAM
Associate Professor of Aeronautical Engineering, Massachusetts Institute of Technology

WITH the development of airplanes of large size and high speeds the requirements for a satisfactory wind tunnel, in which aerodynamic research and investigations are to be carried out, have become difficult and expensive to fulfill. The first requirement – that the force and moment coefficients found from model measurements should be directly applicable to full scale – is economically impracticable. These coefficients as found in a wind tunnel test are only directly applicable to full scale when the Reynolds Number of the tests is the same as for flight. Full-scale Reynolds Numbers for the modern high-speed airplanes are now of the order of (2×10^7) twenty million. The duplication of such Reynolds Numbers in a wind tunnel presents a large financial problem as well as an engineering one.

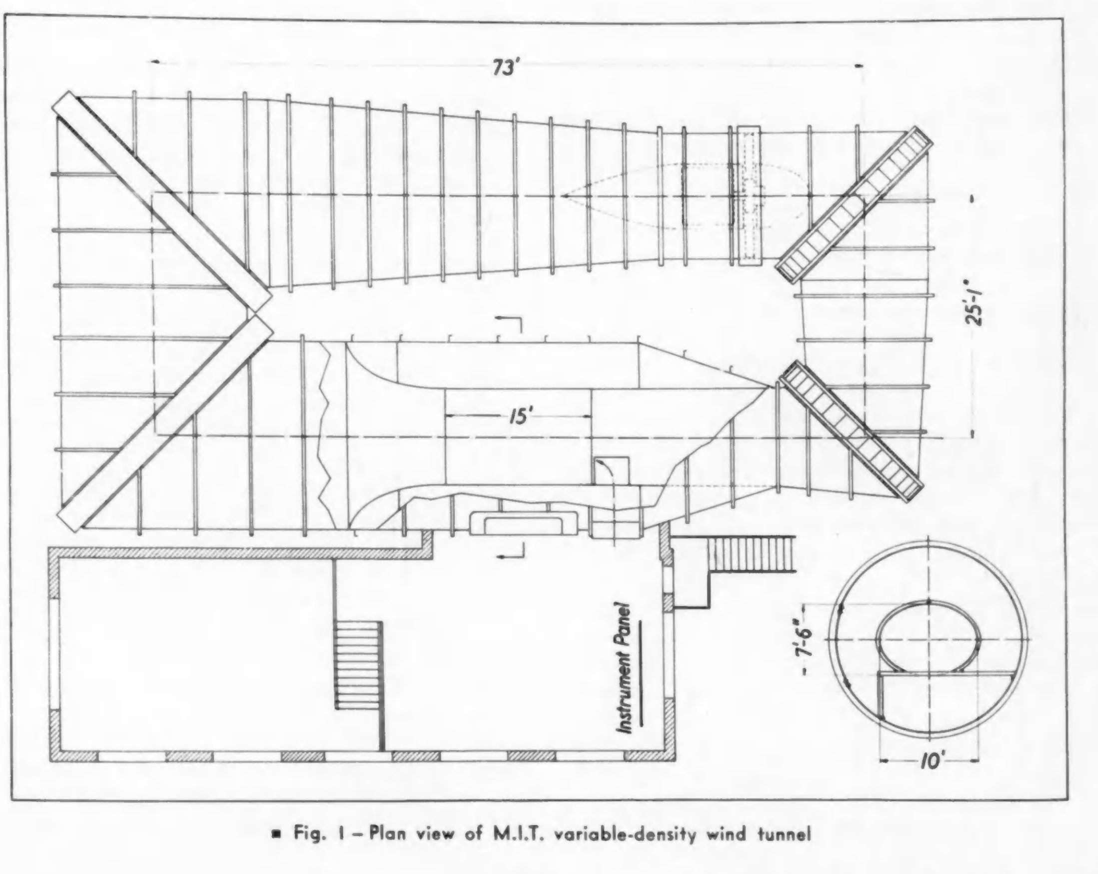
Reynolds Number is the non-dimensional term of velocity times density of the fluid times a characteristic dimension of the object, divided by the viscosity of the fluid; in symbols:

$$R = \frac{\rho v l}{\mu}$$

Fortunately, the aerodynamic coefficients vary slowly at high values of Reynolds Number so that test Reynolds Numbers of the same order of magnitude as those of full scale are satisfactory.

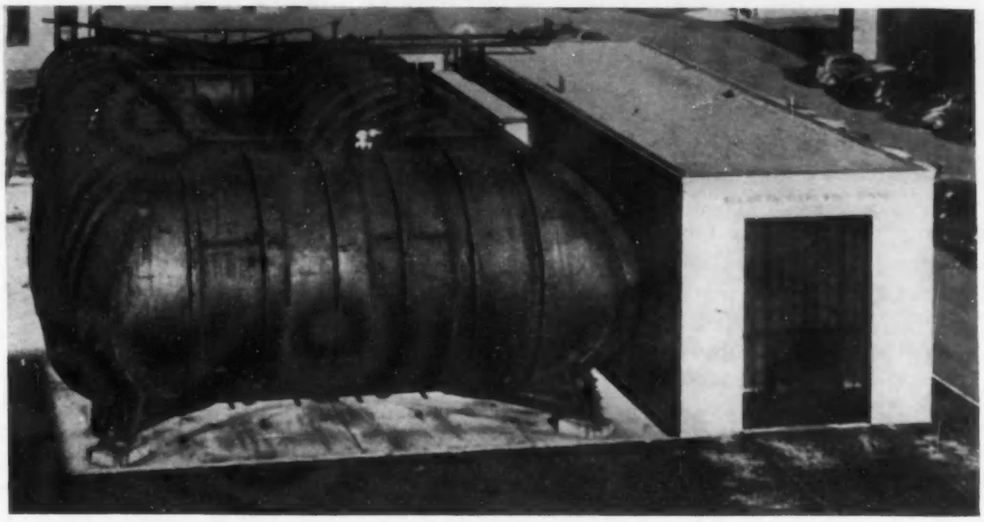
The manner in which the variables that are contained in the Reynolds Number may be changed are seen readily. If one quantity is made smaller than for actual conditions,

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 6, 1941.]



■ Fig. 1 – Plan view of M.I.T. variable-density wind tunnel

WIND TUNNEL Operating Equipment



▪ Fig. 2—General view of wind tunnel

WITH the M.I.T.-Wright Brothers wind tunnel and balance system, Prof. Markham explains, it is possible to obtain measurements very rapidly on a complete airplane model from which the coefficients for all six components can be plotted with a minimum of calculation. It is usually possible, he continues, to mount the model so that the point representing the center of gravity of the airplane is on the balance axis, a feature which increases the accuracy of the results and the simplicity of the calculations.

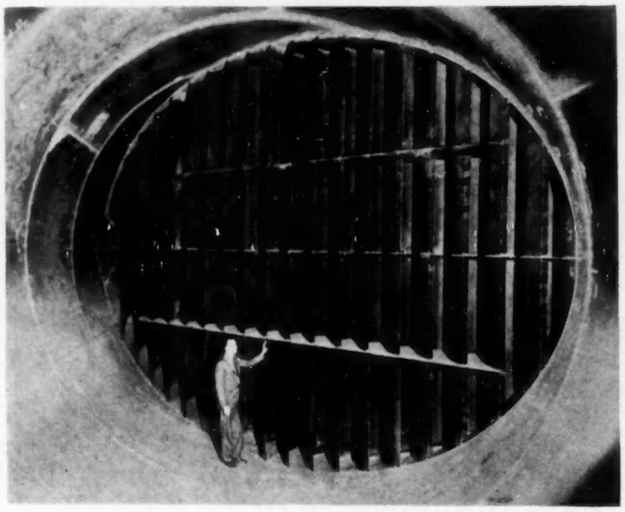
This balance system was designed to fulfill the following general specifications: It had to be compact for the pressure-type tunnel in which it operates requires that the shell surrounding the test section be kept to a minimum diameter; it should be arranged so that measurements of all six components could be made remotely and that angular settings of the model in pitch and yaw could be made from a remote station. It was required further that each component be measured independently and totally so as to simplify and shorten the numerical calculations.

The test section was made an ellipse in order to obtain as large a span-wise dimension as possible for a given test-section area. It is then possible without requiring extra power, to test a model with a larger span in an elliptical tunnel than in a tunnel of circular cross-section of the same area at the same speed.

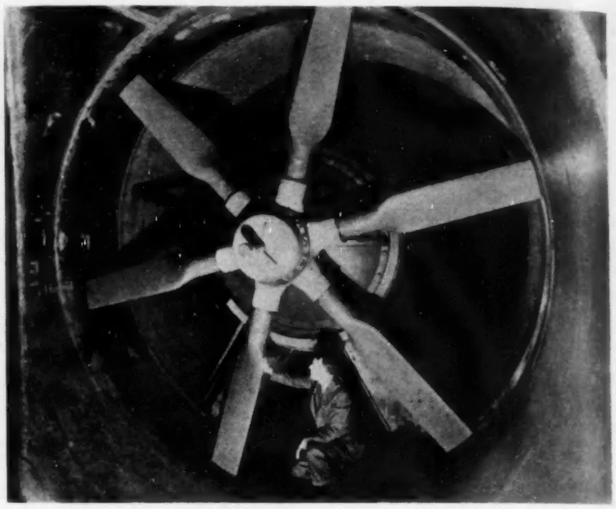
one or more of the remaining quantities must be increased. If the diameter of the tunnel is made small, the scale of the model must be small, and a high velocity becomes necessary. The power required varies rapidly with the velocity, and initial and operating costs limit the power which may be used. The density, however, may be increased considerably with only a moderate increase in the initial and operating costs of the tunnel if the tunnel is to be of the closed-return type. The density being proportional to the pressure, the use of a pressure wind tunnel presents the possibility of higher Reynolds Numbers. If the shell is made of steel, it can be made strong enough to withstand a fairly high pressure without increasing the cost too much. Of course, the high density will reduce the speed obtainable for a given power, but only as the cube root of the density. Reduction of pressure will permit higher velocities for a given power; thus, with the addition of sufficient stiffeners, the same tunnel may be operated at very high speeds for certain kinds of research.

These general considerations led to the decision of the Department staff responsible for the design to select a closed-return tunnel with an elliptical throat 10 ft x $7\frac{1}{2}$ ft with an airspeed of 250 mph at atmospheric pressure, but which could be operated at pressures of 4 atmospheres and $\frac{1}{4}$ atmosphere. This was a compromise between cost and research requirements. Figs. 1 and 2 show the general dimensions and outline of the tunnel.

Arc-welded steel plate construction was used for simplicity of joints and tightness of seams. Plate thicknesses varying from $\frac{1}{2}$ in. to $\frac{3}{4}$ in. are adequate for the pressures above atmospheric, and the addition of a sufficient number of external stiffening rings made the shell strong enough to withstand the collapsing pressure when internal pressure is $\frac{1}{4}$ atmosphere.



■ Fig. 3 – Guide vanes



■ Fig. 4 – Propeller and motor

Having established the size of the test section, the geometry and the dimensions of the remaining parts of a closed-return wind tunnel are fairly well defined from considerations of minimum energy losses. For this tunnel, however, the centerline length was arbitrarily shortened to reduce the cost of construction. As a consequence, diffuser and corner losses are somewhat higher than need be. There results a lower than optimum energy ratio for the tunnel. This is reflected in the operating power costs.

The test section has been made an ellipse in order to obtain as large a span-wise dimension as possible for a given test area. It is thus possible without requiring extra power to test a model with a larger span in an elliptical tunnel than in a tunnel with a circular cross-section of the same area, at the same speed. In addition, the bottom section of the ellipse has been made flat to simplify the balance support and fairing installation. In order to eliminate the difficult problems of transmitting the forces on the model from the test section to the measuring system without the need for airtight and frictionless joints between the supporting elements and the shell of the test section, the entire test section is surrounded by another shell, extending from the large end of the nozzle to a point just ahead of the first corner. By venting the test section the annular compartment surrounding the test section is at the same static pressure as the airstream. With this arrangement the supporting elements for the model can easily be passed through the walls of the test section with suitable clearances without causing any appreciable effect on the airstream. The test section thus carries relatively small pressure loads and hence can be constructed of wood, which facilitates changes in suspensions which may be necessary for certain kinds of tests.

Entrance to the pressure chamber is provided by two doors. One is a large opening to make possible the construction of the test section and the installation of the balances. This door is normally kept closed and sealed by a gasket and many bolts. The smaller door, which is pivoted on swinging hinges and can be tightly closed in a few seconds, provides normal access to the tunnel. Opposite this door is a sliding one in the diverging cone which permits entrance to the test section proper.

Transition from the elliptical section at the end of the test chamber to a circular cross-section is made in the diffuser before the first corner. The remainder of the tunnel is circular in cross-section except for a transition back to the ellipse at the entrance to the test section. The tunnel diameter gradually increases until the maximum diameter of 19 ft 4 in. is reached at the third corner. Thereafter the diameter is constant until the nozzle is reached. The contraction ratio, the ratio of the areas at the entrance and exit of the nozzle, is approximately 5.0.

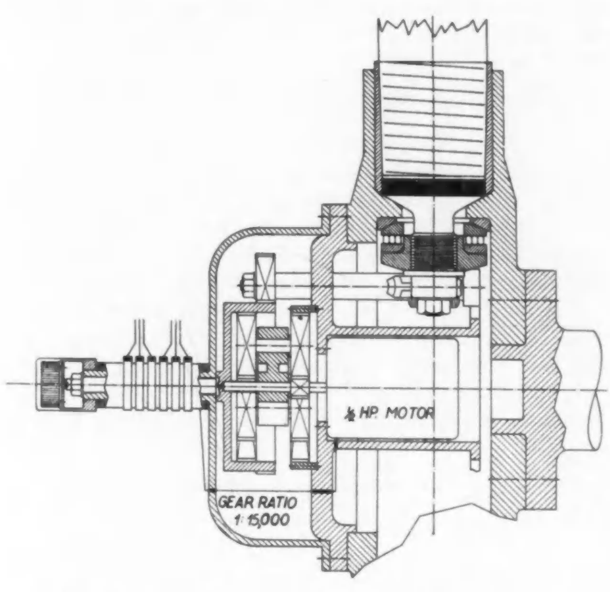
The propeller is located just beyond the second corner, going in the direction of flow and numbering the corners starting from the test section. The diameter at this point is 13 ft and is constant for some distance in front of and behind the propeller. At this point the velocity in the tunnel is reduced sufficiently so that the tip speed of the propeller is well below the speed of sound. The driving motor is mounted on a heavy foundation at this point, which incidentally is the only rigid anchorage for the tunnel. An annular ring surrounds the tunnel in the plane of rotation of the propeller. The space between tunnel and the ring is filled with concrete to insure against vibrations of the tunnel shell with respect to the propeller. With this rigid connection of tunnel and motor, it is possible to use a very small clearance between the blade ends and the shell. This clearance varies between ⅛ in. and ¼ in.

The resistance of the motor is reduced by a streamlined fairing. The rotation of the fan will induce rotation in the airstream, so guide vanes are installed behind the propeller to eliminate this rotary motion. Five guide vanes are used; two conveniently are used to support the motor and serve as conduits for cooling air from outside. Another contains the power and control leads to the motor.

The design of the corners is extremely important, for they cause a large percentage of the total tunnel energy losses, amounting to about 15% of the energy in the approaching section. In the large corners single-plate guide vanes are used. Hollow guide vanes are used in the smaller corners for cooling purposes. In order to obtain smooth flow through the corners, the gap chord ratio of the vanes has to be kept to 0.4. In each corner the vanes terminate in a strong elliptical girder (Fig. 3).

At a right angle intersection of two circular cylinders the cross-section is an ellipse. From a casual consideration it might appear that, in the high-pressure condition all the corner vanes would be carrying tension stresses. Since the tunnel is a closed tube, tension stresses will be produced along the axis of the tubes as well as in the tangential direction parallel to the stiffening rings. At the elliptical intersection these two stresses will have a component which will produce compression in the outer half of the vanes and tension on the inner half of the vanes. The shape of the vanes is such as to make them strong as columns, and the length of the vanes as columns is reduced by horizontal plates between the vanes. The small corners have one plate on the horizontal diameter; the large corners have three.

For a given density the velocity will depend on the rpm of the propeller and the pitch of the blades. A large range of air speeds is desirable, so this would indicate the need for a d-c motor because of the possibility of making small adjustments in the rpm. However, the d-c motor is larger than the a-c motor for the same power, since it is not possible to use high voltages for d-c operation. Furthermore, since the motor was to be inside the tunnel and sealed from the tunnel pressure, the need for commutator and brush service precluded the use of this type of motor.



■ Fig. 5 – Controllable-pitch propeller hub (diagrammatic)

An induction motor of 2000 hp operating at 2200 v is used to drive the propeller. The motor is designed to run at four different speeds by pole changes; namely, 1175, 880, 585 and 440 rpm. The cooling requirements of such a motor are rather difficult, not only because of the large power, but because of the tunnel temperature which may rise to 160 F under some conditions. Cooling air is supplied to the motor through the hollow motor supports by a 40-hp centrifugal blower. A large air filter is used to remove foreign material from the cooling air. Since moisture from the cooling air is likely to condense inside the motor when the motor is not operating, a small heating unit and blower circulate hot air through the motor when it is not in use.

A variable-pitch propeller is essential equipment for a motor which permits only large variations in the rpm. Furthermore, the variable-pitch arrangement is a necessity to utilize the power available for all pressure conditions in the tunnel. As was mentioned previously, the propeller is 13 ft in diameter. This large diameter and the required blade width gives wood a decided advantage over metal as a material for the blades because of the large reduction in centrifugal forces. Six blades are required, so the hub casting must of necessity be large. A 40-in. diameter steel casting provides for the attachment of the six blades and contains the pitch-changing mechanism (Fig. 4).

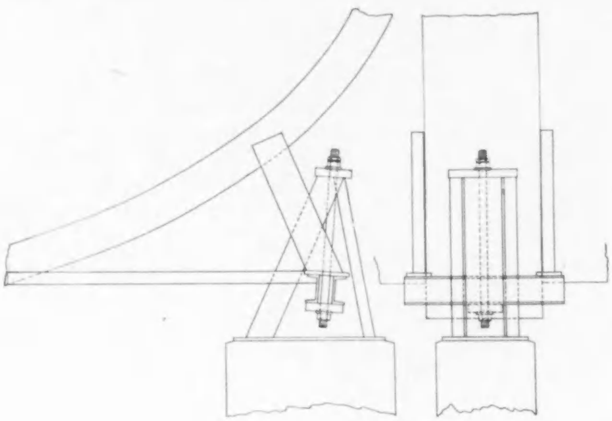
A ½-hp, three-phase motor, 1700 rpm mounted in line with the propeller axis supplies the power for changing the pitch. To obtain the large torque required to turn the blades, a reduction gear is necessary. Fig. 5 shows schematically the method employed. An epicyclic system with two ring gears, one having one tooth more than the other, is mounted on the hub axis. Since the fixed ring gear has one more tooth than the free gear the resultant motion of the free gear will be greatly reduced. The free ring through external teeth transmits its motion to the blade ferrule through a worm and worm gear mounted on the blade shank.

Current is supplied to the motor by slip rings on a shaft at the front of the hub. The pitch changing can then be made from the control panel by a switch which merely changes two phases of the motor. There are extra slip rings for the electrical circuit of a variable potentiometer, the coil of which is attached to the hub and the sliding contact to the slow-moving ring gear. The potentiometer gives an indication of the pitch setting at the control panel.

When the tunnel is operating at maximum speed, the entire output of the motor will be converted into heat. In order to keep the tunnel temperature at a reasonable value, a cooling system must be used. An investigation showed that the equivalent of 2000 hp of heat energy could be withdrawn from the tunnel by a running film of water on the tunnel shell and by water flowing through the guide vanes in the smaller corners. Water is sprayed on the shell by nozzles which are fed by a pipe running along the top of the tunnel. The water film covers the entire area between the stiffening rings and adheres to the tunnel surface almost to the very bottom of the cylinder. Water is sprayed into the top of the guide vanes instead of allowing them to run full. The cooling water is collected in a large concrete apron and drained to the river which is the water source. With this system it is not expected that the temperature of the tunnel can ever exceed 160 F.

Provision must be made for the expansion and contraction of the whole structure. As mentioned previously, the structure is rigidly held at the motor support. At each of the other corners the tunnel is suspended on a long bolt which is anchored at the top of a tripod (Fig. 6). This suspension allows the structure to move in a horizontal plane.

A water seal compressor driven by a 150-hp induction motor is used to increase or to decrease the air pressure in the tunnel. The intake of the compressor can be opened to the outside air or to the tunnel by a system of valves. The volume of the tunnel is approximately 32,000 cu ft, and the time required to compress the tunnel to 4 atmospheres



■ Fig. 6 – Suspension tripod

absolute is about 2½ hr. To exhaust the tunnel to ¼ atmosphere requires somewhat less time.

At present the tunnel is operated at atmospheric pressure only. The need for wind-tunnel tests by the airplane manufacturers has kept the tunnel busy without interruption since the atmospheric calibration was made. At normal density the maximum speed with full rated power is 260 mph, which gives an energy ratio for the tunnel of approximately 3.5. The energy ratio is the ratio of the kinetic energy per second at the throat to the power input to the motor. This value is somewhat lower than might be obtained with the best possible design; but, as mentioned previously, the overall length of the tunnel was shortened arbitrarily to reduce the initial cost.

The variation in velocity distribution across the elliptical test section is less than 1% of the average velocity for 8/10 of the length of each axis of the ellipse. Thus a model of normal size will be in a stream of reasonably uniform velocity. The velocity distribution was adjusted by the use of screens ahead of the nozzle.

The turbulence in the tunnel was measured by means of the usual pressure tests on a 6-in. sphere, with the screens for correcting the velocity distribution in place. The critical Reynolds Number for this sphere was found to be 3.55×10^5 for a sphere pressure difference of 1.22. Compared with the free flight critical Reynolds Number for a sphere of 3.85×10^5, the turbulence factor for this tunnel is 1.08, which is reasonably low.

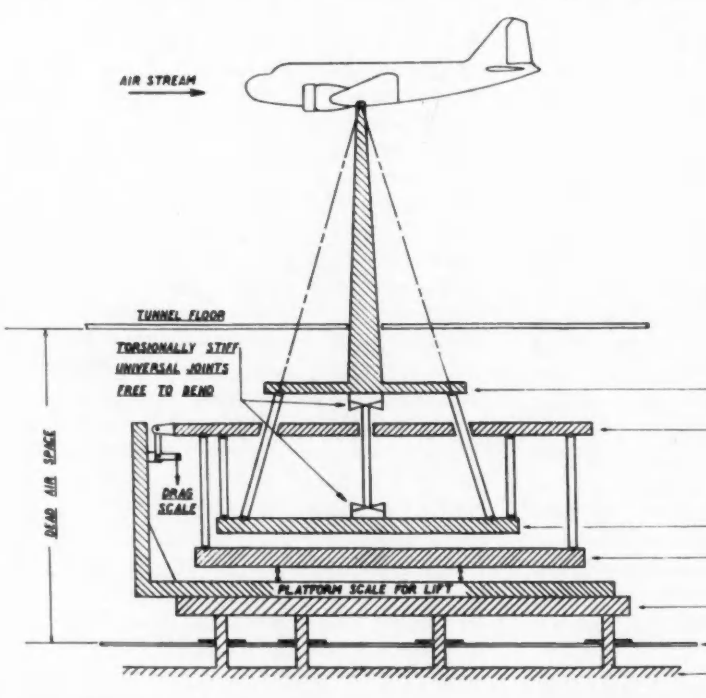
Based on a normal wing chord of 15 in. for a model, the maximum Reynolds Number attainable at atmospheric conditions is three million (3,000,000).

A few measurements have been made with the tunnel at pressures above and below atmospheric pressure. At 3.5 atmospheres, the measured air speed at rated hp of the motor was 145 mph. For an airfoil of normal size for the tunnel the Reynolds Number would be 5.6×10^6. At 0.23 atmospheres a speed of 396 mph was measured at rated power of the motor. The corresponding Reynolds Number is 9×10^5.

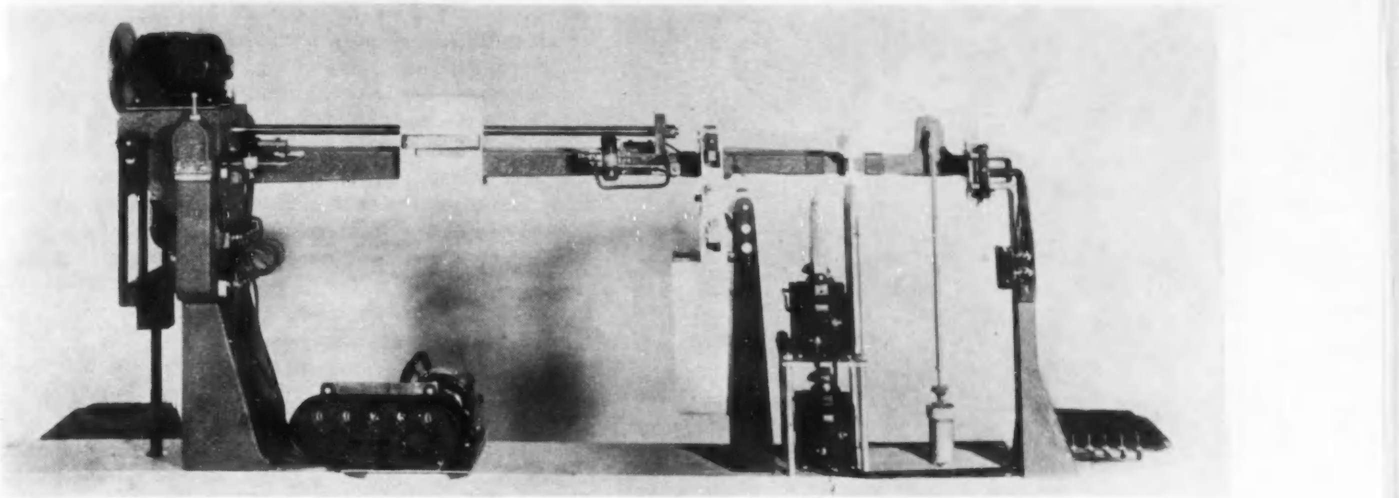
■ The Aerodynamic Balance System

For this wind tunnel a balance was desired with the following general specifications: It must be compact, for the pressure-type tunnel requires that the shell surrounding the test section be kept to a minimum diameter. Measurements of all six components must be made remotely with angular settings of the model in pitch and yaw made from a remote station. It was further required that each component be measured independently and totally so as to simplify and shorten the numerical calculations.

The size of the model and the value of q, the dynamic pressure, determine the magnitude of the forces and mo-



■ Fig. 7 – Scheme of six-component measuring system



■ Fig. 8 – Automatic beam

ments to be measured. A maximum lift force of 3000 lb and drag force of 600 lb were set as the capacities for the lift and drag scales although, under some conditions, the lift and drag might greatly exceed these amounts. Capacities of the balance system for the side force, pitching moment, rolling moment and yawing moment were set at 350 lb, 400 lb-ft, 350 lb-ft and 300 lb-ft, respectively.

Fig. 7 shows diagrammatically the general scheme of the six-component measuring system. The model is held in the normal flying position and is supported on the balance by three vertical struts.

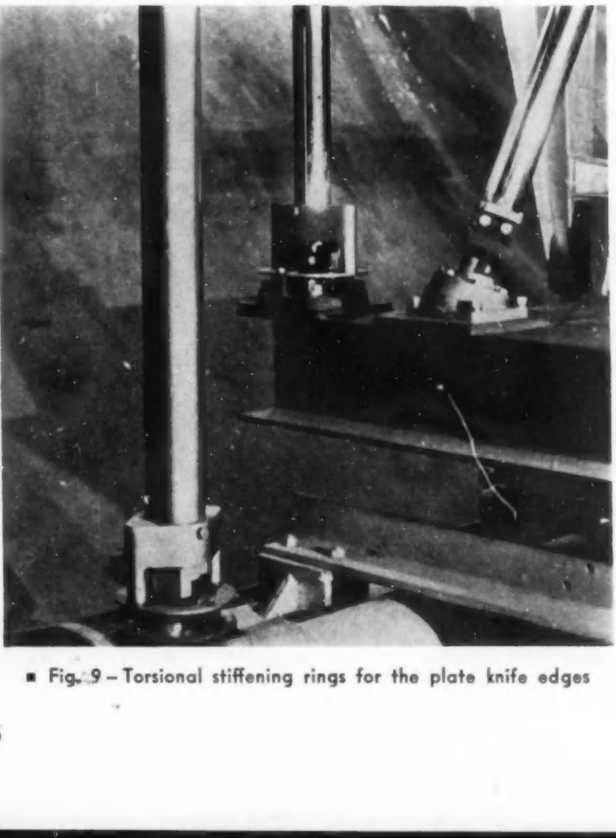
The three forces are measured by a system of three "tables" with freedom in lift, drag, and side-force direction. To obtain a moment axis at the center of the tunnel, a four-strut truncated pyramid with apex at the center is located between the upper "table" and the model supports. A similar scheme utilizing a truncated pyramid to obtain an axis center in the model above the balance had been used earlier in the balance of the 8 x 12 ft wind tunnel at the University of Washington.

The model and its supports with all the devices and members for measuring the remaining components are mounted on a platform scale which measures the lift. The lift scale uses thin steel plates for knife edges. The motion of the lift scale is limited so that it can move only vertically. This is accomplished by means of check rods and plates arranged to prevent deflections of the plate knife edges which would be produced by the side forces or yawing moments on the model which ultimately must pass through the lift scale to the balance foundation. Through a lever system the lift force is measured on a beam which operates automatically. Fig. 8 shows the operating features of the beam.

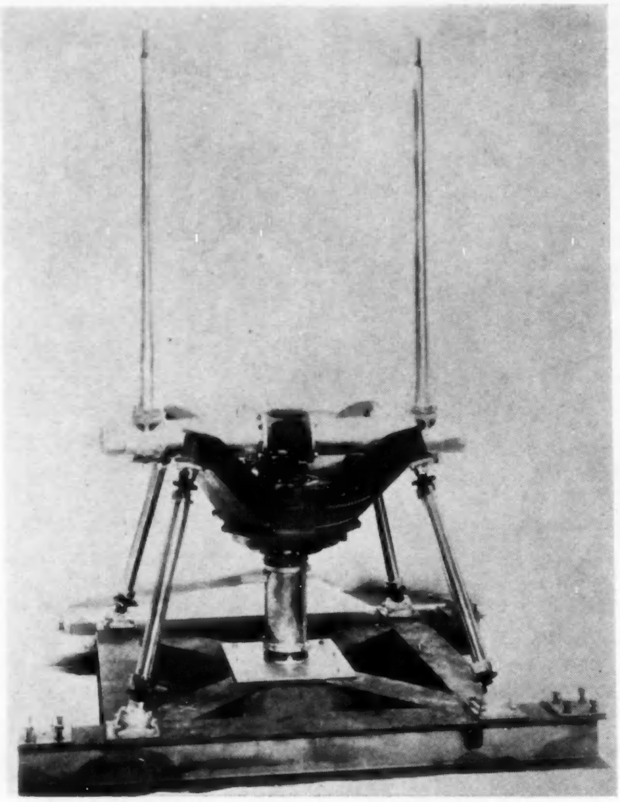
An electric motor, the direction of rotation of which is controlled by contacts at the end of the beam, moves a weight by means of a threaded rod along the beam. The total movement of this weight corresponds to 500 lb but, by the addition of four poises, which can be added or removed from outside the tunnel, the range of the lift scale can be increased to 3000 lb. A Selsyn motor system geared to the driving rod operates a counter which is located on the observation panel. Lights on the observation board show which poises are in action and what forces must be added to the counter reading to give the total lift. The lift scale carries the weight of the entire balance system as a tare load. This condition might appear to impair the accuracy of the system at small values of the lift. The measured force, however, is dependable to 1 lb throughout the range.

The drag is measured by a large rectangular frame supported at its corners by tubes attached to the lift scale. At the ends of each tube are plate knife edges normal to the wind direction. At the downstream end of the drag frame is a large bell-crank which connects the drag frame to the drag measuring system. An automatic beam similar to the lift beam measures the force. Increments of 0.1 lb may be measured throughout the range.

It is essential that the drag system be free in the wind direction but torsionally stiff to resist deflections due to yawing moments and side forces. Fig. 9 shows a device



■ Fig. 9 – Torsional stiffening rings for the plate knife edges



■ Fig. 10 – Moment separating system

for stiffening, torsionally, the drag legs while still allowing displacement in the drag direction. The movement of the automatic beams for both the lift and drag force measurements is very small. The contacts at the end of the beams have a clearance of approximately 0.01 in. which, considering the overall lever ratio of each system, provides in effect a nul system for each force. Changes in angle of attack and angle of yaw of the model do not affect the wind-off readings for either force; hence the forces are read directly on the panel board.

Suspended from the drag frame by four tubes is a third frame. At the ends of each of these tubes are plate knife edges located so that they can deflect in a direction across the wind stream. The motion of this frame with respect to the drag frame is restrained by a calibrated spring. The motion of this spring is measured by a strain gage which indicates the cross wind force in pounds at the instrument panel. The tubes connecting the drag and cross wind frames are stiffened torsionally in the same way as the legs supporting the drag frame.

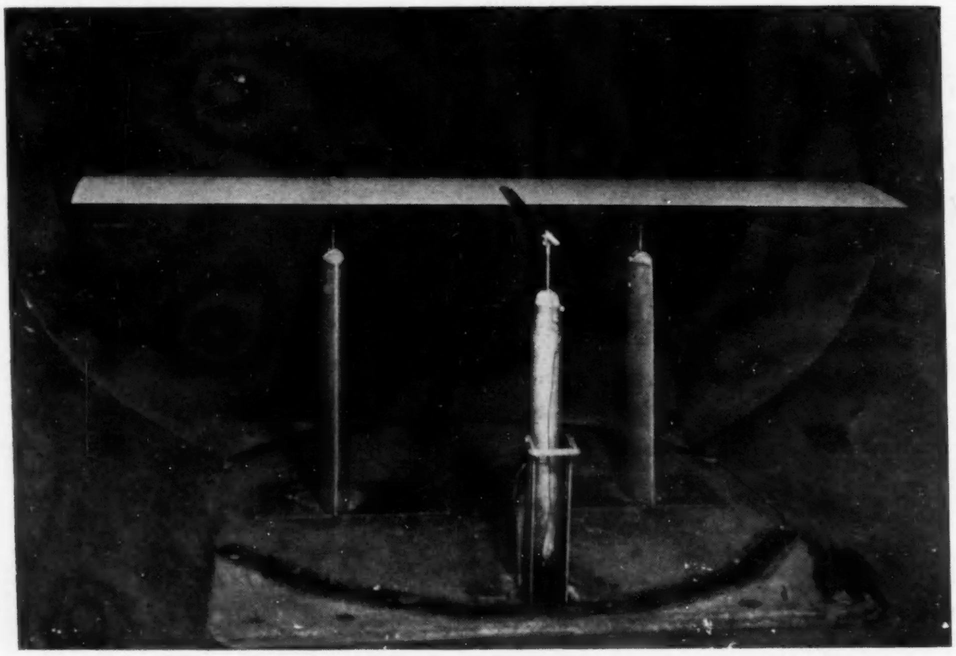
■ Measuring Side Force

In this method of side-force measurement a component of the lift will be included, and it may be a large portion of the measurement if the sidewise deflection of the frame is large. The side-force restraining spring limits the angular displacement of the tubes to 0.2 deg for the maximum measured force. This small displacement does not affect the accuracy of the side force measurement by more than 3% in the most unfavorable condition.

Mounted on the side-force frame are four tubes sloping inward and upward with pairs of plate knife edges at right angles near their ends (Fig. 10). The upper ends are attached to the arms of a large steel casting in which rotates a large hub. Bolted to this hub are large tubes to which the struts that support the model are attached. The attachment to the model is made at a height at which the sloping legs, if prolonged, would intersect.

A force acting through this intersection would produce only changes in tension and compression in the sloping legs. A force on the model not acting through this intersection will produce moments about this point and consequently bending in the legs.

A stiff rod is located on the axis of the pyramidal structure and attached to automatic beams through universal joints. One beam is located in the direction of the wind,



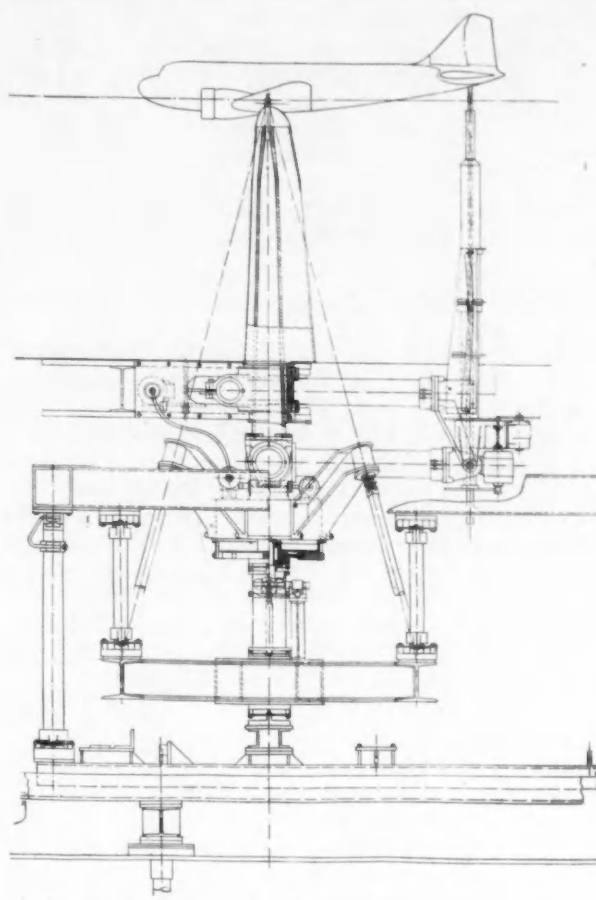
■ Fig. 11 – Model mounted in tunnel

the other at right angles to the wind. These beams apply moments which offset the tendency of the pyramidal structure to bend. Thus they, by proper design, can be made to measure directly the pitching and rolling moments of the model about the point of support.

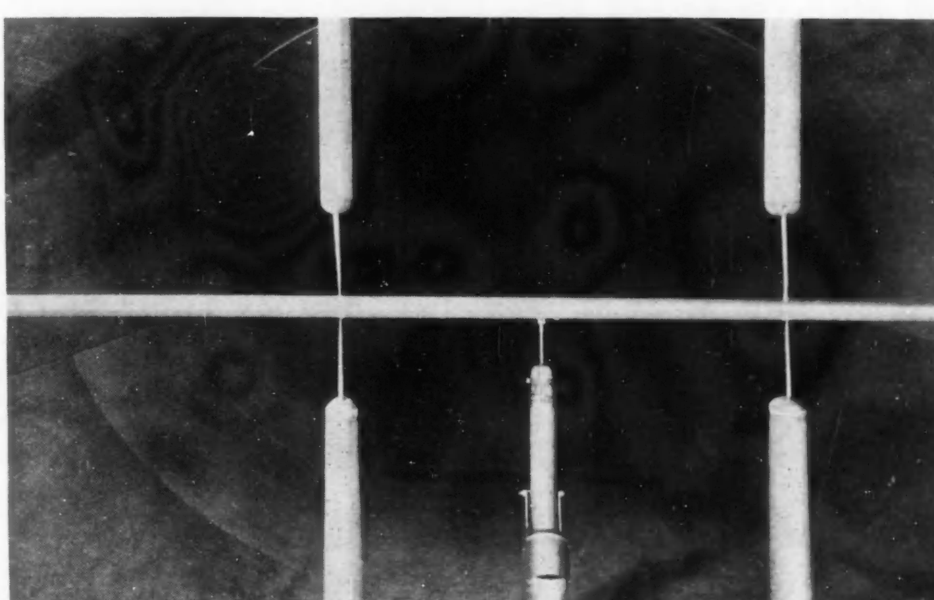
The tendency of the pyramid to twist is counteracted by a torque tube located on the axis of the pyramid between the base of the pyramid and the large steel casting. This tube is made weak in bending to permit motion of the pyramid in the pitching and yawing plane, thus not interfering with the measurements of these two components. Between the torque tube and the large steel casting, springs are located which oppose the twisting moment transmitted to the pyramid by a yawing moment on the model. An electrical strain gage is used to measure the deflection of this system, and a meter is calibrated to read the yawing moment of the model directly in foot-pounds. The deflections in yaw are kept extremely small by using stiff springs, and no measurable error in the yawing moment due to lift forces has been observed.

■ Remote Control of Attitude

Wind shields for the model supports must be used (Fig. 11). Otherwise, large tare corrections would have to be applied to all the measured components. The attitude of the model can be varied in pitch and yaw from outside of the tunnel with angular settings indicated by Selsyn motors at the observation panel. The wind shields for the support struts move in yaw and pitch in synchronism with the struts but must not touch any part of the balance proper. Supports for the shields are therefore independent of the measuring system. Synchronism in yaw is obtained by connecting the operating motors for the shields and model by a flexible coupling. The fairings for the front model supports are streamlined and rotate in their individual supports as well as turn with the model. Alignment with the wind is obtained by means of an external parallel linkage. The pitch-changing device and its shield are shown in Fig. 12. A single motor moves the model and shield. However, the toothed clutch, by means of which the fairing and the rear model support are synchronized, is engaged only during the actual process of moving. The shield must rise and descend with the rear support and must also move backward and forward with



■ Fig. 12 – General assembly of balance (side view)



■ Fig. 13 – Airfoil mounted in tunnel for tare and interference measurements

it so as always to surround but not touch it. A screw and cam arrangement accomplishes this result (Fig. 12).

The measuring system for each component is damped so that reasonably steady readings of the forces and moments can be made. Alignment of the balance to the wind direction is easily accomplished by moving the drag frame forward or backward until the supporting legs are perpendicular to the wind direction. In this attitude there is no component of the lift in the drag system. The small component of the drag in the lift system is negligible.

■ Tare, Interference, and Alignment Tests

A tare and interference test along with a balance alignment test is usually made for each model at the start of its test program. The model is mounted on the balance inverted and a polar run at the speed or speeds at which tests are to be made. Images of the front fairings are then lowered from the top of the tunnel and dummy front and rear supports fastened to the model. The front dummy supports extend into but do not touch the front image fairings. The dummy rear support is equal in length to the exposed part of the rear balance support, but the rear fairing is not represented as an image. The inclusion of the rear fairing presents great practical difficulties, but the aerodynamic interference of the rear fairing on the model and on the rear support is believed to be negligibly small (Fig. 13).

Polars are then run at the same speeds as without the image fairings and dummies. At a given angle of attack and speed the difference in forces between the tests with and the tests without dummy fairings and supports are the tare and interference corrections.

The alignment of wind stream to balance is found by turning the model to the right-side-up position and leaving all the dummy fairings and supports in the same location with respect to the tunnel. Polars are then run at speeds corresponding to the previous polars. The alignment is determined from plots of the polars with dummy fairings and supports in place and with model upright and inverted in the tunnel.

It is to be noted that the balance alignment is determined with the dummy fairings and supports in the air stream and that in testing they are removed from the air stream. Any change in average wind-stream direction due to removal of the dummies is included in the tare and interference corrections and is canceled out when these corrections are applied.

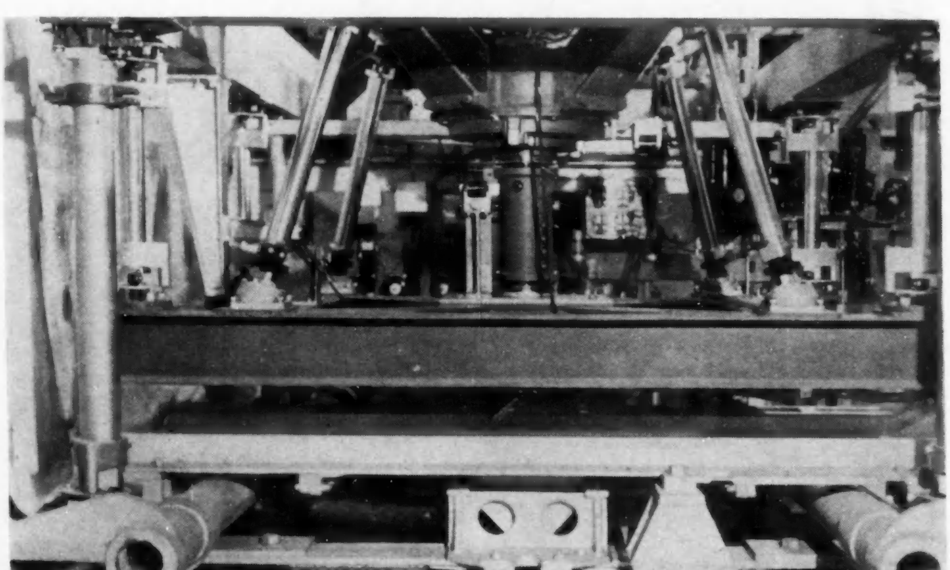
For usual model wings the drag tare and interference corrections vary nearly linearly from about 30% of the net drag of the wing alone at zero lift to about zero at a lift coefficient of 0.8.

On a model with large motor nacelles, tare and interference tests are usually run for the wing without nacelles and for the wing with nacelles. In some cases when the nacelles were near the front support points, the presence of the nacelles modified the drag tare and interference corrections appreciably.

With this wind tunnel and balance system it is possible to obtain measurements very rapidly on a complete airplane model from which the coefficients for all six components can be plotted with a minimum of calculation. For example, the polar diagram for the lift, drag and pitching moment coefficients for a model wing can be obtained in about 25 min after the model is mounted. It is usually possible to mount the model so that the point representing the center of gravity of the airplane is on the balance axis, a feature which increases the accuracy of the results and the simplicity of the calculations. Principal balance members are shown in Fig. 14.

This wind tunnel was made possible by a fund voted by the Corporation of the Massachusetts Institute of Technology and the gift of a substantially equal amount by interested individuals. The tunnel was named for the Wright brothers to commemorate and perpetuate the methods of research and controlled experiment used in the development of the first airplane.

Recently special equipment for the testing of powered airplane models has been donated by the Curtiss-Wright Corp. and the United Aircraft Corp. This includes a power supply of 100-hp output with frequency variation up to 400 cycles, dynamometers for calibration of motors installed in models, and the necessary instrumentation and control equipment.



■ Fig. 14 – Principal balance members

American Experience with BUCHI TURBO-CHARGING

by J. P. STEWART
Elliott Co.

RALPH BOYER
Cooper-Bessemer Corp.

JOHN W. ANDERSON
American Locomotive Co.

THIS paper reviews briefly the background of the adoption of the Buchi turbo-charging system in the United States, particularly with reference to the American Locomotive Co. and the Cooper-Bessemer Corp. who were most active in sponsoring it. Broad design considerations are discussed, concerning the requirements for best results with the use of this type of supercharger. Performance curves covering several makes of engines are included, and special characteristics of the turbo-charged engine are discussed. The paper also contains a brief description of turbo-charger units of American manufacture.

In remarks concerning the probable trend of future supercharging developments, the authors predict that the four-cycle supercharged diesel engine is on the verge of rapid evolution toward a substantial reduction in specific weight and space, to be accompanied by considerable research and field operating experience.

Tests reported show that a turbo-charged diesel engine will carry suddenly applied full load in the same manner as will the unturbo-supercharged engine. No additional maintenance difficulties or costs are encountered with turbo-charged engines, the authors contend.

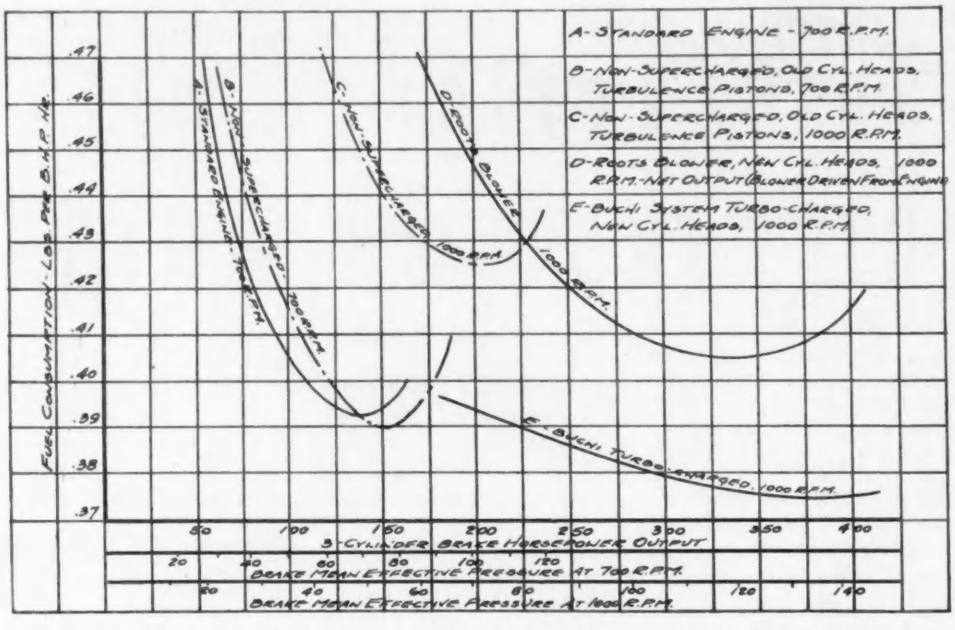
THE Buchi system of supercharging four-cycle diesel engines now enjoys wide acceptance, and is in common use today throughout the world. Excluding installations in the United States, the latest available figures show a total in excess of one million horsepower installed with this type of supercharging system, with individual engines rated from approximately 150 hp to 6000 hp. The inventor, Dr. Alfred J. Buchi of Winterthur, Switzerland, is well known, either personally or by reputation, to most of the American diesel engine industry. Also, a great deal of information concerning the subject has been published during the past several years, in both European and American technical publications. For these reasons, the authors will not attempt to deal with the fundamental considerations of the Buchi system, but rather will endeavor to present some factual information concerning the application of the Buchi system to engines manufactured in this country.

The first tests of exhaust-gas-turbine-driven superchargers were conducted by the inventor in Switzerland in 1911. It is interesting to note that these early tests embraced charging pressures up to approximately 30 psi gage, combustion pressures exceeding 1400 psi and brake mean effective pressures over 200 psi. The original tests employed a constant-pressure system at the engine exhaust to serve the turbine and, as might be expected with such an advanced idea of that day, a long period ensued before the development reached a state of practicability and general acceptance. This evolution is now a matter of history, but the most important single factor involved discarding the constant-turbine-pressure idea for the present method of creating timed pressure pulsations in the exhaust pipes. This provided a means of efficient scavenging with low air pressures, and made possible increases in output with lower charging pressures than would be possible otherwise. The development was accompanied also by numerous refinements in details of engine design, as well as centrifugal blower and high-temperature turbine design. Although the majority of the European and English installations have been made in the past ten years, unquestionably the Buchi system is now commonly accepted abroad and considered out of the experimental phase.

American Development Gaining

For a number of reasons, the American diesel engine industry lagged somewhat behind the European developments, although from present indications it is now making up for lost time. So far as is known, Cooper-Bessemer was the first actually to apply the Buchi system in this country. They imported a Brown-Boveri turbo-charger unit in 1934 and applied it to their 8-cyl, 8 x 10½-in. engine. The tests on this engine extended through a year or more, during which time they developed some basic information on the problem of exhaust manifolds for 8-cyl engines. In the meantime, they have made a number of successful installations for marine and locomotive service.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 5, 1941.]



■ Fig. 1 – Performance developments – American Locomotive Co., 9½ x 10½-in. diesel engine

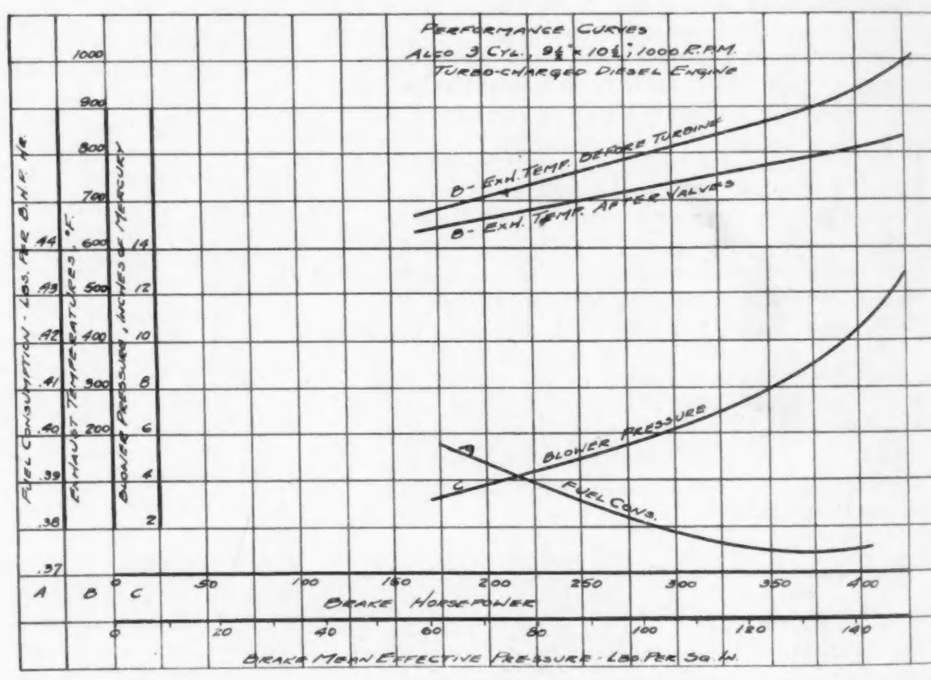
American Locomotive imported Brown-Boveri turbo-chargers until early 1940, at which time it began production of a turbo-charger of its own manufacture.

In January, 1940, the Elliott Co. consummated a license agreement with Dr. Buchi, permitting the manufacture and sale of turbo-chargers in the United States and its possessions to licensed diesel engine manufacturers, for marine, locomotive, and stationary applications involving supercharged ratings of 250 hp and upward. The following year was spent in developing, testing, and perfecting an advanced design, and in completing plans for quantity production of these units. At this time, production shipments are being made regularly, and Elliott Co. has on order from 8 engine manufacturers, for delivery within the next 12 to 18 months, turbo-chargers to serve approximately 245,000 engine hp.

In applying the Buchi supercharging system successfully, a number of design considerations must be observed in order to accomplish the best results. It is not quite so simple as merely attaching a turbo-charger to the engine and piping up the exhaust gas and inlet air connections. Dr. Buchi, in his paper presented to the ASME in December 1936,[1] discussed the basic principles of this system in considerable detail.

At the present writing, Cooper-Bessemer has on order for delivery within the next year, engines totaling approximately 70,000 hp, employing the Buchi system.

The American Locomotive Co. completed its first engine for use with the Buchi system in October, 1936, and shortly after that time this engine went into service in a switching locomotive. It performed so satisfactorily that, beginning in February, 1937, the company began to produce such engines in regular production quantities. Since then, this company has built approximately 150 engines turbo-charged according to the Buchi system, with a total rated power of approximately 140,000 hp. The performance of these engines has been so generally satisfactory in service that American Locomotive is now building a larger proportion of its engines turbo-charged today than it ever has formerly, and now has on order for delivery within the next year or so far more turbo-charged engines than it had built in the preceding four and one-half years. The applications of these engines cover railroad switching service, railroad locomotives for passenger and freight service, application to generator sets both for general power purposes and for electric propulsion drive, in addition to directly reversing propulsion units.



■ Fig. 2 – Performance curves – American Locomotive Co., 9½ x 10½-in., 1000 rpm, 3-cyl, turbo-charged diesel engine

[1] See ASME Transactions, Vol. 59, February, 1937, pp. 85-96: "Supercharging of Internal-Combustion Engines with Blowers Driven by Exhaust-Gas Turbines," by Alfred J. Buchi.

Fortunately, these design considerations are well established from previous experience, and are available to the prospective user of the Buchi system. Among the most important design considerations concerning the engine itself are the following:

1. Proper valve timing to permit adequate scavenging. Scavenging is just as important, or perhaps more so, than the actual charging of the cylinder under pressure before compression. Proper valve timing is necessary to permit the air to flow through in spite of the fact that the peak exhaust pressure is higher than the air charging pressure. Timing must be such that the exhaust pressure has dropped sufficiently by the time the air intake valve is opened, to permit the air to flow through with the charging pressure available.

2. Care must be taken that sufficient area is provided in the cylinder-head passages, valves, and combustion space so that the air pressure drop is not excessive during the scavenging period when the piston is at the upper end of its stroke, and the inlet and exhaust valves are open. In this connection, a four-valve engine is more adaptable to turbo-charging than a two-valve engine, since a greater flow area may be provided for charging and scavenging air.

3. The larger quantity of air charged into the cylinders requires careful attention to the combustion problem, in order that the increased quantity of fuel may be burned properly. In general, the open-type combustion chamber with central injection has given the best results with the Buchi system.

4. The design of the exhaust manifolds must be given careful attention with respect to size and combinations with cylinders in each specific case, in order that the pressure pulsations will be of the proper magnitude and will not cause interference with pulsations from other cylinders, or resonant oscillations. It is important that the exhaust manifolds be as straight and short as possible, in order that the energy in the exhaust gas be utilized most efficiently. Adequate expansion joints must be provided, and the exhaust manifolds insulated. In some cases, each individual pipe is insulated, although many of the latest designs employ a single insulated housing surrounding the entire assembly of exhaust pipes.

5. The most desirable manifolding arrangement can be made on an engine which has the air manifold on the side opposite the exhaust manifolds, although satisfactory arrangements have been worked out on numerous engines with the intake and exhaust on the same side.

The foregoing considerations are not intended to infer that engines of existing design cannot be turbo-charged successfully. In fact, a great variety of sizes and types of existing engines have been and can be turbo-charged successfully with relatively few internal design changes, and a power increase of at least 40 to 50% provided over the normal continuous unsupercharged rating. However, further progress in increasing the specific output of supercharged engines beyond present practice will require closer attention to such details. The biggest opportunities will come to those manufacturers developing new engine designs specifically for turbo-charging, so that the turbo-charger and its connected system are an integral part of the complete unit. By this means, even higher percentage increases in output are practical.

The experience of the American Locomotive Co. in this respect has been most enlightening. Fig. 1 shows the progressive steps of development of their 9½ x 10½-in. engine. The original unsupercharged fuel consumption is shown on Curve A, in which case the 3-cyl engine was normally rated 150 hp at 700 rpm. The second step in the development was the installation of a special turbulence combustion system, and the non-supercharged fuel consumption at 700 rpm is shown by Curve B.

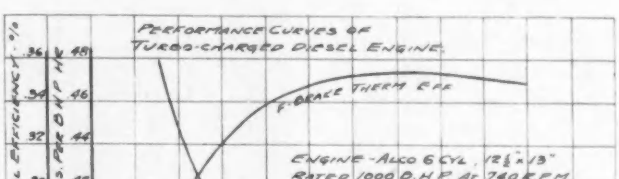
Curve C is on the basis of this same improved combustion system, but at a speed of 1000 rpm. The fuel consumption is relatively high, due primarily to the low volumetric efficiency of the engine at the increased speed, caused by insufficient area in the cylinder-head gas passages.

The next step in the development was to install new cylinder heads with specially designed induction passages, and to supercharge the engine by means of a Roots blower. The performance on this basis is shown by Curve D, with the blower power supplied by the engine.

Curve E shows the latest test results with the use of the Buchi system. Note the flatness of the curve, the excellent fuel consumption, and the fact that the bmep is 142 psi at 400 bhp.

Fig. 2 shows complete test data on this same engine when using a turbo-charger with the Buchi system.

Fig. 3 shows complete shop test performance of a regular production engine rated 1000 bhp at 740 rpm. The test was carried to 10% overload, with a maximum bmep of 122 psi, at rated speed. Note particularly the very flat fuel consumption Curve A, as measured in lb per bhp-hr. From 60% load to 110% load there was a variation of about 1½%, and at 25% load the fuel consumption is still only 0.48 lb per bhp-hr. Curve F shows the very flat brake

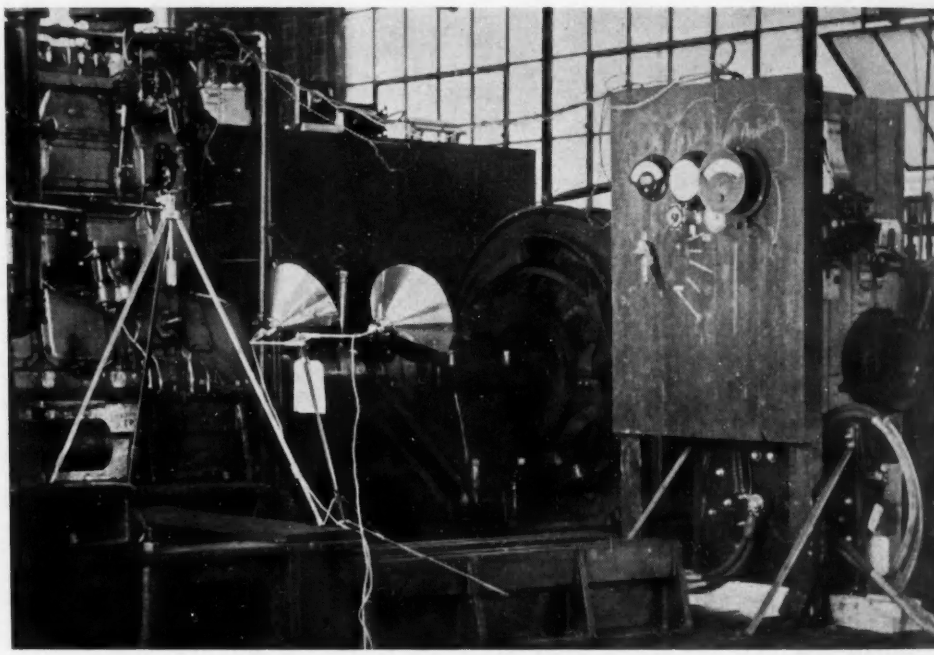


■ **Fig. 3 – Performance curves – turbo-charged production diesel engine**

thermal efficiency corresponding to the flat fuel consumption curve. Curves B and C show a very favorable ratio of charging pressure to average exhaust manifold back pressure. Curve D shows the temperature of gas at exhaust valves to be only 750 F at 10% overload, while Curve E shows the gases entering the turbo-charger at the same load are only 950 F.

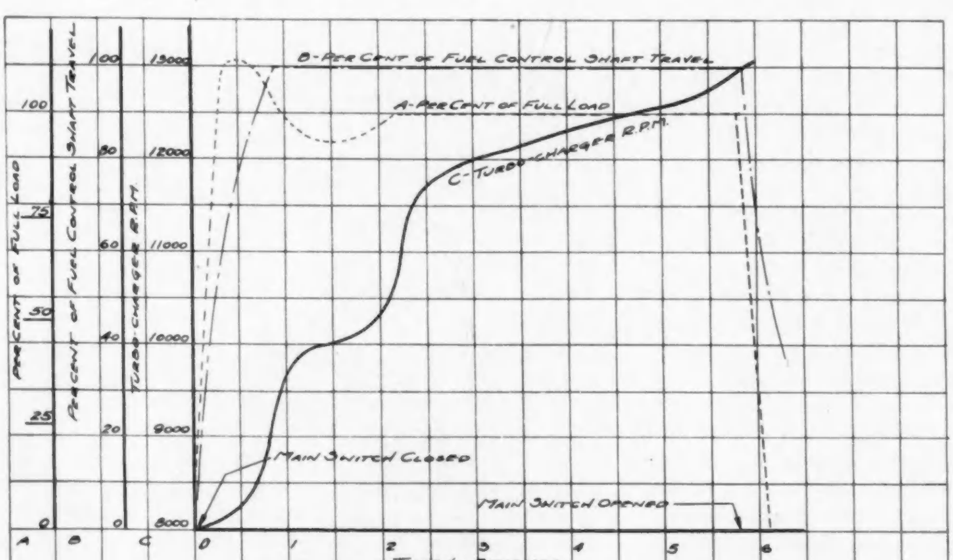
In April, 1941, a series of extensive tests was conducted by Cooper-Bessemer in its Mt. Vernon, Ohio Plant on its Type KN8, 8-cyl, 8 x 10½-in. engine, using an Elliott-Buchi turbo-charger. The object of these tests was to determine several characteristics of turbo-charged engines, concerning which little information was available in this country.

Probably the most interesting single test made was that of determining the time interval necessary for the turbo-charged engine to reach full load-full speed from a no load-full speed condition, with the sudden application of load by means of a direct-current generator. In order to be sure of accurate results, all of the necessary instruments were mounted on a board and photographed by means of a motion-picture camera. These instruments included an electric clock, voltmeter, ammeter, engine tachometer, blower tachometer, and a mechanical indicator attached to the engine governor to indicate the position of the fuel control shaft. In addition, a hole was cut through the board, and on the back side a light was installed and connected to the generator circuit. This light indicated the exact time at which the circuit breaker was thrown in. Generator loading was provided by means of a water rheostat, and load application and rejection were accomplished by a solenoid-controlled air circuit breaker. The set-up for these tests is shown in the photograph, Fig. 4.



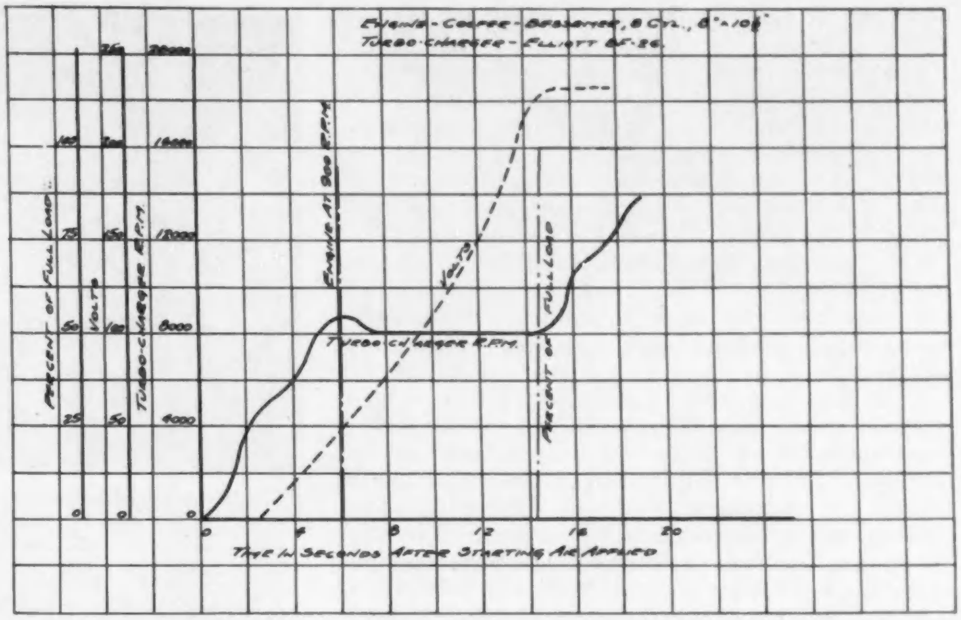
■ Fig. 4 – Set-up for sudden loading test of Cooper-Bessemer diesel engine

Fig. 5 shows the results of this test plotted from the movie film, which had an exposure rate of 64 frames per sec. Curve A shows the generator loading in percentage of rated engine load. Curve B shows the percentage of travel of the fuel control shaft. Curve C shows the turbo-charger speed in rpm. It will be noted that full engine load is reached in approximately 0.25 sec from the time the circuit breaker is closed. However, the blower requires several additional seconds to attain full speed. The explanation of the dip in the blower speed curve is believed to be that the engine speed dipped slightly as soon as the inertia of the rotating masses was overcome. This is further substantiated by the fact that, at 0.25 sec, at which time the ammeter showed full loading, the fuel control shaft was at 50% of full travel, showing that the rotating inertia forces in the engine absorbed the load during the early stages of the loading cycle. Observation of the exhaust showed an increase in exhaust density covering a period of about 2 sec



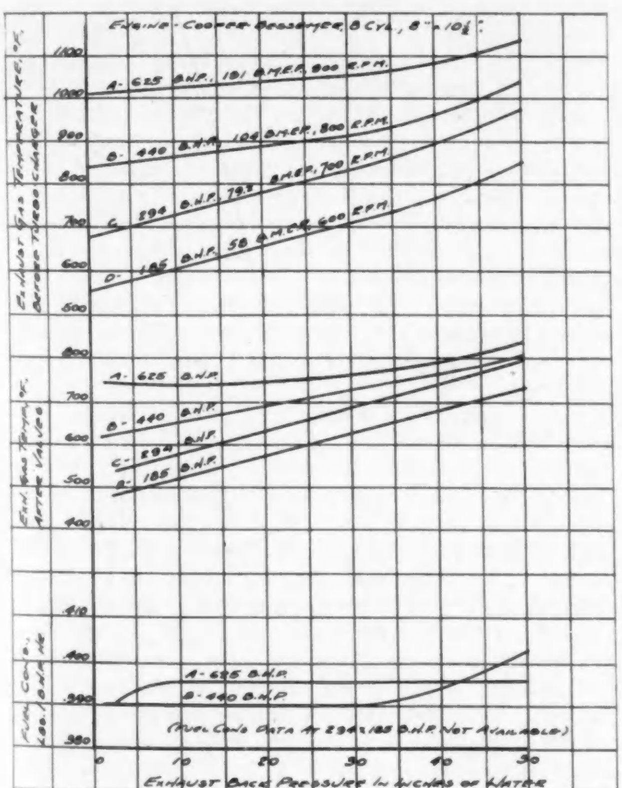
■ Fig. 5 – Effect of full load suddenly applied to a turbo-charged diesel engine – Cooper-Bessemer, 8-cyl, 8 x 10½-in. engine; turbo-charger, Elliott BF-26

which coincides with the change of rate of acceleration of the blower occurring at 10,000 to 11,000 rpm. It would therefore appear that the turbo-charger begins to show its effect on engine performance in about 2 sec after the load is applied. These tests proved conclusively that the turbo-charged engine would carry suddenly applied full load in the same manner as an unturbo-charged engine, and that the size and weight of the flywheel could be decreased materially over an unsupercharged engine for sudden loading requirements.

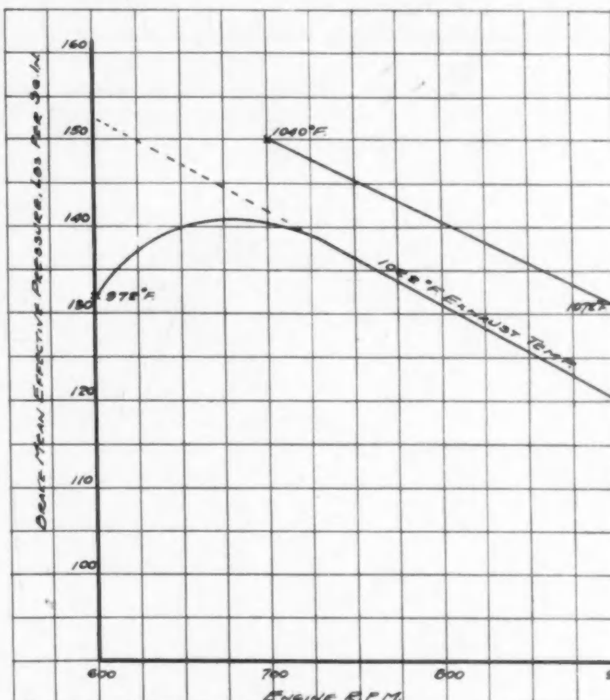


■ Fig. 6 – Turbo-charged diesel-engine loading characteristics, starting from full shutdown – Cooper-Bessemer 8-cyl, 8 x 10½-in. engine; turbo-charger, Elliott BF-26

Fig. 6 shows results of a test made with the same set-up, starting from complete shutdown to full load-full speed. The blower acceleration started when starting air was admitted to the engine, and the engine began to fire after 2 sec, when the blower speed had reached 4000 rpm. The engine attained full speed in 6 sec, at which time the blower had reached full idling speed. However, it will be noted that 14.7 sec elapsed from the time air was applied to the engine until full load was developed. This lag is explained by the curve of generator field voltage, showing that due to this lag it was impossible to apply load before this voltage had built up to full value. If a separate source of excitation had been available, the engine would have been capable of starting and carrying full load in approximately 6.5 sec,



■ Fig. 7 – Effect of back pressure on turbo-charged diesel engine under propeller law loading – engine, Cooper-Bessemer, 8-cyl, 8 x 10½-in.



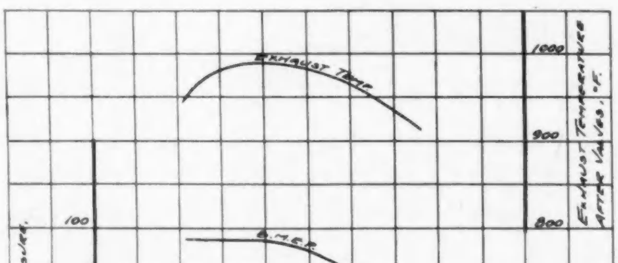
■ Fig. 8 – Maximum power developed by turbo-charged diesel engine at reduced speeds – engine, Cooper-Bessemer, 8-cyl, 8 x 10½ in.; turbo-charger, Elliott BF-26

including the 6 sec required for the engine to reach full speed.

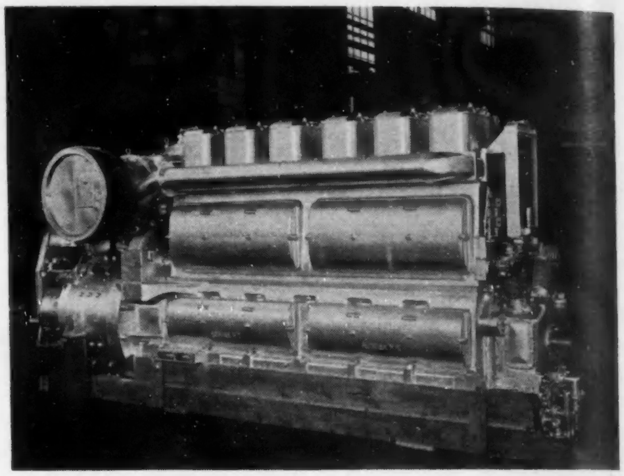
Another series of tests, shown in Fig. 7, covered operation of the engine under propeller law loading at various back pressures up to 50 in. H_2O gage, at the turbine exhaust. Probably the most significant curve is the top one of each set for maximum engine rating, illustrating the effect of back pressure on temperature at turbine inlet, temperature at exhaust valves, and fuel consumption.

Fig. 8 represents an attempt to determine the relationship between engine speed and maximum allowable bmep for the turbo-charged engine, without exceeding predetermined temperature limits before the turbine. The tendency for the bmep of the turbo-charged engine to increase directly as the engine speed decreases, with a constant temperature before turbine, is considered an important characteristic for traction applications and, under certain conditions, for marine propulsion. The reduction in bmep below approximately 650 rpm was due to limitations imposed by the particular fuel system used on this test set-up. The same limitations developed in attempts to establish the limiting bmep at these speeds when operating with an exhaust temperature of 1100 F before the turbo-charger, resulting in the points of 1072 F at 900 rpm, 1060 F at 800 rpm, and 1040 F at 700 rpm.

Fig. 9 shows performance of the engine with the turbo-charger rotor removed, but with the turbine nozzle ring in place. It will be noted that approximately 85 psi bmep was carried at 900 rpm and that this figure increased at lower speeds. Exhaust temperatures were higher than encountered on the standard unturbo-charged engine, and the exhaust color was somewhat darker. This difference in combustion qualities is mainly due to the difference between the valve timing used and that for the unsupercharged engine. It is felt further that slightly more output could have been obtained if the turbine nozzle ring had been removed, since it has a considerably smaller area than



■ Fig. 9 – Maximum power of turbo-charged diesel engine with turbo-charger rotor removed – engine, Cooper-Bessemer, 8-cyl, 8 x 10½ in.



■ Fig. 10 – American Locomotive Co. turbo-charged diesel engine

Table 1 – Heat Balance – Cooper-Bessemer 8-Cyl Model KN8 Diesel Engine, 8-In. Bore, 10½-In. Stroke

	Turbo-charged	Unsuper-charged
Bhp	625	400
RPM	900	900
BMEP, psi	131	83.5
Heat Input to Engine	100%	100%
Heat to Useful Work	32.45%	33.1%
Heat Loss to Cooling Water	17.92%	28.2%
Heat Loss to Oil	3.93%	2.44%
Heat Lost to Exhaust	30.00%	24.5%
Heat to Radiation and Unaccounted for	15.70%	11.76%

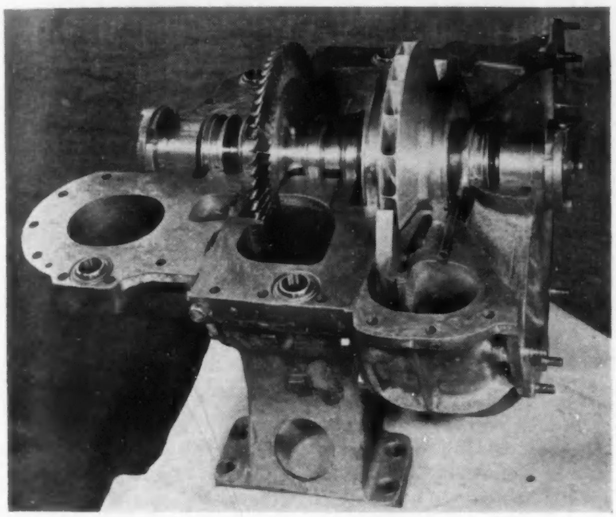
the exhaust passages and, hence, affects the back pressure. A further improvement also could have been made by providing removable covers on the air manifold, which would have removed the restriction to air flow through the blower, thus increasing the volumetric efficiency of the engine. However, this test does show that approximately the unsupercharged rating can be carried on a turbo-charged engine, under emergency conditions, if the turbo-charger becomes inoperative.

Table 1 shows a comparison between the heat balance of the Cooper-Bessemer KN8 turbo-charged engine used in these tests, and the same standard engine without supercharging.

Table 2 contains test data concerning a Union 6-cyl, 16 x 20½-in. engine, on the basis of standard unsupercharged design and turbo-charged by the Buchi system. This engine was equipped with a Brown-Boveri turbo-charger and has been in operation for more than a year. Both engines are equipped with the common rail fuel system. The only change in fuel system for the turbo-charged engine was to increase the fuel pump and nozzle capacity. Other engine changes consisted of new valve timing and exhaust manifolding. About the only outward difference manifested in the operation of the two engines was that the turbo-charged engine appeared to run smoother, and this improvement was attributed to the smoother combustion shown on "pull" cards. The heat removed by the jacket cooling water was substantially the same as for the unsupercharged engine.

One question of possible interest is that of operating maintenance cost on turbo-charged engines as compared with unsupercharged engines. It has not been possible to

Table 2 – Test Record – Union 6-Cyl Marine Diesel Engine 16-In. Bore, 20½-In. Stroke	Turbo-charged	Unsuper-charged
Date	1939	1937
Manufacturer's Engine No.	39506	37518
Bhp	840	600
RPM	240	240
BMEP, psi	111	80
Average Temperature at Exhaust Valves, F	720	690
Temperature Before Turbine, F	930	...
Air Manifold Pressure, psi	4.5	Atmospheric
Turbine RPM	12,400	...
Fuel Pressure, psi	3,800	3,500
Lubricating Oil Pressure, psi	15	15
Cooling Water Temperature to Engine, F	95	95
Cooling Water Temperature from Engine F	117	120

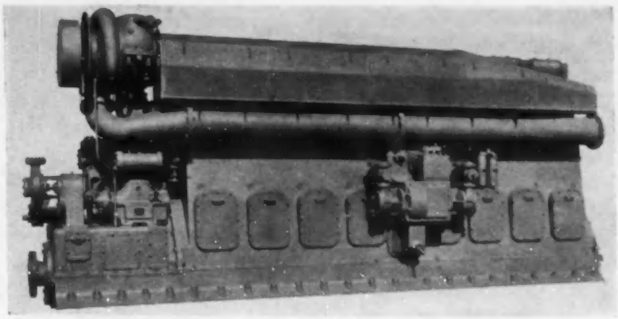


■ Fig. 11 – Lower half of Alco-Buchi turbo-charger

secure any specific figures up to this time to provide an accurate comparison. However, the experience of American Locomotive Co. has been that no additional maintenance difficulties or costs are involved with Buchi turbo-charged engines rated up to approximately 50% above the unsupercharged output of the same engine. In fact, the turbo-charged engine is found to be in better condition during the routine maintenance inspection than the unsupercharged engine. This result applied particularly to the exhaust valves. About the only concrete evidence in this respect is negative, in the sense that inquiries from several sources have not uncovered specific complaints to the effect that turbo-charged engines are more expensive to maintain.

It is quite possible that even a higher rate of turbo-charging in the future may not bring additional maintenance difficulties, assuming that proper design accompanies such increases, but at this writing sufficient experience is not available to show the line of demarcation where additional maintenance might be expected.

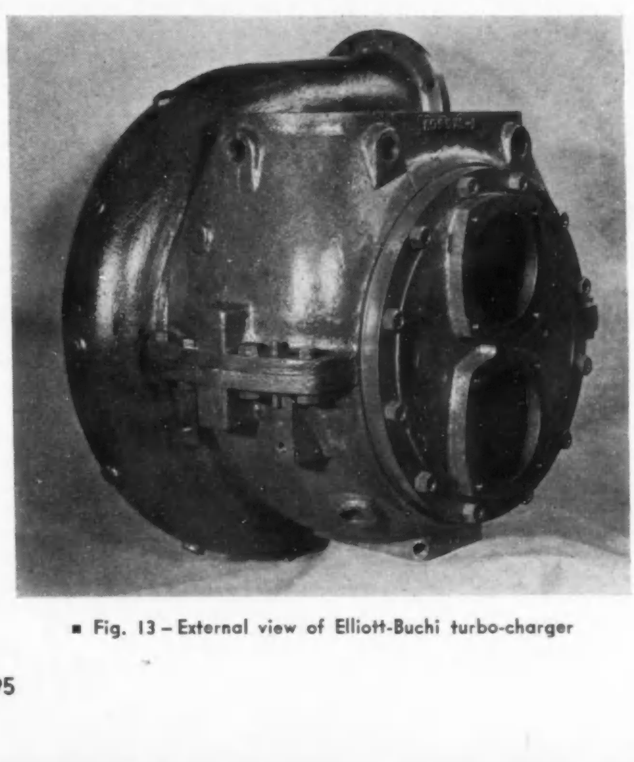
Since the turbo-charger unit itself is such an important adjunct of the turbo-charged engine, and since these units have been manufactured in the United States for such a short time, it is felt that a few words concerning the American-built units would be in order. At this time, there are only two manufacturers of turbo-chargers for



■ Fig. 12 – Cooper-Bessemer turbo-charged engine

the Buchi system in the United States, namely, American Locomotive Co. for engines of their own manufacture, and the Elliott Co. for the diesel-engine industry in general. Fig. 10 illustrates an American Locomotive Co. railway-type engine with Alco-Buchi turbo-charger. It is a 6-cyl, 12½ x 13-in. engine normally rated 1000 hp at 740 rpm, but capable of considerable overload capacity. The turbo-charger is at the left and is supported on the generator. This view shows the blower intake filter attached to the turbo-charger inlet, and the air header along the cylinder heads. Fig. 11 is a view of the lower half of the Alco-Buchi turbo-charger with the rotor in place. The exhaust gas inlet passages are at the left, and the blower inlet is at the extreme right. The rotor construction is more or less conventional, employing outboard bearings at both ends of the machine. The turbine wheel is at the left, and the centrifugal blower impeller is at the right.

Fig. 12 shows a Cooper-Bessemer 8-cyl 15½ x 22-in. engine equipped with an Elliott-Buchi turbo-charger. This combination is capable of a maximum continuous turbo-charged rating of 1575 hp at 327 rpm. It will be noted that the exhaust manifolds are enclosed completely by an insulated jacket, and that the air manifold is directly underneath. Fig. 13 shows the external view of an Elliott-Buchi turbo-charger unit, and Fig. 14 contains a separate view of its complete rotor. The Elliott construction differs some-



■ Fig. 13 – External view of Elliott-Buchi turbo-charger

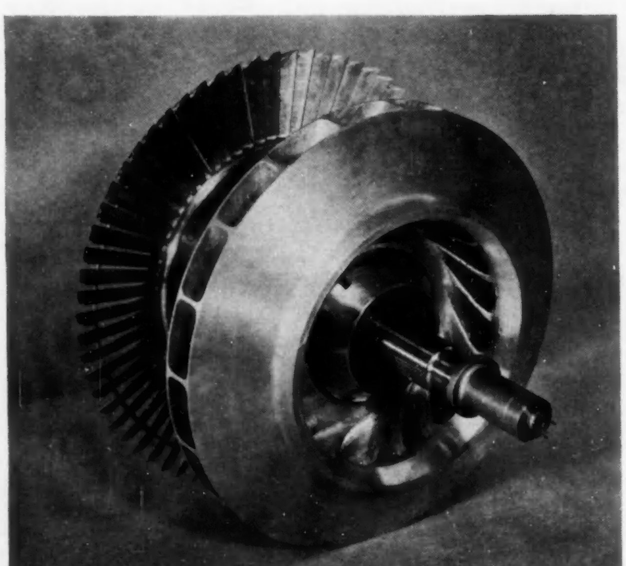
what from that of Alco since, in the former, the rotor is supported entirely from the blower intake end and a bearing is not required at the hot turbine end.

Both the Alco and the Elliott designs have several features in common. The blower impellers are of the conventional enclosed type, and are cast of a special aluminum alloy by a semi-permanent mold process, with 100% X-ray inspection. Plain sleeve type bearings are employed, with lubrication by oil under pressure. Both types of rotor operate at all times below their first critical speed, because of the variable-speed nature of their operation.

The turbo-charger unit itself is a relatively simple machine, requiring only a normal amount of operating attention, principally in connection with adequate lubrication. Since the rotor operates at high rotative speed, it must have clean lubricating oil for the bearings. Alco-Buchi units are being lubricated successfully with engine lubricating oil, and Elliott-Buchi units are available with either this method of lubrication or a separate self-contained system.

Inevitably, the comparison will be made between Buchi turbo-charged engines and those with the positive-pressure supercharger. The Buchi system offers a relatively simple and fool-proof means of supercharging, since the only connections between the turbo-charger and the engine are the exhaust gas and air piping. It does not require a mechanical drive such as V-belts, gears, or chains, and further requires no control since it is entirely self-regulating. The turbo-charger rotates in the same direction regardless of direction of engine rotation, so that it is ideal for directly reversing engines as compared with the mechanically driven positive blower.

Since a large percentage of the power for driving the blower is obtained without expense to the engine, the fuel economy is better with the Buchi system than with positive-pressure blowers, and the maximum power obtainable is generally greater. Buchi turbo-charged engines have a very flat fuel-consumption curve between approximately one-half load and overload. The turbo-charger speed and output depend upon the quantity of gas flowing through the engine, and this quantity in turn is a function of engine speed and load. Therefore, the engine operates practically



■ Fig. 14 – Rotor of Elliott-Buchi turbo-charger

as an unsupercharged engine at low outputs, which in turn brings about a high mechanical efficiency.

On a marine engine, as with a locomotive engine, where the engine is variable speed, and the power requirement reduces rapidly with speed, the exhaust gas turbine is ideal in that it performs only as required, and more or less "floats on the line" when not needed. On the stationary type of engine which operates at constant speed, with variable load, it is again ideal since the turbo-charger adjusts itself to load conditions automatically, while the positive-pressure blower requires substantially the same power at no load as it does at full engine load.

Buchi turbo-charged engines are suitable for practically any class of service where a diesel engine is used. This system of supercharging at present has its greatest usefulness with those engine builders who do not manufacture a few standardized engines in production quantities such as the automotive type, but who have a number of available standard sizes which are adapted to specific requirements in each case. This enables them to offer for sale normally aspirated engines when suitable and turbo-charged engines when required, with few changes from the standard engine construction. With the positive-pressure blower, a mechanical drive must be provided, requiring in the neighborhood of 6 to 10% of the total output of the engine. In many cases the normal camshaft drives are not designed to carry such a load, nor is space available for the blower without affecting overall dimensions.

The question naturally arises as to the future possibilities or trends with respect to supercharged engines. The indications are that supercharged engines must be used increasingly in the future, and it is not inconceivable that the time will come when practically all four-cycle engines will not only be supercharged in some form, but will be designed specifically for supercharging. The most important reason for supercharging is, of course, to obtain more output with less weight and space. At the present time, a cost saving in dollars per total horsepower due to using supercharging can be shown, but it is not of great magnitude. However, there are good reasons to believe that costs will develop to be more and more in favor of the supercharged engine as time goes on and supercharging is more universally used.

The ultimate possibilities of supercharging are still very largely unexplored. A still greater degree of supercharging is possible with the single-stage turbo-charger blower, than is now being used commercially. Multistage blowers with higher charging pressures, coupled with advances in metallurgy to permit higher operating temperatures, are a further possibility.

Fig. 2 shows that 140 psi bmep is already attainable with existing temperature limits. This is by no means the limit, but undoubtedly additional problems will remain to be solved in the development of even higher bmep values. Also, as the degree of supercharging is increased, the greater quantity of fuel burned in the cylinder will bring new combustion problems. No doubt increasingly higher piston speeds will also be used in conjunction with higher values of bmep, and these will bring their own problems in connection with the mechanical design of the engine. To sum up, we feel that the four-cycle supercharged diesel engine is now on the verge of a rapid evolution toward a substantial reduction in specific weight and space, although these objectives will be accomplished only by considerable research and field operating experience.

ICING PROBLEMS in Operation of TRANSPORT AIRCRAFT

by R. L. McBRIEN
United Air Lines Transport Corp.

THE information given in this paper was obtained principally from trip icing logs and test flights conducted on a major transport system. A copy of a trip icing log is included to show the type of information obtained from normally scheduled flights.

The different types of ice formed are explained and the general effect upon the performance on the airplane for each type of ice is stated.

Airplane ice accumulations are divided into two major classes: (1) Those producing a loss of flight performance and (2) those which serve as an annoyance to the crew. The main portion of the paper deals with these two classes, explaining when, how, and why they are of importance. Numerous pictures are shown depicting the various conditions which were found to exist in scheduled airline operations and the shortcomings of the present anti-icing equipment is explained.

Consideration is given to ice accumulations on the wings, empennage, propeller, pitot mast, radio loops, windshield, and so on. It is hoped that a dissemination of these actual airline operating conditions and experiences will result in further improvements to airline anti-icing equipment. It is also hoped that instrumentation means will be developed whereby ice accumulations can be better analyzed and reported.

■ ■ ■

IN presenting this discussion of icing problems attendant to the operation of transport aircraft, it is desired to first express appreciation to all of those United Air Lines pilots who reported in considerable detail their specific experiences when encountering ice formations during routine operation of scheduled flights. Without their genuine interest and cooperation in supplying photographs and supplementary information, a writing of this nature would not be possible. Members of the United Air Lines Engineering Department have contributed substantially by assisting in collecting information and by offering helpful criticism in the arrangement of material. Acknowledgment is also accorded to the flight test and engineering divisions of American Airlines whose conduct of the first tests to determine the nature of ice formation on propellers in flight has been of inestimable value.

The major source of information upon which the following discussions are based is a collection of trip icing logs covering individual experiences of scheduled flights over a period of five winter seasons from 1936 to 1941. These logs are in the form of questionnaires (see Fig. 1), details of which have been completed by our flight officers as the conditions were met. On occasions special flights were conducted into regions where weather data indicated that ice accumulations should be encountered. These flights, however, were not always satisfactory and proved to be an expensive method of study, as the ever-changing meteorological conditions often brought in different air masses before the locality could be reached. Where moderate to heavy icing had been reported by scheduled operations, only a trace or no ice at all would remain when the test airplane arrived. Rather than by continuing to chase these "will-o'-the-wisp" icing regions, the more practical method, though more time-consuming, has proved to be that of maintaining a systematic record of our cumulative experiences gained in the conduct of routine schedules.

■ Types of Ice and Occurrence

Ice accumulation on aircraft has been reported in our experience at indicated atmospheric temperatures between −20 F and +37 F. The frequency with which ice accumulations are encountered at various temperatures over the transcontinental route is indicated by the graph of Fig. 2. The curve shown is plotted on the basis of data obtained from the "Trip Icing Logs" over the past five years.

Lowest altitudes at which ice was reported for each

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 14, 1941.]

TRIP ICING LOG
(1940 - 1941)

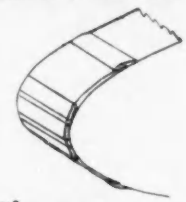
Plane No.________Captain________________1st Officer____________
Trip No.________Date____________From__________To____________
Gross Load____________Lbs. out of______________(Station)________

I. NATURE OF ICE FORMATION

Ice encountered from________to________(location) at________ft. alt.
Temp.__________°F; Length Time in Ice____________.
Avg. thickness ice formed______________Where?______________

II. EFFECTIVENESS OF DEICING EQUIPMENT

1. Wings: Show ice formation on sketch
Was ice glaze or rime?__________
Did ice form on fairing strip?______
How far back of strip?__________
How thick?______________
Did Deicers show any tendency to
"Balloon" when not in operation_____
Which Section?________of__________
Wing.



2. Propellers: Did props throw ice against fuselage?______________
Vibration objectionable?__________Did controls, inst.board, etc.,
vibrate?__________, Did change of RPM reduce vibration?__________
Changed from______RPM to__________RPM

III AIRPLANE AND ENGINE PERFORMANCE.

1. Level Flight. Power used in ice__________RPM.
______________"Hg.M.P.,______________°F Carb. Air, Air speed_____MPH

2. Climb Altitude_______ft., Rate of climb________ft./min.
Power used in ice________RPM,______________"hg.M.P.,________°F Carb.
Air, Air Speed__________MPH

3. Loss of Airspeed in Level Flight Alt.__________ft.
From______________MPH to______________MPH.

4. Did Carburetor ice form?__________How determined?______________

Time required to clear______________at__________°F. carb. air.

IV. PILOTS COMMENTS, GENERAL (Windshield icing, unusual landing or take off characteristics, etc. believed due to adhering ice.)

(Note - Use back of sheet if necessary for additional comments.)
COMPLETED LOG SHOULD BE SENT TO THE OFFICE OF THE SUPERINTENDENT OF FLIGHT OPERATIONS, WHO IN TURN WILL FORWARD IT TO ENGINEERING RESEARCH, CHICAGO.

■ Fig. 1 – Typical trip icing log

month covered by these reports are shown in the following table:

Month	Average of Lowest Altitudes Reported, ft	Lowest Altitude Reported, ft	
October	9,500	6,000	
November	7,000	1,600	
December	7,500	3,000	
January	6,700	600	(Surface)
February	8,100	3,000	
March	8,900	4,500	
April	11,700	6,000	
May	12,000	12,000	

The above altitudes are in reference to sea level.

Meteorologically classified, ice formations fall into three basic types, listed in the following. As encountered, however, adhering ice is usually one or a combination of Types 1 and 2. The degree of roughness and the particular forms resulting are seldom the same for any two separate experiences.

1. RIME – This type is formed by small cloud particles. It is hard but porous, white and opaque. Small grains, air spaces, and frost-like crystals are found within the mass. Accumulations of this type of ice form on the leading edges of exposed parts and project forward into the air stream. They exhibit little or no tendency to adhere to the contour of an airfoil outside of a limited region at and near the stagnation point. However, points of localized surface roughness, such as protruding rivet heads, skin laps, and edges of fairing strips, that are exposed to the direct impingement of the air flow also serve as anchorages from which rime-ice protuberances build.

Rime is usually encountered in a stable air mass. If surface temperatures are above freezing, descent to a lower altitude with temperature above 32 F affords the most certain exit from the icing region.

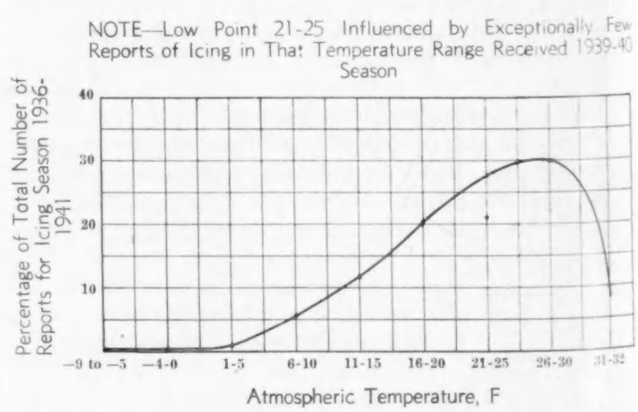
2. GLAZE – Glaze, or clear ice, is produced by freezing raindrops and sometimes by large cloud particles. Its surface may be either smooth or rippled. It adheres very firmly to the surfaces upon which it forms and is most difficult to remove with present available anti-icing equipment. Only on relatively infrequent occasions is pure glaze encountered by aircraft. More often the formation also includes sleet or snow which render its surface rough and irregular. Due to the larger size of the particles contributing to glaze accumulations, higher rates of icing are experienced than with rime. Also larger areas of the aircraft surfaces are covered. The formations tend to adhere more generally to the surface contours of exposed members.

It is characteristic that glaze formations are encountered along sections of the airway where, due to air mass conditions, a temperature inversion exists because of an overrunning warm front. To emerge from glaze-forming conditions, without turning back, the aircraft must ascend to higher altitudes.

3. FROST – Frost may be formed on a metal airplane which suddenly flies out of a cold air mass into warmer air. If the temperature of the metal surfaces is below 32 F and also below the dew point of the warmer air, a sudden ice condensation will result that hardly more than dulls the shiny metal surfaces. The formation soon evaporates as the blast of air passing over the plane quickly warms it to the new air temperature. Occurring during flight, this is of no practical importance. However, airplanes have been reported to exhibit peculiar performance characteristics, during early-morning take-offs, that were attributed to frost accumulations forming on the top wing surfaces of planes which had been parked outside of hangars overnight. The regular use of canvas wing covers, which are removed prior to dispatching, has been successful in eliminating any hazard from such cause.

■ Airplane Ice Accumulations

The ice formations adhering to the component parts of an airplane may be grouped into two major classes: (1)



■ Fig. 2 – Frequency of airplane ice accumulation as affected by various atmospheric temperatures

those producing a loss-of-flight performance, and (II) those which serve as an annoyance to the operators or occupants without primarily affecting the flight characteristics inherent to the design of the aircraft or its engines.

I. Performance-Reducing Formations

Ice accumulations causing a loss in performance of aircraft will here be identified by the name of the part of airplane on which they form and accordingly the performance of which part they directly affect.

1. Wing Ice – The flow of air about the surface contours of a wing results in the production of forces which support the airplane in flight. This flow of air, and the resultant forces, follow in definite relationships with regular changes in the contours of airfoil sections. For instance, changing the wing section by the extension or lowering of the wing flaps produces a definite, predictable change in the forces acting on the wing. Irregular changes in the wing section occurring at unprescribed locations, however, result in an indefinite and unpredictable change in the supporting forces. Briefly, this is the background of the importance attached to the prevention of or removal of ice from airfoils whose function is to support or propel aircraft in flight. It is not the purpose of this paper to delve into aerodynamic theories regarding the influence of different ice formations. Rather, it is intended to describe and to illustrate, in so far as possible from our experience, the variety of types of ice which are encountered and to discuss the use and limitations of present de-icing systems.

Wing de-icing of the large majority of commercial transport aircraft operating in this country is by means of rubber inflation tube de-icers. These devices are installed on wing leading edges, and their function is to remove ice after it has formed on the rubber surfaces by the stretching of the rubber in contact with adhering ice. The tubes extend along the wing's leading edge spanwise and, as they inflate they will, under favorable conditions, crack their covering of ice loose so that it is caught by the air stream and blown clear of the wing.

Removal of ice forming on wings as described in the preceding paragraph is not always assured by merely starting de-icer operation as the icing region is entered and stopping their operation after emerging. Experience has shown that such a procedure often aggravates ice roughness along the wing without removing any appreciable amount. Fig. 3 illustrates the condition developed in one instance when de-icer operation was continuous through a region where glaze ice was being accumulated. This was only a light icing condition but the ice was forming at such a rate that, between each successive tube inflation, only a thin coating of ice was added to the de-icer. Upon inflation this thin coating, rather than being blown clear, was crazed into many small patches of ice which remained on the rubber surface and served as anchorage points for further building of ice on them as bases. Successive inflations produced the same patterns in crazing the accumulating ice with the result that many individual projections formed to present the roughened de-icer surface covering the wing's leading edge. The smoothness of the coating of glaze ice over the landing light gives an indication as to the evenness of the adhering ice for portions of the wing not equipped with de-icer shoes. The procedure adopted as a result of such happenings has been purposely to avoid operating the de-icers until an ice thickness of approximately ⅛ in. is attained. They are then operated for a short period to sufficiently crack loose this thicker formation so that it is blown from the wing in sizable sheets. The de-icers are then turned off until the ice thickness again approximates ⅛ in. Intermittent ice removal in this manner is continued until the region is passed. The thicker ice does not craze extensively as does the very thin ice under which the de-icers work when they are operated continuously at and from the time the icing zone is first entered.

At night the intermittent procedure cannot be followed easily due to the inability of flight personnel to observe satisfactorily the progress of ice formation. Lights are being installed in the engine nacelles which will have sufficient power to illuminate the wing along its leading edge and permit close observation of the conditions developing.

During the building up of formation at a very rapid rate, however, it may be advisable to operate the de-icers continuously as it is possible to permit a thickness of ice to build up which is strong enough to prevent inflation of the tubes. With present de-icing equipment its successful operation depends much upon the experience of flight personnel and upon their judgment in its use.

Fig. 4 shows a ridge of rime ice ¾ in. thick and 1½ in.

■ **Fig. 3 – Ice accumulation when de-icer operation was continuous through a region where glaze ice was being accumulated**

wide extending along the wing leading edge of a transport plane in flight. This is considered a light condition. Flight characteristics were not affected in any way and the de-icers were not operated.

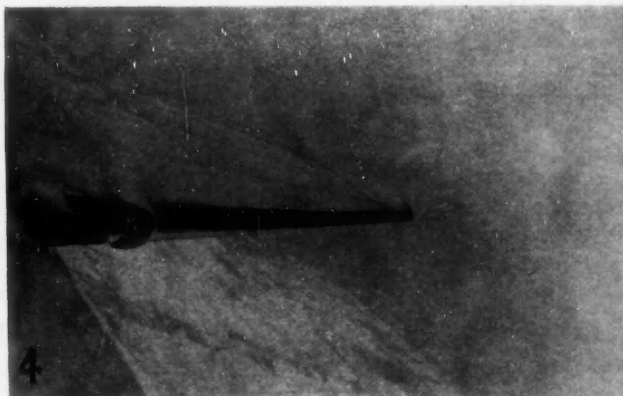
Fig. 5 is of a characteristic rime formation between engine nacelle and fuselage where de-icers are not installed.

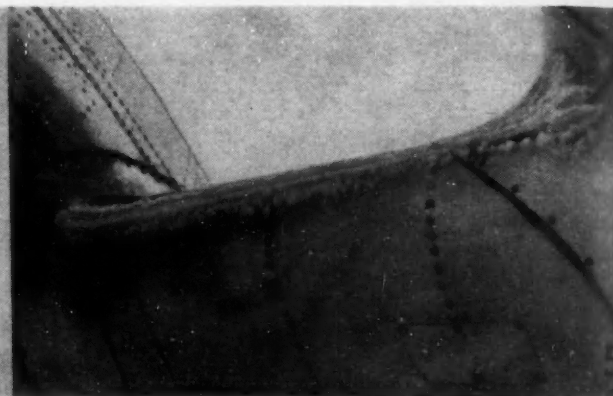
Fig. 6 depicts tenacious rime ice formed at 15 F which de-icer operation did not dislodge from the wing-tip section. Note also accumulations on protruding wing-tip navigation light.

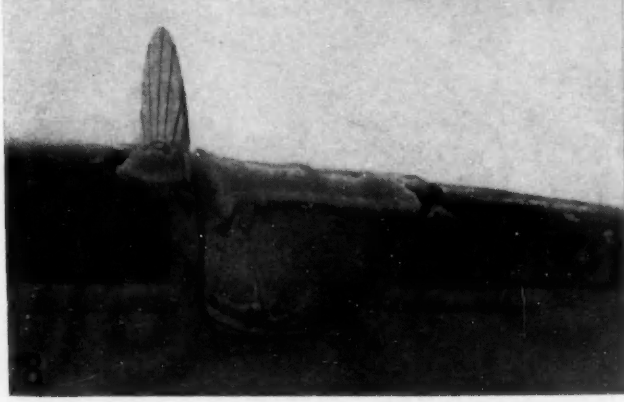
Fig. 7 shows ice remaining on landing light. De-icers had given very satisfactory operation in this case. Lack of means effectively to prevent ice formation on lights has, on occasions, necessitated night landings without appreciable illumination from the airplane itself. Conditions such as this have brought retractable landing lights into favor with certain transport designers.

Rime and glaze ice formations remaining on landing light are shown in Fig. 8. The de-icer apparently had removed the rime satisfactorily but still has patches of glaze adhering to its surface.

Fig. 9 – Rime ice on wing-tip navigation light indicates need for flush installation to clean up wing aerodynamically.







■ Fig. 4 – Ridge of rime ice $\frac{3}{4}$ in. thick and $1\frac{1}{2}$ in. wide extending along the wing leading edge of a transport plane in flight

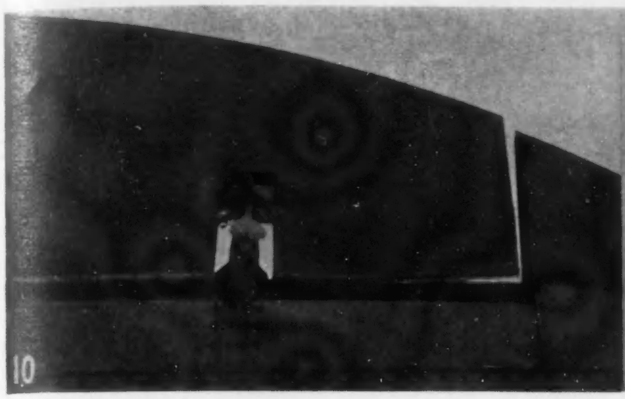
■ Fig 6 – Tenacious rime ice formed at 15 F which de-icer operation did not dislodge from the wing-tip section

■ Fig. 8 – Rime and glaze ice formations remaining on landing light

■ Fig. 5 – Characteristic rime formation between engine nacelle and fuselage where de-icers were not installed

■ Fig. 7 – Ice formation on landing light

■ Fig. 9 – Rime ice on wing-tip navigation light



■ Fig. 10 – Rime ice on aileron hinge

■ Fig. 11 – Comparison of 1941 de-icer (left of light) and 1940 installation

Fig. 10 – Rime ice on aileron hinge indicates desirability of hinge covers, formerly omitted, but which are now installed.

Fig. 11 – Comparison of 1941 de-icer (left of light) and 1940 installation. The 1940 type does not conform smoothly to wing's leading-edge contour. Inflation tubes bulge from surface even when deflated. Attachment fairing strip, exposed, often was source of aggravation as ice sometimes formed in a ridge along its line of contact with de-icer where, especially on the top side of the wing, this ice ridge was considered to be critically located from an aerodynamic analysis.[1] The 1941 design has extended the rubber surface back over the attachment fairing and backed the remainder of the shoe with sponge rubber to smooth out the installation as shown.

An unusual case of wing ice accumulation was reported during January, 1941. The captain of one of the night trips between Denver and Omaha reported that his flight was at 9000 ft in a moderate rain. Outside air temperature was 31 F, but windshield ice was not forming, and de-icer shoes were not used as ice was not forming on the leading edge of the wing. A steady falling off of the airspeed from an indicated 155 mph to 125 mph made him suspicious of an increasing parasitic resistance. An inspection of the wings with a long-beam flashlight showed a rough accumulation of ice on the de-icer attachment fairing strip and forming on the top surface of the wing in patches from 1 ft inside the landing light to the tip. During descent when a temperature of 35 F at 7000 ft was reached, the ice melted and washed off in the rain, and the airspeed indication returned to normal.

An explanation of this occurrence is that the scrubbing action of the impinging rain kept all leading edges free of any ice that might have had a tendency to form. As the water drops streamed back over the top surface of the wing and entered the region of high negative pressure, they were turned to ice as a result of evaporation freezing.

Rubber inflation de-icers bring up other points that bear mentioning during a discussion of their use. After their rubber ages or takes on a slight permanent set as a result of the initial tension with which it is essential to install them, they lose some of the resilient qualities originally possessed. As a result, they do not recover rapidly during tube deflation to the contour of the wing surface. Eventually "ballooning" develops. Ballooning is a lifting of the de-icer from the wing surface which makes the part appear to be partially inflated, though system air pressure is not present in the tubes. In advanced stages, the elastic area of the shoe between the attachment strip and the inflation tubes is raised from the upper wing surface by the forces created by airflow over the wing at that location and the aerodynamic lift of the wing is reduced. Such shoes, of course, must be replaced as soon after detection as possible. Employment of vent holes in low-pressure areas has only partially relieved the likelihood of ballooning.

Powdered graphite with a suitable vehicle has been used to provide lubrication for the de-icer as it stretches over the metal wing surfaces. However, moisture which becomes trapped between the de-icer and the wing surface acts as an electrolyte between the graphite and aluminum and brings about corrosion of the wing leading edge. Talc is now used as de-icer lubricant. This material has eliminated corrosion, though its lubricating properties are inferior to those of dry graphite.

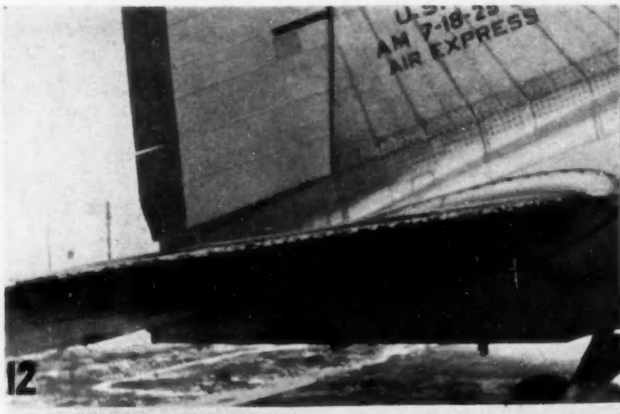
Because of cuts produced by particles blown against de-icers by propellers, frequent patching is required. When performed in the field, a cold patch provides the only practical means. Patches, of course, roughen the de-icer surface and present protuberances which aggravate the formation of ice and tend to defeat the purpose for which the installation is intended.

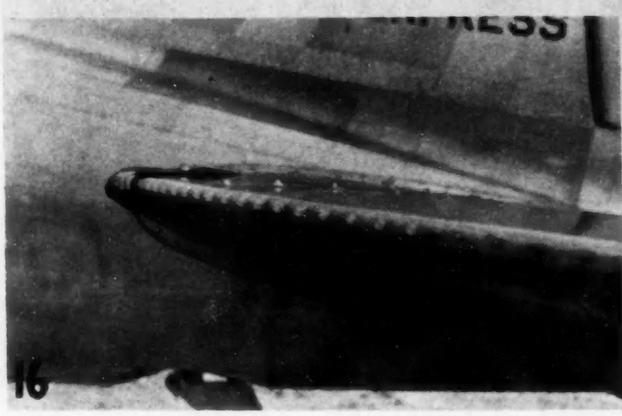
Sunlight and weather soon cause de-icer rubber to craze. This materially roughens the surface and reduces effectiveness of ice removal. Inclusion of Neoprene has improved this condition noticeably but has not entirely eliminated it.

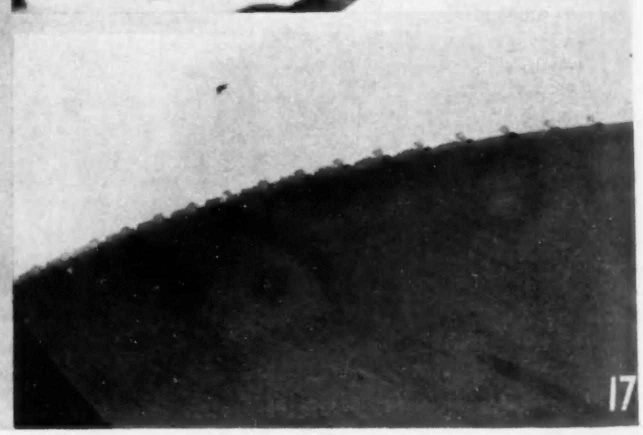
2. Empennage Ice – The importance of ice removal from the tail surfaces of aircraft evolves from the effect which accumulations may exert on the directional and longitudinal control of flight.

The most notable case reported in our experience in which empennage ice figured is one concerning longitudinal control obtained by operation of the elevators. One of our flights, having emerged from an icing region after a beam procedure through the overcast, made a normal approach for landing at a terminal station. A narrow strip of ice about 1½ in. wide and ⅜ in. thick remained along the extreme leading edge of the wing. The descent to the field was normal in all respects, and the pilot had leveled the plane's flight path about 5 ft above the runway preparatory to bringing the tail of the airplane down for a

[1] See *Journal of the Aeronautical Sciences*, Vol. 7, No. 1, November, 1939: "Analysis of Best Rate of Climb and Ceiling as Affected by Ice Accretions," by Albert Gail.







■ Fig. 12 – Rime ice remaining on stabilizer and not removed by sine-tube de-icer during flight

■ Fig. 14 – Rough rime deposit remaining on stabilizer tip before de-icers were extended to include this section

■ Fig. 16 – Rime ice protuberances adhering to exposed rivet heads on stabilizer

■ Fig. 13 – Patches of glaze ice not completely removed by stabilizer sine-tube de-icer

■ Fig. 15 – Side view of rough rime formation shown in Fig. 14

■ Fig. 17 – View from below leading edge of stabilizer shown in Fig. 16

three-point landing. The nose of the airplane suddenly became very heavy. At 85 mph indicated airspeed, the airplane began to shake. The nose dropped and the control wheel fell back into the pilot's lap with no resistance from the elevators. Before the throttles could be opened, the airplane hit the ground on both wheels. The maximum indicating pointer of the accelerometer registered 3.8 G. Inspection of all surfaces at the station revealed scattered pieces of ice ½ in. thick on the leading edge of the horizontal stabilizer which the de-icers had not removed. There were also ice accumulations beyond the de-icers on the tip sections.

There was no sudden falling off to one side or the other as is characteristic of the stall of the wing of the DC-3 airplane. It is believed that the tail section had stalled causing loss of the downward load on the elevators. This allowed the nose to drop (or the tail to come up). Stabilizer de-icers have now been extended around the tip section.

Present procedure for landing an airplane that has come through an icing region is to come in with a considerable margin of speed above the stall and land on the main wheels with the tail high.

Photographs illustrating empennage ice formations are

included here. Most of the conditions shown are for mild cases but will serve to reveal some of the different types encountered.

Fig. 12 – Rime ice remaining on stabilizer and not removed by sine tube de-icer during flight.

Fig. 13 – Patches of glaze ice not completely removed by stabilizer sine tube de-icer.

Fig. 14 – Rough rime deposit remaining on stabilizer tip before de-icers were extended to include this section. Note that, in this instance, the de-icer proved quite effective.

Fig. 15 – Side view of rough rime formation shown in Fig. 14.

Fig. 16 – Rime ice protuberances adhering to exposed rivet heads on stabilizer. Note protuberance at point where de-icer attachment fairing strip meets rubber surface.

Fig. 17 – View from below leading edge of stabilizer shown in Fig. 16 illustrating how rime protuberances often build from small surface discontinuities directly forward into air stream.

Empennage de-icers in general suffer from the same deteriorating elements that reduce the effectiveness of wing de-icers. They receive more severe punishment from objects blown against them by the propeller blast. It is an essential part of the taxiing procedure, particularly on a field with slushy spots on the runways, to position all control surfaces so that they will be least subject to damage caused by pieces of ice that may be blown against them by the slipstream. Similar care must also be exercised during take-off and landing runs.

3. Propeller Ice – Since propellers supply the essential motive force by converting engine power into thrust, the prime importance of keeping them ice-free requires no further elaboration. One additional factor, however, is influenced by ice accretions on them aside from aerodynamic efficiency. That is, dynamic unbalance of the rotating blades. Without ice removal unbalance may become so objectionable that flight cannot be continued.

The anti-icing system now employed by transport operators with some degree of success is the slinger-ring discharge-tube arrangement. This is comprised essentially of an annular ring bolted to the propeller hub with a circular channel around its periphery. From points in the channel near to each blade outlets are provided through open-ended tubes extending and almost touching the blade shanks.

An open-end supply line secured to the engine nose section overhangs the circular channel and pours fluid into it as it rotates with the propeller. Centrifugal force then serves to drain the fluid from the channel through the discharge tubes to the surface of each of the blade shanks. Here centrifugal force is further effective in causing the fluid to flow radially out along the blades thereby producing a surface to which ice cannot adhere. The problem is to obtain a proper and adequate distribution of the fluid out along the blade.

This system was quite successful a few years back on slower-speed transports whose propellers were of shorter radius and were driven directly from the crankshaft at the same rpm as the engine. The speeds of present-day transports have been increased. Propellers are driven through speed-reduction gearing and their blades are of longer radius. These later trends all serve to lessen the effectiveness of the anti-icing system as originally developed. Several attempts to improve coverage of blades with the fluid were made by experimenting with different locations of the discharge tubes. The best distribution that we have been able to obtain is shown in Fig. 18. This is for the propellers of DC-3 airplanes equipped with Hydromatic propellers driven from the engine through 16:9 reduction gearing. The pattern was obtained during cruising flight of 160 mph indicated airspeed with engines turning 2050 rpm (propellers 1153 rpm), and with anti-icer fluid flow at the rate of 3 qt per hr per propeller for 10 min. As will be observed, the fluid covers the shank and leading edge of the blade down to a point opposite the entering edge of the engine ring cowl. There the flow pattern breaks sharply toward the blade's trailing edge because of the windage resulting from the airplane's forward velocity.

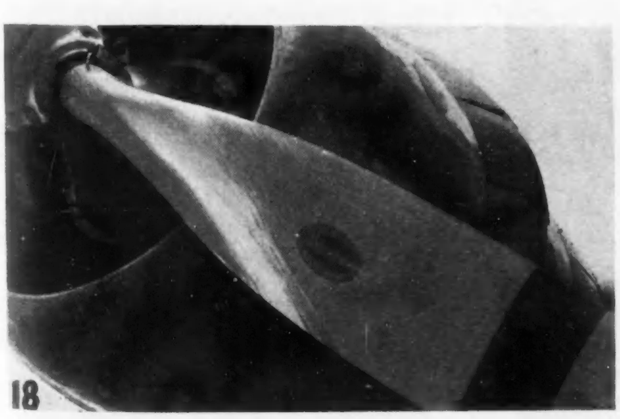
An idea of the extent to which this system accomplishes ice removal during flight was first obtained by American Airlines in a series of flights conducted in the vicinity of Chicago during the winter of 1938-39, and on which the writer was extended the courtesy of an invitation to ride as observer and photographer. The procedure followed was that of flying in an ice-forming overcast with the anti-icer system of one propeller not operating. After sufficient time had elapsed for the accumulation of an appreciable amount of ice, the engine on that side was shut down and the propeller feathered so that rotation was stopped. Pictures were then taken of the ice accumulations that had formed.

Fig. 19 shows the propeller ice, after feathering, which formed during operation without use of the anti-icing system.

Fig. 20 is of the same propeller, after refeathering, following 10 min operation of the anti-icer system under the same conditions as preceded the taking of the photograph of Fig. 19.

It will be noted that the fluid was fairly effective in removing ice from the shanks of the blades, but that ice farther out the leading edge was not disposed of so readily. The formations of Figs. 19 and 20 were encountered during only light icing at a temperature of 31 F, just below freezing, so were not considered a severe test of the anti-icing system.

During other flights of these tests, it was desired to determine the effect of different rpm's on the extent to which ice would adhere to the blade. Stripes were painted on the blades 6 in. apart, the first stripe 6 in. from the hub. Pictures were taken after feathering during flight as be-



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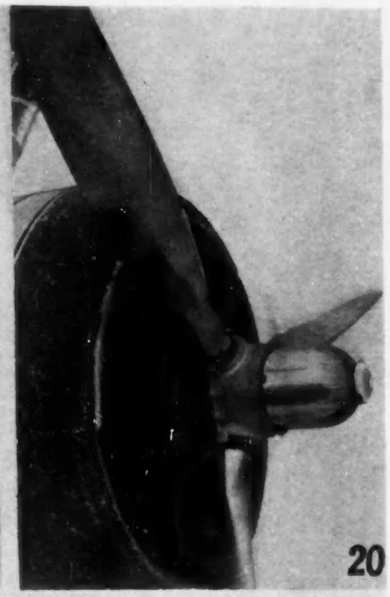
■ Fig. 18 – Distribution of fluid obtained by propeller de-icing system



fore. Anti-icer fluid was not used on the propeller under test.

Fig. 21 was taken after feathering following operation at 1443 propeller rpm (2100 engine rpm with 16:11 reduction gearing). The outer extremity of the main portion of the rime ice came at approximately 13 in. from the hub.

Fig. 22 taken during the same flight followed operation at 1304 propeller rpm (1900 engine rpm). This slower speed permitted appreciably more ice to remain on the blades. The first step off is at approximately 18 in. from the hub, the second at 24 in. A thin coating extends further out on each blade.

Because of the inadequacy of the unaided slinger-ring discharge-tube system under some conditions, an attempt is being made to supplement it with grooved rubber feed shoes to conduct the fluid a greater distance out on the leading edges of the propeller blades.

Fig. 23 illustrates the first installation of Goodrich anti-freeze feed shoes for flight and shows the manner in which the grooves are molded.

Fig. 24 shows the fluid coverage obtained during flight with the feed shoes.

Before using feed shoes in scheduled operation, it was necessary to demonstrate by flight performance tests that the effect of the installation on propeller efficiency was not objectionable. Tests which we conducted last fall showed a loss of only 200 ft from 12,000 ft in usable single-engine ceiling.

Several installations of a revised model of the feed shoe are now in service. A number of favorable reports have been received concerning their effectiveness in icing weather. Sufficient experience has not been obtained as yet for a full appraisal of their merit, however.

An incident which merits special mention occurred when the engines of one of our

▪ Fig. 19 – Propeller ice formed when flying through ice-forming overcast without use of anti-icing system

▪ Fig. 20 – Same propeller shown in Fig. 19 following 10 min operation of the anti-icer system

▪ Fig. 21 – Distribution of rime ice formed at 1443 propeller rpm shown by stripes 6 in. apart

▪ Fig. 22 – Rime ice formed at 1304 propeller rpm

▪ Fig. 23 – First installation of Goodrich anti-freeze feed shoes for flight

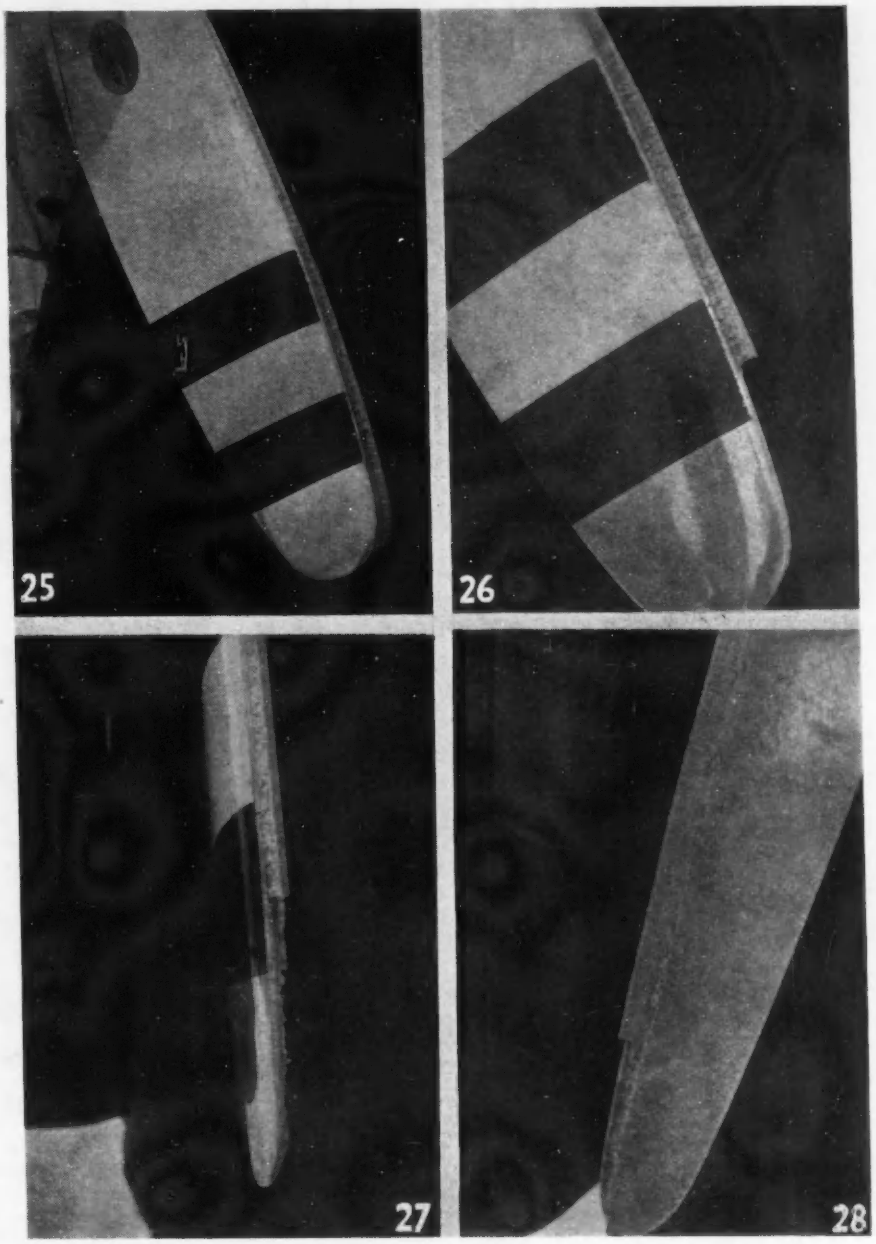
▪ Fig. 24 – Fluid coverage obtained during flight with anti-freeze feed shoes shown in Fig. 23

airplanes were being warmed up at Moline, Ill., on the morning of Jan. 27, 1941. Due to lack of hangar facilities, it had been necessary to park the airplane out-of-doors during the night and to run up the engines occasionally to insure easy starting prior to dispatching a trip which had been delayed there due to weather conditions. The outside air temperature was +4 F and the dew point +4 F. Visibility was 1/8 mile, a dense fog prevailing. Both temperature and dew point had dropped from +12 F during the preceding 1½ hr. At 8:00 a. m. the engines were warmed up for a period of approximately 15 min at speeds varying from 500 rpm to 1400 rpm. After the engines were shut down, the ice formation shown in Figs. 25, 26, 27 and 28 were observed on the propeller blades. This ice was very hard and was difficult to remove from the blades by manual means. It will be noted from these pictures that the formation was pure rime extending to the tip of the blade. The thickness of the ice adhering to the leading edge increased uniformly with blade station radius from the hub to a maximum of approximately 1 in. at the tip. Two of the propeller blades presented the appearance shown by Fig. 25. The third blade had thrown off a portion of the ice near the tip as shown by Figs. 26, 27, and 28. From Fig. 26, it would appear that the dislodging of the section of ice near the tip had left that portion of the blade clean and free of any other adhering particles. However, inspection of Fig. 27 taken practically straight ahead of the leading edge and Fig. 28 of the flat side of this same blade show that such was not the case. Rough formations perpendicular to both cambered and flat sides of the blade near the leading edge still remained. It can be appreciated that the aerodynamic efficiency loss that would result from this accumulation, coupled with the propeller unbalance attending the throwing off of sizable pieces of ice during a take-off run, might prove serious.

Occurrences like the foregoing are very rare and, with present operating procedures, are of no concern since trips are not dispatched under such adverse conditions. It is well to know, however, that such formations can and will form for, with the further development of instrument and radio flying, such a possibility will have to be borne in mind prior to making a take-off under weather conditions approaching those experienced in this instance. In any event, when take-offs are made in conditions where visibility limits are low, it is considered a good procedure to operate the propeller de-icing system during engine run-up and to leave it in operation during the take-off.

4. Parasitic Ice – While ice forming on airfoil sections definitely results in the increase of parasitic drag as well as in a decrease in lift, we will consider here only the accumulation of ice on exposed struts, masts, radio antenna, and the like in this classification. This is because these parts, in themselves, even without adhering ice, cause parasitic drag without contributing to the forces lifting or propelling the aircraft. The photographs shown in Figs. 29 to 34, inclusive, will serve to illustrate the nature of ice accumulations on these parts.

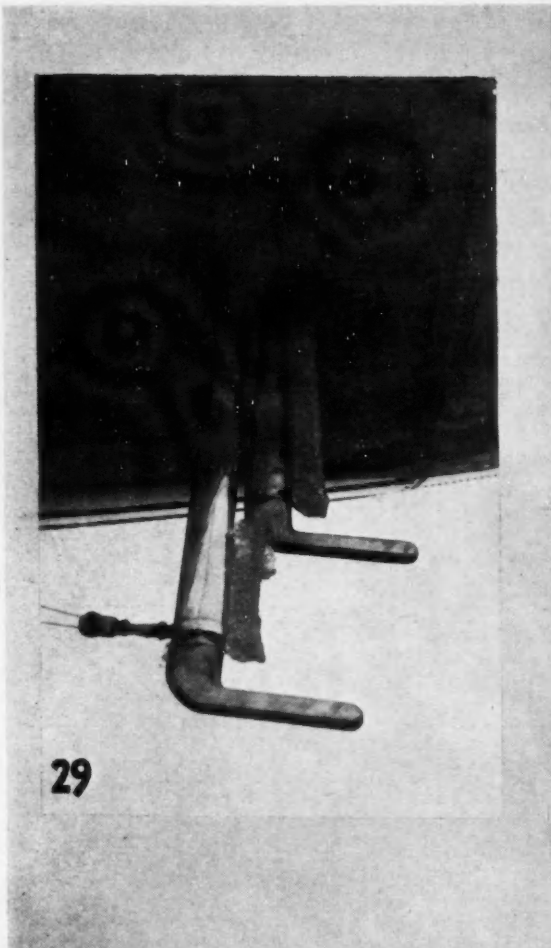
5. Carburetor Ice – This subject is a special study in itself and will only be mentioned in this writing as being a part of the general icing problem. No attempt will be made to present an analysis of this problem in this paper. Briefly, it can be said that the major difficulty is detecting the presence of carburetor ice prior to the time when it has built up to serious proportions in the scoop or adapter. It cannot be seen, and no suitable ice detectors are avail-

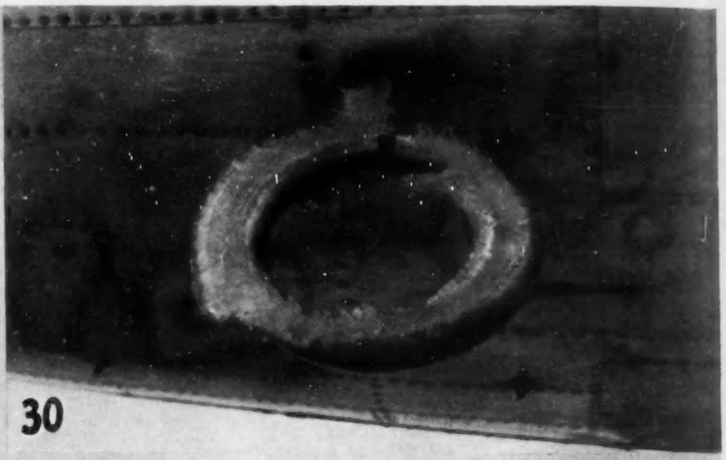


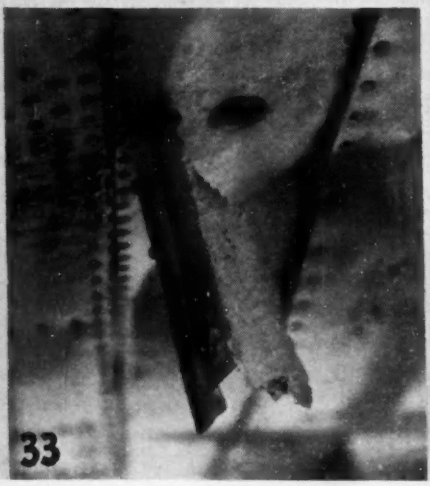
■ Fig. 25 – Ice formation on two propeller blades after 15 min warm-up and subsequent shutdown of engines during dense fog and temperature and dew point of 4 F

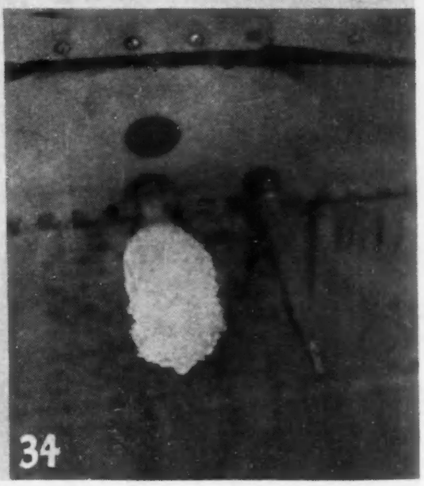
■ Figs. 26, 27, and 28 – Ice formation on third blade of propeller of Fig. 25, under same conditions

able. It varies with different installations. It must be anticipated by the operating personnel. Carburetor ice formation does not generally coincide with wing ice. Much of it is formed directly within the carburetion system by the refrigeration produced at the time the fuel is sprayed or aspirated into the carburetor air. Carburetor ice can be prevented by adequately preheating the carburetor air (100 F has proved adequate for most installations and fuels), although this method results in some loss in power if it is necessary to use full throttle. The injection of









■ Fig. 29 – Pitot tube masts with $1\frac{1}{2}$ in. rime ice

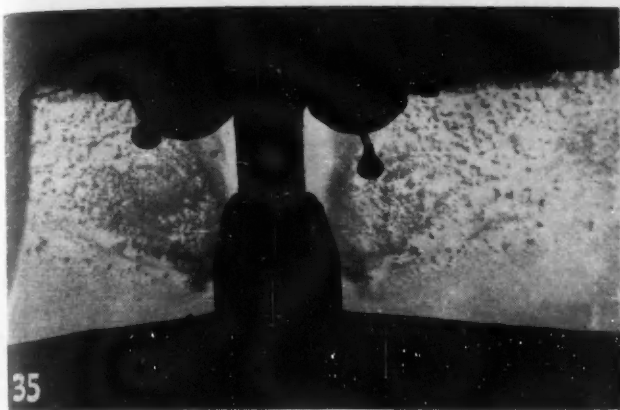
■ Fig. 31 – View from below and in front of radio loop antenna shown in Fig. 30

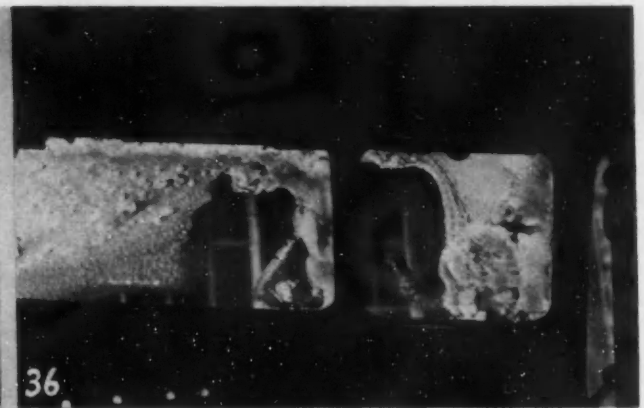
■ Fig. 33 – Rime ice accumulation on de-icer exhaust tube, side view

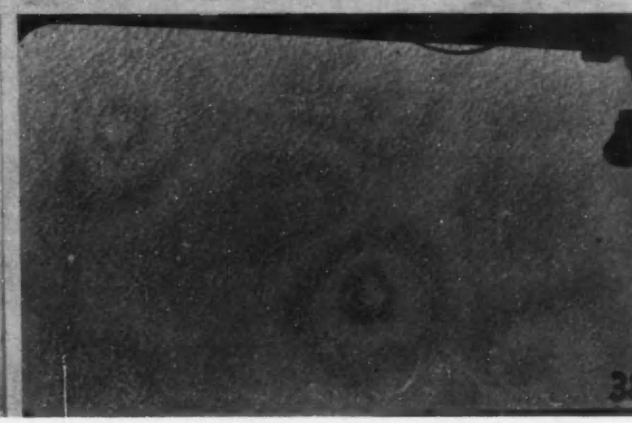
■ Fig. 30 – Side view of radio directional loop

■ Fig. 32 – Ice accumulations on masts and radio antenna underneath airplane – note loose antenna, which has broken from vibration caused by adhering ice, hanging below center section

■ Fig. 34 – Rime ice on de-icer exhaust tube, front view







■ Fig. 35 – Light scattered rime which does not totally obstruct but which may confuse vision – photograph taken during flight

■ Fig. 36 – Heavy rime on windshield through which visibility has been maintained by use of heated air blast directed against inside surface of glass by duct outlets at top edge – photograph taken on the ground

■ Fig. 37 – Outside view of airplane with windshield ice accumulations shown in Fig. 36 – note also dents in side of fuselage caused by ice thrown from propellers

■ Fig. 38 – Complete obstruction to vision produced by pure glaze – photograph taken during flight

alcohol into the carburetor scoop in the form of a spray is also often used as a means for removing ice already formed.

Several current independent carburetor ice investigations are now under way and should provide adequate material for engineering papers later this year.

II. Annoyance Formations

1. Windshield Ice – The obstruction of the pilot's forward vision by ice accumulations on airplane windshields is probably the most outstanding and most frequently encountered annoyance formation. Illustrations of different extents to which visibility may be affected are presented in Figs. 35 to 38, inclusive.

The most common method now in use for anti-icing windshields is the use of a heated air blast on the inside of the glass. A tap off from the airplane's cabin heating system supplies air at the windshield duct outlets at a maximum temperature of 170 F. The system is fairly effective in light and moderate icing conditions. In glaze, such as shown in Fig. 38 and for rime forming at low temperatures, however, it is of no benefit.

One of the purposes of the air blast on the windshield is to keep the inside of the glass from frosting over when the airplane is being taxied on the ground. Different fluids (alcohol, glycerine, ethylene glycol, and so on) have been sprayed against the outside of the glass. None has proved satisfactory in itself. The more volatile ones evaporate so rapidly that the cleared portion of the glass freezes over as soon as their use is discontinued. The low-volatile fluids remain on the glass and become a more undesirable obstruction to vision than the ice, since they cannot be removed even by manual means after the icing zone is passed. Humorous as it may seem, most transport planes are equipped with paint scrapers which the pilot not infrequently uses, after emerging from an overcast, by opening the side sliding window and reaching around to scrape a clear area on the front glass.

Power-driven wiper blades are being used to a limited extent. While they appear effective for use in rain, they are not at present satisfactory for ice removal unless they can be supplemented with sufficient application of heat and anti-icing fluids.

Deflectors have been used over a portion of the windshields of some airplanes, but so far have been of no value other than in very light ice or rain.

2. Pitot Tube and Pitot Tube Mast Ice – Ice forming on these parts also may be classified as a parasitic accumulation. Primarily, though, it is an annoyance. Until sufficient wattage was employed in pitot-tube heating elements, the indications of the airspeed, altitude, and vertical speed instruments were frequently rendered unreliable. At present pitot-tube ice prevention is quite satisfactory. Heating

elements of the new units are rated at 130 w at 12 v. When, at infrequent times, irregularities have been reported, high-resistance electrical contacts have been found which reduce the current flow to the heaters.

Pitot-mast ice, such as the formation shown in Fig. 29, is believed to have figured in reports concerning sizable losses of indicated airspeed. Such formations tend to reduce the velocity past the pitot tube openings, thereby decreasing the differential pressure between the pitot and static tubes. Rubber inflation de-icers have been operated on a few installations and appear to be satisfactory in preventing the growth of large accumulations. The use of anti-icing fluids on the leading edge of the masts is also being tried by some operators with apparent success.

3. Propeller Ice – When propeller anti-icing is not effective, larger strips of ice form on the blades before the action of centrifugal force is sufficient to dislodge them. Then, as they come off they are thrown against the side of the fuselage with resultant disconcerting noise. Often dents are produced on the skin of the fuselage by such ice. See Fig. 37. Propeller vibration resulting from the unbalance caused by ice is sometimes annoying because a shuddering is set up throughout the airplane's structure.

4. Miscellaneous – The functioning of the cabin heating system has been impaired on one or two isolated occasions when, under extreme conditions, the nose air duct (6 in. in diameter) was iced over and the flow of air through the steam radiator stopped. This cut off the circulation of warm air upon which passenger comfort depends.

During flights through dense snow storms, oil radiators have become packed so rapidly that oil temperatures reached near limiting values before the region was passed. Also snow packing inside the engine cowl has on occasions impaired the air flow through the cylinder cooling fins and caused considerably higher than normal head temperatures.

In conclusion it can be stated fairly that much progress has been achieved during the past few years in the improvement of all aircraft anti-icing equipment. However, much still needs to be accomplished before air transport operations will be able to operate without regard to possible icing conditions. At the present time both company and Government regulations prevent operation of scheduled air-transport craft into icing situations where the orderly completion of the trip is dependent upon maximum efficiency of the anti-icing equipment. This limitation does not often act as a restriction from the standpoint of enroute flying, but it does frequently curtail operations because of icing conditions which exist near the ground and near airports.

It has been the purpose of this paper to outline briefly the icing problems which at the moment are considered only partially solved. It is hoped that the material presented will provide additional material about actual operating experiences to those in the industry who may be able to contribute to the further improvement of equipment. Perhaps an entirely new approach to some of these problems is advisable.

It should also be mentioned that instrumentation is needed, which will help analyze icing severity (rate of deposit), type of ice, thickness, effect upon performance (loss of lift, increase in drag, increase in stalling speed), and so on. Much of the ice formed cannot be observed by the crew because of obstructions, darkness, ice on the windshield, and so on. Therefore, dependence upon visual indications and analysis of ice accumulations is inadequate.

Bringing Road Octane Testing Into the Laboratory

MANY laboratories are so located that climatic conditions prevent actual road testing for a part or the majority of the time each year.

From the automotive and petroleum manufacturers' commercial point of view, neither the production of engines nor the blending of fuels can wait on a long period of inclement weather. Therefore, it would be highly desirable if the laboratories of both industries could at all times evaluate their products by some device which would duplicate road test data.

Since the purpose of any device of the type proposed in this paper is to duplicate road-test data, engine data and performance on the road had to be studied. As a starting point for full-throttle borderline work, the data for the engine acceleration curve from 10 to 70 mph was taken on a level road. The reference fuel framework and the 1941 API and CFR and other fuels were then run. It might be well to point out that these same data could just as readily be taken on any grade, thereby giving acceleration curves for any given grade. Further, that any grade, or combination of grades, could be used, even to the point where deceleration might occur as the grade is increased.

The next step would be to plot these data, remove the engine from the car, and proceed to load it on the dynamometer in such a manner that in the laboratory, the acceleration curve under full-throttle conditions duplicated the road acceleration curve under the same engine operating conditions. Obviously, these same data could be taken for part-throttle work, for increments of intake vacuum.

The apparatus, itself, simply consists of an oversized dynamometer, having a carefully, yet flexibly, controlled field excitation current, so that loading can be applied as desired. A dynamometer of sufficiently high capacity was chosen so that, first, it has a fairly high inertia effect of its own relative to the engine; and, secondly, such that extremely low field current values are needed to produce the necessary load. Thus, extremely small fluctuations in field current will produce relatively large torque characteristic changes. Hence, the eccentric mask relationship need be changed only minutely to effect necessary load changes.

The fact that a Borderline reference fuel framework, with a good spread, has been obtained, and fuels have been properly evaluated in relation to this pattern, it is believed, would justify the assumption that unknown fuels could be rated and expected to fall in their proper place on the reference fuel pattern.

Lastly, it should be clearly understood that this type of laboratory correlation work does not, and cannot, replace road testing, but should be looked upon as simply an adjunct to road testing, useful in predicting fuel behavior on the road.

It is to be hoped that other laboratories will proceed along these or similar lines, so that a method, or methods, will shortly be evolved to the end that fuel and engine testing may be made by means of these and similar devices, independent of road and weather conditions.

Excerpts from the paper: "A Proposed Method for Duplicating Road Octane on the Multicylinder Engines in the Laboratory," by Joseph A. Moller, H. L. Moir, F. C. Minor, and R. R. Proctor, The Pure Oil Co., presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 6, 1941.

MERCEDES-BENZ DB-601A AIRCRAFT ENGINE

(Design Features and Performance Characteristics)

by RAYMOND W. YOUNG
Wright Aeronautical Corp.

THE major part of this paper reports a teardown inspection of the Mercedes-Benz DB-601A engine with detailed descriptions and comments on some of the design features of major component parts. It is followed by a check of the magnaflux inspection method, analysis and comparison of materials, and comparison on quality of finish.

Quoting specifications, the author reveals that the Mercedes-Benz DB-601A engine is an inverted 60-deg V-type, 12-cyl in banks of 6, liquid-cooled Otto-cycle engine with fuel injection. Its bore and stroke are 150 x 160 mm; its displacement is 2070 cu in; and its compression ratio is 6.74:1.

Comparing the physical and performance characteristics of the engine with contemporary American, British, French, and German powerplants of the same general design, Mr. Young reports that German and French liquid-cooled designs tend toward larger displacement and lower crankshaft speed while American and British practice favors higher engine speeds with relatively smaller piston displacement; that the Germans favor the inverted type of construction in in-line engines because of the excellent visibility which this type of engine permits in a single-engine airplane; and that American and British engines average about 5.7% higher in bmep under military rating.

Discussing workmanship and quality of finish in his conclusions, Mr. Young points out that no useless effort in man-hours or finish has been expended where there is not a direct return in increased reliability and performance, and that handiwork in the polishing of stressed parts is of the highest order. Although stating that the design represents good mass-production practice for military aircraft engines, he notes that the highly stressed bolts do not have ground threads.

On the important factor of performance, he summarizes: "Despite wishful thinking to the contrary, the performance of the DB-601A with respect to sea level and altitude output, fuel consumption, and weight, seems to be on a par with contemporary powerplants of the same general type."

Discussions of this paper were presented by W. H. Sprenkle and E. K. Von Mertens, both of Pratt & Whitney Aircraft, Division of United Aircraft Corp., but were not prepared for publication.

THE Mercedes-Benz Model DB-601A aircraft engine is a development of the Daimler-Benz Aktiengesellschaft of Stuttgart, Germany, a firm which has been engaged in the manufacture of automotive and aircraft engines for over 50 years. During the first World War the Daimler Motoren Gesellschaft of Stuttgart produced the famous Mercedes aircraft engines in three 6-cyl types with ratings of 160 hp, 180 hp, and 260 hp. Equally renowned were the 160 hp and 230 hp 6-cyl aircraft engines built by Benz and Co. in Mannheim. After the war and as a result of the economic and financial crisis which brought almost complete stagnation to the automotive industry in Germany during the early 'twenties, these two companies were practically forced to combine their activities in order to survive. Accordingly in 1926 a merger was consummated between the Daimler and Benz organizations. Thus came into being the firm of Daimler-Benz A.G. and its product, the Mercedes-Benz line of automotive vehicles and aircraft powerplants.

Restricted by the terms of the Versailles Treaty, German aircraft engine development had remained virtually at a standstill for a number of years, particularly in the high-

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 2, 1941.]

Table 1 – General Data – DB-601A Engine

Name	Mercedes-Benz
Model	DB-601A
Type	Inverted 60 deg vee, liquid-cooled, Otto-cycle, fuel injection
Number of Cylinders	12, in two banks of 6
Bore	150 mm (5.9 in.)
Stroke	160 mm (6.3 in.)
Piston Displacement	33.9 l (2070 cu in.)
Compression Ratio	6.74:1 (average of four cylinders)
Reduction Gear	Spur-gear type, 14:9 $\left(\frac{\text{crankshaft}}{\text{propeller shaft}}\right)$ ratio
Direction of Rotation (From anti-propeller end)	Crankshaft—clockwise; Propeller shaft—anticlockwise
Valve Gear	Underhead camshaft with rocker arms actuating two inlet and two exhaust valves per cylinder.
	Valve clearance (running): Inlet 0.010 in.; Exhaust 0.020 in.
	Valve timing (with above clearances): I.O. 21 deg BTC; I.C. 72 deg ABC; E.O. 50 deg BBC; E.C. 21 deg ATC
Ignition Timing	Full Retard 8 deg ATC; Full Advance 40 deg BTC
Fuel Injection Timing	Beginning of plunger stroke 37 deg ATC; End of plunger stroke 15 deg BBC; Crank angle during injection 128 deg
Supercharger Gear Ratio (Fluid-Drive Rotor)	10.38 x crankshaft
Supercharger Impeller Diameter	10.25 in.
Engine Dimensions	Length 84 in. (overall); Width 29 in.; Height 40.5 in.
Weight (As Received)	1520 lb, including: Propeller shaft, Spark plugs, Ignition harness, Propeller control motor, Fuel injection system, Coolant header tank, Automatic controls, Fuel pump
Heat Rejection to Coolant	20,800 Btu/min
Heat Rejection to Oil	3,650 Btu/min

Ratings	Hp	Rpm	Absolute Manifold Pressure in. hg
Take-off (1 min)	1050	2400	40.4
Normal Sea Level (5 min)	950	2400	36.5
	970	2300	34.7

Fuel Consumption	
Take-off	0.54 lb/bhp/hr
Normal Rating, S. L.	0.49 lb/bhp/hr
Altitude Rating	
75% Cruising Power	0.45 lb/bhp/hr
60% Optimum Cruising Power	0.43 lb/bhp/hr

power-output field. After the merger of Daimler and Benz, however, the aircraft-engine activity was renewed, and this firm in 1928 produced a 12-cyl vee water-cooled powerplant of 800 to 1000 hp known as the F-2 model. This engine was subsequently redesigned as a diesel powerplant with a rating of 750 hp and produced in some quantity in 1930-32. In 1933-34 Daimler-Benz introduced a 16-cyl vee water-cooled diesel airship engine of 900 to 1200 hp which saw active service in the "Hindenburg." During the succeeding years, the development of an inverted 12-cyl vee liquid-cooled aircraft engine was actively pursued by Daimler-Benz. The success of this model, known as the DB-600, was first widely proclaimed on Nov. 11, 1937, when Chief Pilot Wurster of the Bavarian Airplane Works in a Messerschmitt single-seater powered with this engine established a landplane speed record for a distance of 100 km at a speed of 379.6 mph. On June 4, 1938, Gen. Ernst Udet, flying a Heinkel single-seater with a DB-600 engine, re-established the record at 392.5 mph. A redesign of the DB-600 to incorporate direct fuel injection and improved supercharging capacity resulted in the model DB-601. Worldwide publicity was directed to the performance of the DB-601 installed in a Heinkel He.112U fighter on March 30, 1939, when Flight Capt. Dieterle set up a world's speed record of 463.9 mph, succeeding that of 439.93 mph established by the Italian pilot Angello. Within a month, on April 30, 1939, Flight Capt. Wendel established the existing world's speed record of 469.2 mph in a Messerschmitt Me109 powered by a DB-601 engine. It has been reported from abroad that these two particular DB-601 powerplants were special "souped-up" versions which were boosted to produce 1800 hp at 3500 rpm in contrast to the normal maximum power of 1050 hp at 2400 rpm for the standard engine. From the foregoing chronological résumé it will be observed that the development of a military powerplant for the production of fighting aircraft for the Luftwaffe involved a span of ten years and, like similar technical accomplishments in other countries either in the field of aviation or other industry, it was not brought about overnight.

General Description

Within the last few months there has been made available for study in this country several of the DB-601A model Mercedes-Benz aircraft engines. The unit described in this paper was built by the Niedersächsische Motorenwerke of Braunschweig which is probably a licensee of the Daimler-Benz A.G. The material and data which follow are based on first-hand observation and testing of the actual engine. In Figs. 1, 2, 3, and 4, is illustrated the general appearance of the assembled powerplant.

Characteristics and Features of DB-601A

Table 1 is a brief compilation of the general engine data, dimensional characteristics, ratings and fuel consumption of the DB-601A.

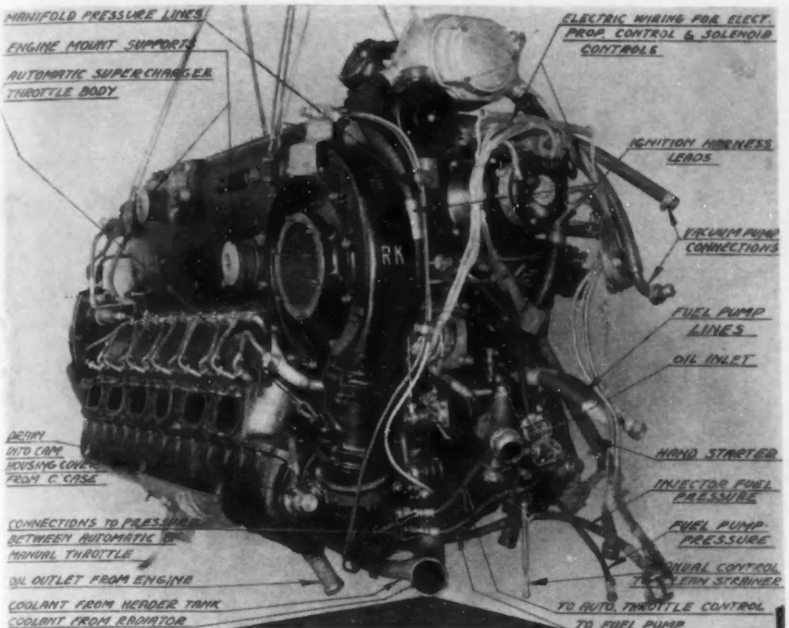
Installation in Messerschmitt & Heinkel Planes

Since the powerplant is an essential and integral part of the aircraft, it is of general interest to illustrate several applications of the DB-601 to German aircraft. Accordingly in Figs. 5, 6, and 7 are reproduced the excellent sketches made by *Aeroplane* of the well-known Messerschmitt Me.109 Fighter, the Heinkel He.112 Fighter, and the Heinkel He.111K bomber respectively, all powered by the DB-601 engine and major units in the striking force of the Luftwaffe.

Comparison with Contemporary Powerplants

Prior to investigating the detail construction of the DB-601A engine, it is of considerable interest to compare its physical and performance characteristics with contemporary American, British, French, and German powerplants of the same general design. Based on published data, a chart has been prepared as Table 2 which lists such a comparison between Mercedes-Benz, Allison, Rolls-Royce, Hispano-Suiza and Jumo aircraft engines for the models indicated. From a consideration of the various items listed, the following observations are made:

(a) German and French liquid-cooled engine designs



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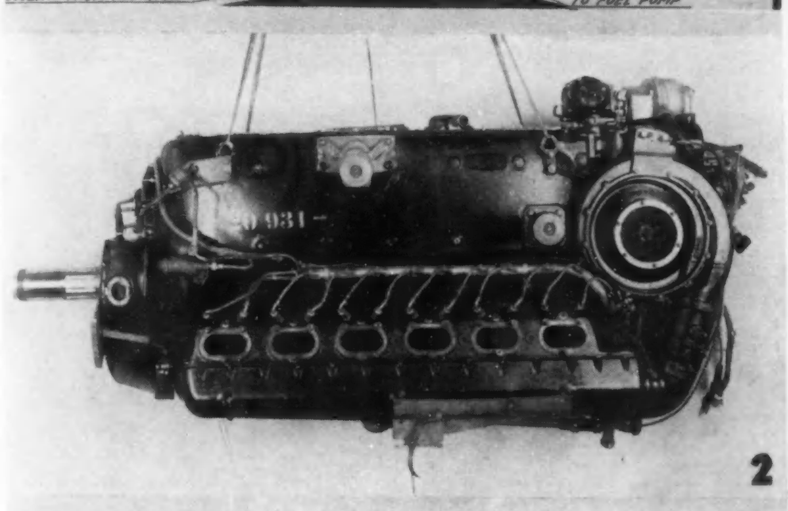

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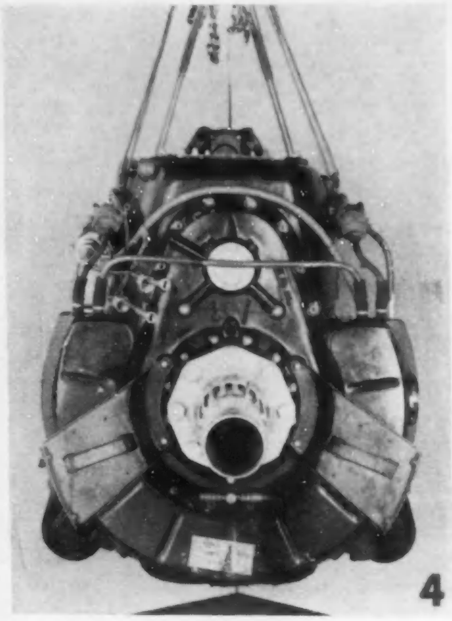

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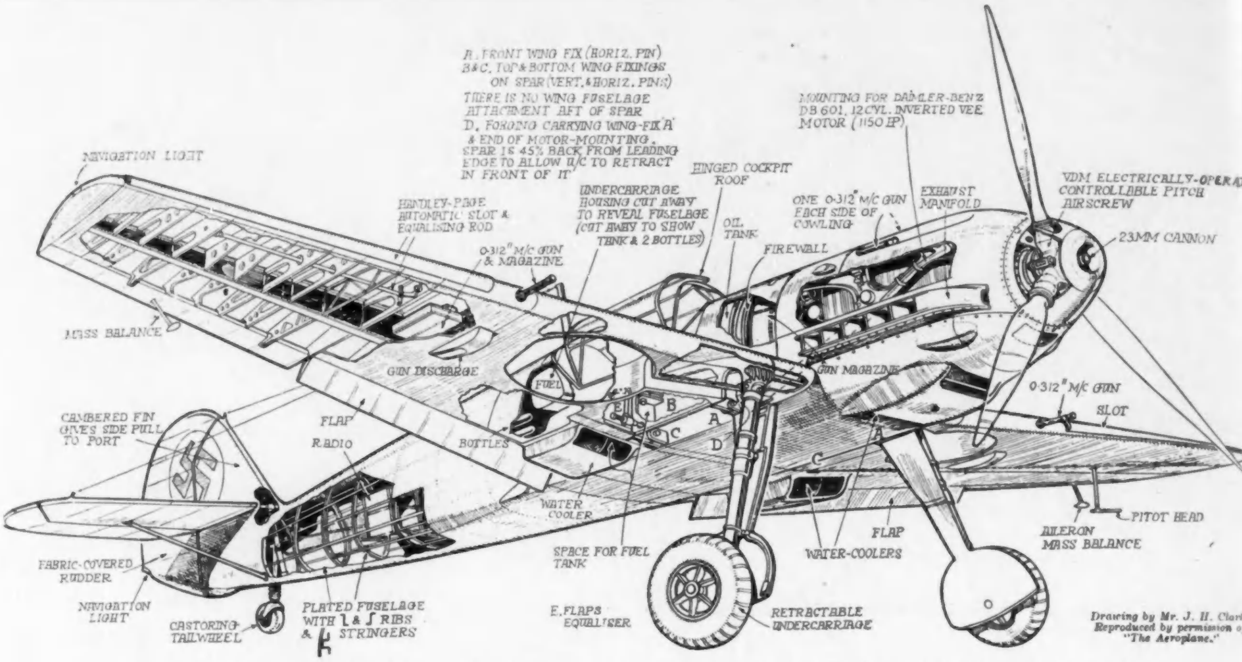
▪ **Figs. 1-4 – Four views of assembled DB-601A aircraft engine**

tend toward larger displacement and lower crankshaft rpm while American and British practice favors higher engine speeds with relatively smaller piston displacement.

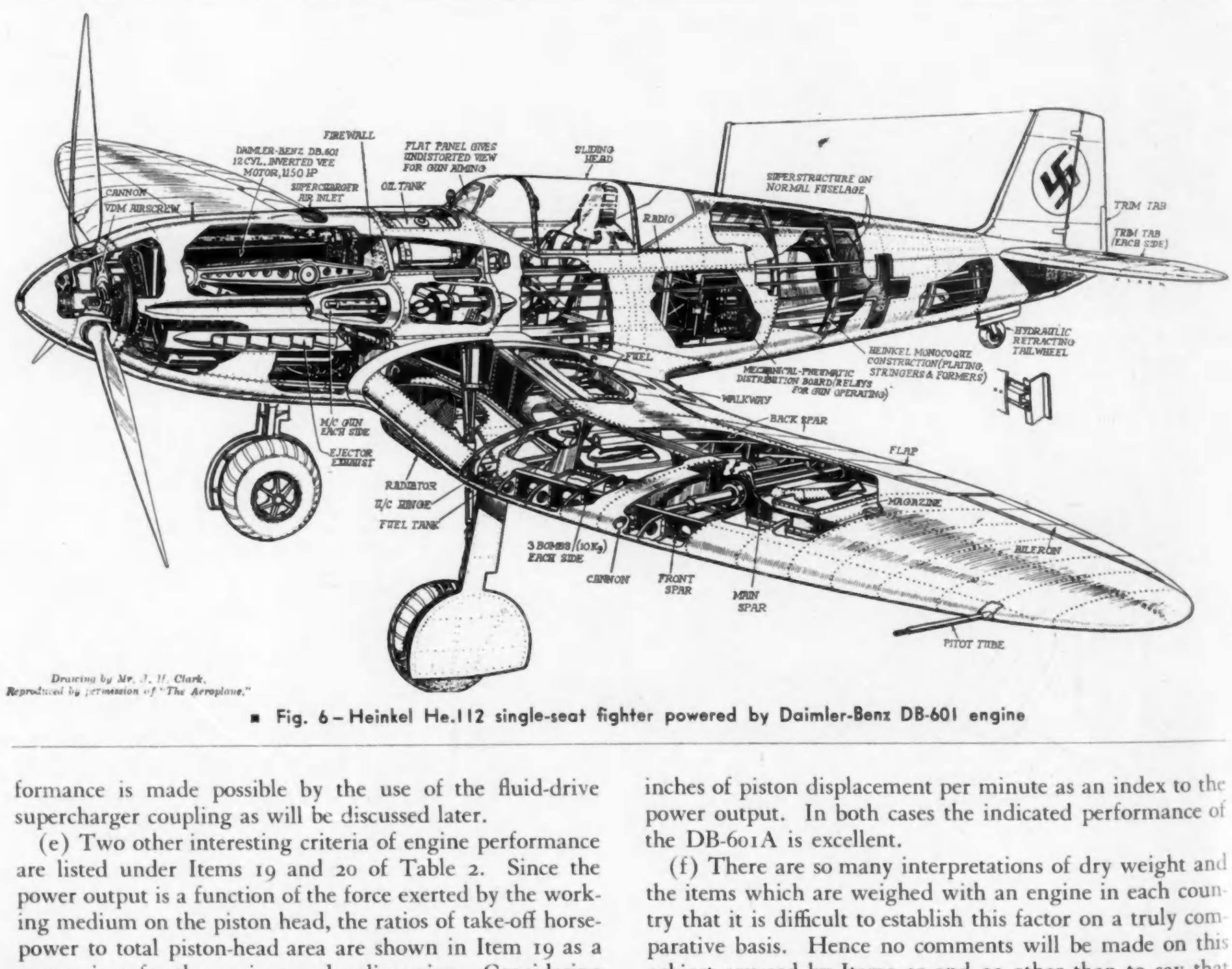
(b) The German aircraft and engine industry favors the inverted type of construction of in-line engines, for not only is this a characteristic of both Mercedes-Benz and Jumo engines, but also of the Argus and Hirth powerplants used in German training planes. It is believed that the basic consideration involved is the excellent degree of visibility for the pilot which the inverted type in-line powerplant permits in a single-engine airplane.

(c) Data on altitude performance may be misleading unless referred to a common basis. In Item 10 of Table 2 is given the published military rating altitude of the various engines. To establish these ratings on a hypothetical altitude basis of 15,000 ft as shown in Item 11, it has been assumed that the power at altitude characteristics for all engines will follow the slope conventionally used in altitude calculations and substantiated by altitude chamber tests. Such a comparison indicates the altitude performance superiority of the Rolls-Royce with its two-speed supercharger and the excellent showing of Allison with a single-speed supercharger. Mercedes-Benz with its fluid-drive supercharger and Jumo with a two-speed supercharger are approximately equivalent on this basis.

(d) Brake mean effective pressure, bmep, covered in Items 14 and 15, is usually considered the conventional index for power-output comparison under different operating conditions. Under military rating it will be noted that American and British engines average 9 psi or 5.7% greater than the German and French engines. For take-off, however, the DB-601A operates at higher bmep than the Allison and approaches, as does the Hispano-Suiza, the bmep values of the two-speed supercharger Rolls-Royce and Jumo engines. In the case of the DB-601A this per-



■ **Fig. 5 – Messerschmitt B.F.W. Me.109 single-seat fighter equipped with 1150-hp Daimler-Benz DB-601 engine**



■ **Fig. 6 – Heinkel He.112 single-seat fighter powered by Daimler-Benz DB-601 engine**

formance is made possible by the use of the fluid-drive supercharger coupling as will be discussed later.

(e) Two other interesting criteria of engine performance are listed under Items 19 and 20 of Table 2. Since the power output is a function of the force exerted by the working medium on the piston head, the ratios of take-off horsepower to total piston-head area are shown in Item 19 as a comparison for the engines under discussion. Considering further the factor of piston speed in engine performance, Item 20 indicates the ratios of take-off horsepower to cubic inches of piston displacement per minute as an index to the power output. In both cases the indicated performance of the DB-601A is excellent.

(f) There are so many interpretations of dry weight and the items which are weighed with an engine in each country that it is difficult to establish this factor on a truly comparative basis. Hence no comments will be made on this subject covered by Items 21 and 22 other than to say that the engines are quite comparable as to weight with the exception of the Hispano-Suiza. The weight advantage in

Table 2 – Comparison of Engine Specifications and Performance

1. Make	Mercedes-Benz	Allison	Rolls-Royce	Hispano-Suiza	Jumo
2. Model	DB-601A	V-1710C-15	Merlin X	12Y-51	211
3. Number of Cylinders	12	12	12	12	12
4. Arrangement	Inverted Vee	Vee	Vee	Vee	Inverted Vee
5. Bore, in.	5.7	5.5	5.4	5.9	5.9
6. Stroke, in.	6.3	6.0	6.0	6.7	6.5
7. Piston Displacement, cu in.	2070	1710	1647	2197	2136
8. Military Rating, hp.	1000	1090	1025	1100	975
9. Military Rating, rpm.	2400	3000	3000	2400	2300
10. Military Rating Altitude, ft.	14760	13200	17750 (High Blower)	10696	15584 (High Blower)
11. Hypothetical Horsepower at 15,000 ft	990	1020	1150	920	990
12. Take-off Rating, hp	1150	1040	1045	1100	1100
13. Take-off Rating, rpm	2500	3000	2850	2400	2400
14. Bmep (Military Rating), psi	158	168	164	156	157
15. Bmep (Take-off), psi	167	160	176	166	170
16. Compression Ratio	6.8	6.65	....	7.0	6.5
17. Take-off Piston Speed	2625	3000	2850	2690	2600
18. Total Piston Head Area, sq in.	306	285.5	275	328	328
19. Take-off Horsepower per Square Inch Piston Area	3.84	3.65	3.81	3.36	3.36
20. Take-off Horsepower per Cubic Inch Displacement per Minute	0.000111	0.000101	0.000111	0.000104	0.000107
21. Dry Weight, lb	1367	1325	1394	1085	1356
22. Unit Weight, lb per take-off Horsepower	1.19	1.27	1.33	0.995	1.23
23. Height, in.	40.5	42.1	41.1	37.2	41.7
24. Width, in.	29.1	30.6	29.8	30.1	31.7
25. Overall Length, in.	84.0	94.5	75.1	84.1	68.7

the case of the 12Y-51 probably results from the characteristic clean-cut cylinder block with wet sleeves, and single direct-acting valve gear operating only two valves per cylinder, so long identified with Hispano-Suiza design practice.

(g) Likewise the dimensional characteristics of the various engines in Items 23, 24, 25 are sufficiently comparable as to require no special comment.

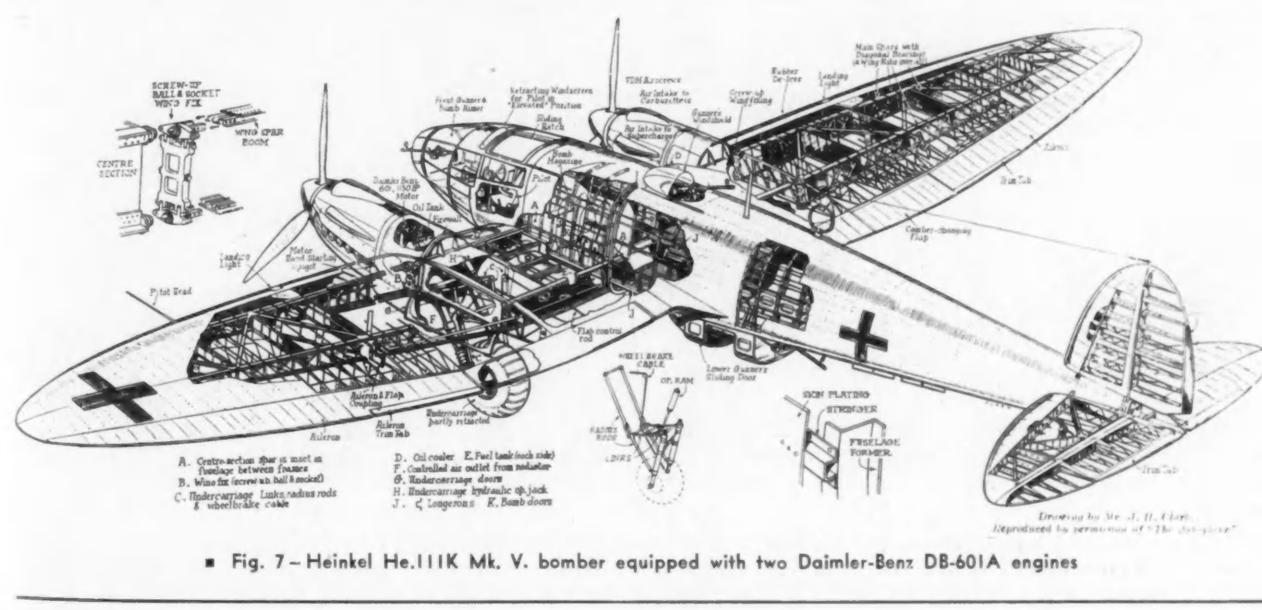
■ Design Features, Material, and Quality

Proceeding next with a teardown inspection of the engine, a detail description with comments on some of the design features of the major component parts of the DB-601A will be undertaken followed by a check by the magnaflux inspection method; analysis and comparison of materials; and comparison on quality of finish.

■ Details of Construction

Crankcase – Cast of aluminum alloy, the crankcase of the DB-601A is of sturdy construction, of deep section and well-ribbed for strength. As shown in Fig. 8 the main bearing supports are carried in heavily webbed sections with wide flanges. A through tie-bolt with an extremely highly polished finish and secured by nuts inside and outside the crankcase, passes through the main bearing cap and stiffens the entire section, preventing localized scuffing. Four stepped and mirror-finished studs secure each bearing cap. As in the case of the connecting-rod nuts, a finely serrated head is used with the addition of a spherical-radius seating washer. On the main bearing cap adjacent to the crankshaft gear, the mutilation from a machine-gun bullet may be observed.

Generous oil drain passages are provided adjacent to the



■ Fig. 7 – Heinkel He.111K Mk. V. bomber equipped with two Daimler-Benz DB-601A engines

cylinder barrels through the main bearing webs to permit the flow of oil to the scavenge pump. The parting flange without gasket for the top cover plate is fitted with a multiplicity of studs in two sizes, approximately 5/16 and ⅛ in. Between each group of studs, consisting of one large size flanked by two small studs, are laid two horizontal dowel pins of approximately 3/16-in. diameter by ¾-in. length. The necessity for this complicated construction is not fully appreciated from our American production standards. In other respects, the crankcase is a good production set-up with flat machining surfaces and accessibility for boring operations. There is, to be sure, considerable evidence of hand work in the polishing of the highly stressed ribs and fillets which is an essential precaution where cast material is employed.

In Fig. 9 is illustrated the forged and anodized aluminum-alloy main bearing cap which was mutilated by a machine-gun bullet. It will be observed that the cap bears in the crankcase support on a cylindrical surface. It is located by means of two dowel pins approximately 5/16 in. in diameter by ⅞ in. in length, inserted for longitudinal shear. A heavy steel shell lined with lead-bronze is secured to the cap by two countersunk steel screws staked in place. At the parting surface of the steel bearing shells are embedded four horizontal dowel pins approximately 3/32-in. diameter by ¾ in. in length.

Lubricating oil from the crankcase pressure passage in the main bearing supports enters the cap through a hole in the cylindrical locating surface. Thence it passes to an annulus on the back side of the steel bearing shells and is distributed through three holes and a connecting groove in each half shell to the lead-bronze bearing surface.

Crankshaft – Magnafluxed all over, the crankshaft gave no indications of any imperfections in the steel. The finish is excellent and comparable to the best foreign and American aircraft engine practice. As indicated in Fig. 10 the design is a conventional seven-main-bearing six-throw type weighing 168 lb. The hardened crankpin journals are 3.361 in. in diameter and 2.875 in. in length. Main journals are 3.936 in. in diameter and 2.125 in. in length. Eight counterweights are attached to the cheek extensions by eight rivets and are well finished.

Crankpins and main journals are hollow. A steel tube is spun at each end into a circumferential recess in the crankpin to form an annulus reservoir for oil throughout the entire length of the crankpin. This construction offers no means of removing the sludge other than boring out the shell and making a complete replacement. A steel tube from the adjacent main bearing journal feeds oil to each crankpin reservoir. The oil hole in the main journal is approximately ⅜ in. in diameter with well-rounded edges. A groove leading in the direction of rotation, extends for ¾ in. from the oil hole. This groove is well blended and highly polished to prevent stress concentration through scratches or sharp corners at this vital point. Distribution of oil to the connecting-rod bearings is finally accomplished by four ⅛-in. holes in the crankpin. These holes are spaced approximately ¾ in. apart and centrally located between the roller-bearing tracks. One pair of holes is on the explosion-loaded side of the crankpin while the other pair is disposed by something over 90 deg to the more lightly loaded portion of the crankpin.

The forward end of the shaft is splined to receive the driving sleeve which, in turn, is splined into the crankshaft pinion to provide some degree of flexibility. A starter jaw is attached to the rear end of the shaft together with the accessory drive gears.

Cylinder Block – The cylinder assembly of the DB-601A is of monobloc construction as shown in Fig. 11. The head and jacket form an unusually clean, smooth-wall casting with port flanges and cam deck presenting a good production milling set-up. It will be observed that the common inlet port for three cylinders is smoothly hand-polished and the bosses and port walls streamlined. The fuel injection nozzles fit into adapters, screwed in at a slight angle and slotted properly to direct the spray discharge. Bronze valve guides are used with a measured clearance of 0.0025 in. on the inlet and 0.0036 in. on the exhaust. Dry cylinder sleeve construction is employed with steel liners screwed into the head. The attachment of the cylinder block to the crankcase is accomplished by a means quite foreign to American practice. The open end of each cylinder sleeve is threaded externally at the section which projects into the crankcase beyond the deck flange. A locking ring nut, Fig. 12 and washer clamp the block with its gasket against the crankcase deck. Tightening of the ring nut is effected through the gear teeth on its periphery which are engaged by a pinion wrench piloting in a hole in the crankcase deck section. Provision is made for the application of two wrenches to each sleeve. A considerable mechanical advantage is obtained by this tightening method and no locking means are apparently necessary on the cylinder hold-down ring nut. The cylinder sleeve in the DB-601A thus carries the explosion loading which in other well-known designs is assumed by long tie-bolts through the block or short hold-down studs through the cylinder jacket flange. Fig. 13 shows a view of the open end of the cylinders. The barrels have an excellent degree of finish as is indicated in the photograph. Bronze valve seat inserts are screwed into the aluminum head by means of the diametrically opposite serrations on the inner surface of the insert. Two spark-plug ports are located side by side on the exhaust side of the head and opposite to the fuel injection port. While this design facilitates greatly the installation and removal of spark-plugs, it is perhaps not the optimum location from the standpoint of proper combustion. The flat-top combustion chamber is also a compromise to obtain simplicity in the valve gear and the use of dual valves. Our own school of design favors the hemispherical combustion chamber or as near an approach thereto as possible for engines of high output. In the case of the DB-601A, however, the compromise has apparently not been too serious for the current rating of the engine, provided the fuel quality is high, as will be discussed later.

Camshaft and Valve Gear – See Fig. 14. The single camshaft in each cylinder block is unique in that one cam with a lift of 0.339 in. operates in proper sequence one inlet and one exhaust valve, a function made possible by proper disposition of the rocker rollers for each valve. Two cams per cylinder are, of course, required to actuate the dual valve system. Light in wall section, the camshaft has a high degree of surface finish as will be discussed later. The seven camshaft bearing supports are made of cast magnesium secured to the cylinder head by two studs and a dowel. These bearing supports are split on the centerline with a finely serrated joint and the camshaft journal runs directly in the magnesium. From an external line oil is fed to the hollow camshaft, thence through a 3/32-in. hole to each journal. On the leading flat adjacent to the base circle of each cam, oil is supplied through a 3/64-in.

hole. It is noted that these camshaft oil feed holes have very slightly broken edges which is quite contrary to the practice in evidence elsewhere in this engine. A pump, driven from the camshaft gear, scavenges through a screen and pipe both ends of each cam cover housing which forms a sump.

A steel forked support secured to the cylinder head by two studs carries the common rocker pin. The exhaust rocker straddles the inlet rocker on this common pin. Bronze bushings take both rocker loads. As previously mentioned, both inlet and exhaust rocker cam followers contact the single cam and accordingly are located between the valve tappet and the rocker pin. This arrangement results in the same valve opening in crankshaft degrees for both inlet and exhaust functions with equal valve clearances. In actual operation, however, the exhaust tappet clearance of 0.020 in. compared to 0.010 in. on the inlet tappet, reduces the exhaust opening as measured, by 21 deg. Valve clearance adjustment is obtained by a conventional lock nut on the tappet which screws into the rocker arm. The valve itself is actuated by a flatted ball secured in a socket formed at the end of the tappet screw.

Valves and Springs – Fig. 15 illustrates the inlet and exhaust valve, springs, locks, and washers. The valves are of conventional design with 45-deg seats and stem diameter of 0.549 in. Double right-hand-wound springs with a slight taper are used interchangeably on inlet and exhaust locations. The heavy spring has a loading of 55 lb and the light spring has a loading of 34 lb in the valve-open position. The washer on the stem end of the valve has a taper seat for the split locks which engage the groove in the stem. An appreciable amount of scoring is in evidence in the valve-stem locking grooves. Fig. 16 shows the hollow-stem construction in a sectioned view of both inlet and exhaust valves. The inlet valve, on the left, with an overall head diameter of 2.24 in., contains no internal cooling medium. The stem is drilled from the head end, somewhat eccentrically on the valve examined, fitted with a plug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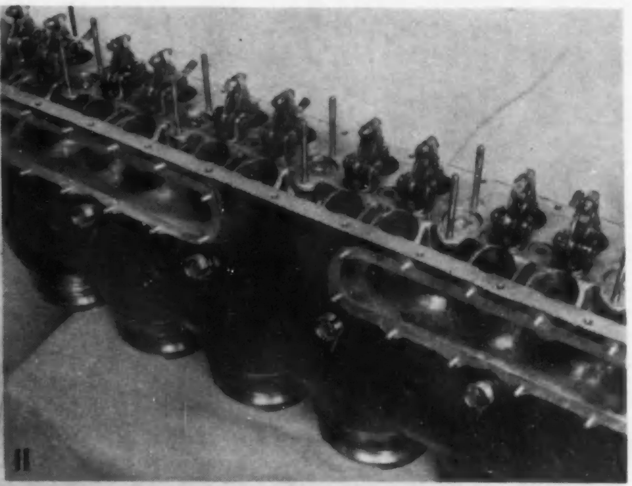
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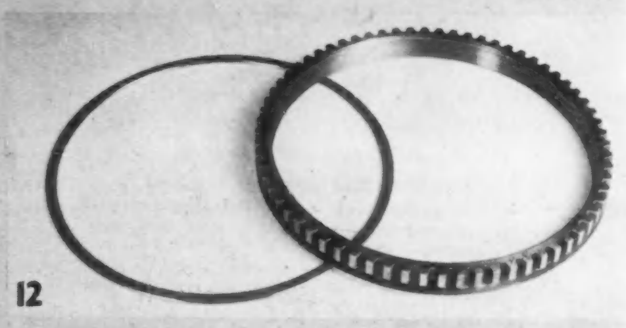
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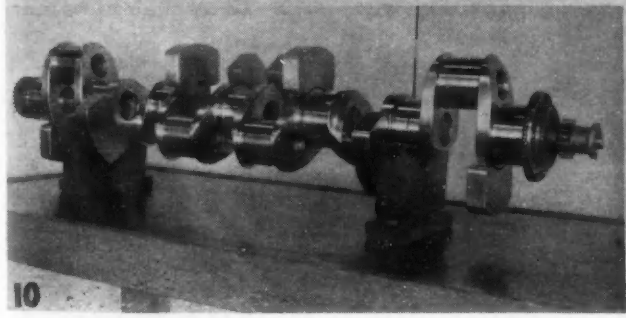
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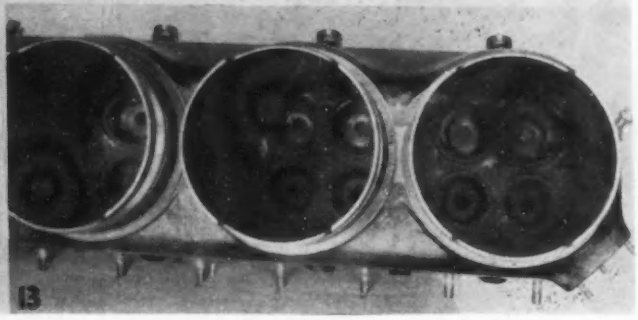
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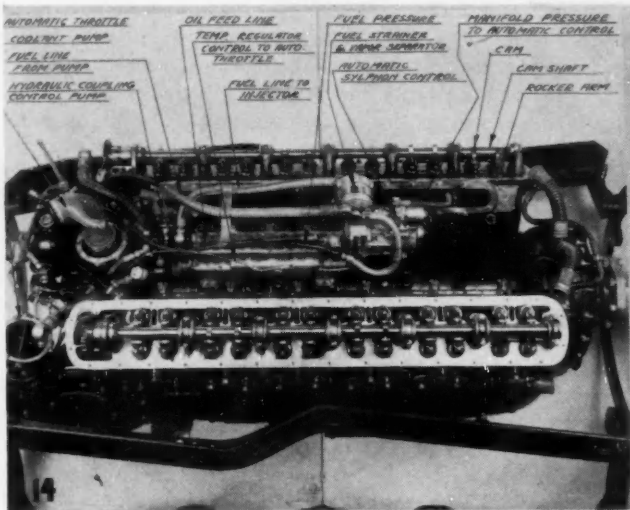
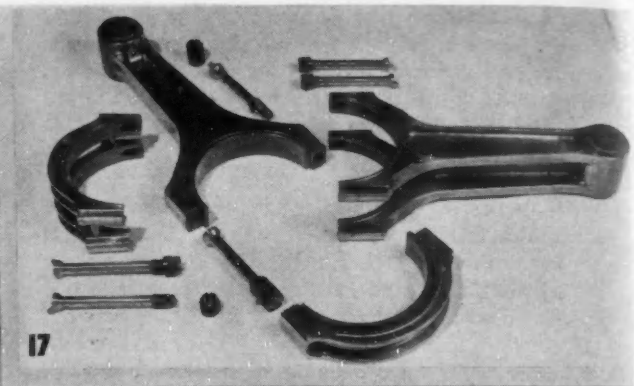
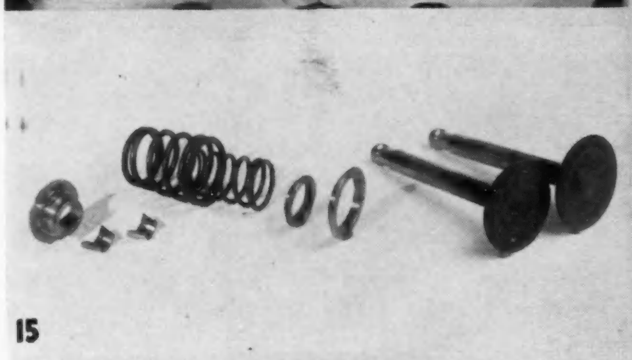
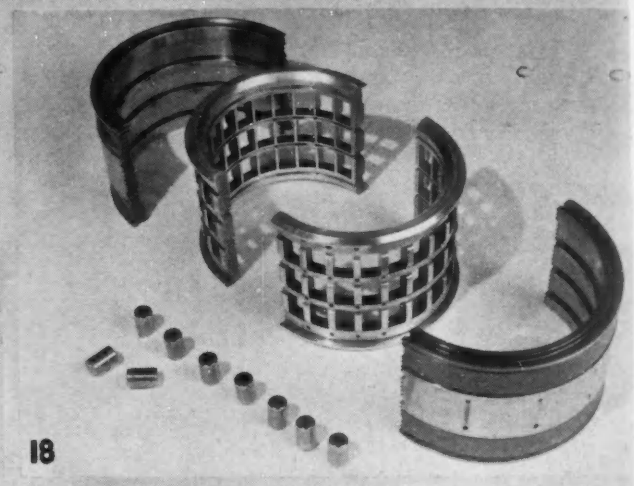
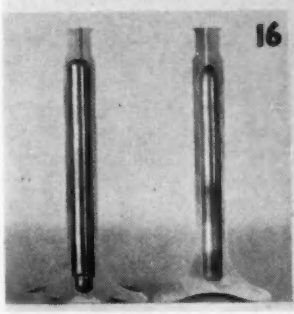
- Fig. 8 – Crankcase
- Fig. 9 – Main bearing cap mutilated by machine-gun bullet
- Fig. 10 – Crankshaft
- Fig. 11 – Cylinder assembly
- Fig. 12 – Locking-ring nut and washer employed to clamp the cylinder block with its gasket against the crankcase deck
- Fig. 13 – Open end of cylinders

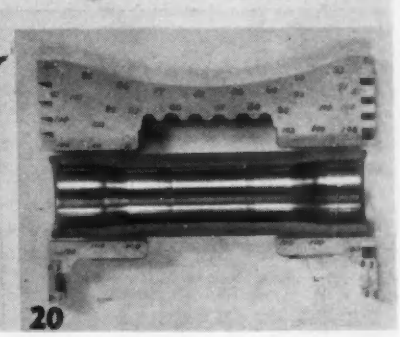












- Fig. 14 – Camshaft and valve gear
- Fig. 15 – Inlet and exhaust valve, springs, locks, and washers
- Fig. 16 – Sections of inlet valve (left) and exhaust valve
- Fig. 17 – Connecting rods
- Fig. 18 – Connecting-rod roller bearing
- Fig. 19 – Piston, rings, and piston pin
- Fig. 20 – Section of piston and piston pin showing Brinell hardness of aluminum alloy

and sealed by welding a button on the head end. This construction is reminiscent of early designs of internally cooled valves used in this country. The bore of the stem is smoothly reamed. The hardness of the valve tip is 53/54 Rockwell C and that of the stem 270/274 Vickers (25/27 Rockwell C).

The exhaust valve shown on the right is somewhat smaller than the inlet valve with an overall head diameter of 2.0 in. The hollow stem is smoothly finished inside and outside. Sodium is used as a cooling medium and occupies 57% of the cavity volume. The valve seat is faced with stellite which is not carried around the radius to the head surface. The valve tip hardness is 58/60 Rockwell C and the stem hardness is 230/233 Vickers (19/20 Rockwell C). While the design of these valves may be satisfactory for their size and particular operating conditions, our American valve designers with their skilled manufacturing technique are producing valves for high-output engines which, in the writer's opinion, are superior to the German product. Further data on the DB-601A valves will be discussed in connection with the analysis of materials.

Connecting Rods – The connecting rods, as shown in Fig. 17, are of the fork and blade type split at the bearing centerline with a vee-serrated joint. The blade rod has a conventional I-section shank and a cap reinforced with two stiffening ribs. Two mirror-finished clamp bolts with keyed heads and long nuts with vee splines instead of hexagonal heads secure the two halves of the blade rod. A lead-bronze lined steel shell is fastened into each half of the blade rod with two countersunk brass screws. The blade rod bears on the outer surface of a steel shell split with vee-serrated joint and fitted into the forked section of the mating rod. In accordance with conventional foreign practice the forked rod design incorporates a roller bearing, Fig. 18, in con-

act with the hardened crankpin. A machined all-over forged duralumin cage for the triple row of rollers fits within the steel shell previously mentioned and is located by means of circumferential flanges. This cage is also split with a vee-serrated joint. The serrated parting line of the roller-bearing housing and cage is rotated approximately 15 deg from the serrated parting line of the forked rod. The 72 rollers seem to run over the serrated joint with no apparent annoyance due undoubtedly to the slight relief at this point. The forked rod has a shank of H-section with reinforcing ribs along each edge of the flanges and blending into the forks and piston-pin eye. The cap with four stiffening ribs is secured through a vee-serrated joint to the forks by four mirror-finished bolts with keyed heads. All connecting rod bolts have copper-plated nuts and threads, undoubtedly to prevent scuffing and potential fatigue failure. The bolt heads bear on a spherical-radius seat in the rods. Surfaces in contact with the bearing shells of both forked and blade rods are shot-blasted to surface-harden the material and prevent fretting. This hardening finish is also given to the mating surfaces of the bearing shells in contact with the rods.

Lubrication to the piston-pin bushing of the blade rod is provided through three holes at the eye, that is, one hole in each flange and one hole at the junction of the web and eye section. On the piston-pin bushing surface the oil is distributed by a circumferential channel centrally located and by two diametrically opposed grooves extending to within 3/16 in. of the bushing edge. In the case of the forked rod piston-pin bushing, two holes through the web junction with the eye supply oil escaping from the connecting-rod bearings. Lubrication of the big-end connecting-rod bearings is accomplished from the supply in the hollow crankpin. Two ⅛-in. holes on the loaded side of the crankpin and two similar holes on the unloaded area located centrally between the roller tracks feed oil to the roller bearings. Through a multiplicity of 1/16 in. holes in the ribs of the duralumin roller cage the oil is fed to the steel shell which in turn, transfers the lubricant through additional holes with carefully michelled grooves to the bearing surface of the blade rod. The lubricating method of the connecting-rod system has been worked out with typical German thoroughness. Its effectiveness is reflected in the excellent condition of all bearing surfaces. The lead-bronze bearings which are characteristically susceptible to scoring show only a relatively few minor circumferential scratches. It is worthy of mention that the blade rod bearing shows evidence of a flash plating of lead to facilitate its seating in the initial stages of operation.

The connecting rods are extremely rugged with an excellent degree of finish in the order of 8.5 micro-in. which is comparable to the best American practice. Careful attention has been paid to the elimination of sharp corners and incorporation of generous fillets to reduce stress concentration. Similarly, all identification marks and symbols are produced with an acid etching process and, in several instances, an electric needle has been used lightly to number parts at assembly. No steel punch letters or figures are in evidence on highly stressed parts.

Piston, Rings, and Piston Pin – The forged aluminum-alloy piston is of conventional design as shown in Fig. 19. Seven light transverse ribs stiffen the inside of the head which on the combustion side is slightly concave. The ring arrangement consists of three ⅛-in. compression rings with staggered left and right-hand 45-deg gaps and one 3/16-in. slotted oil-control ring above the piston pin; and one 3/16-in. slotted oil-control ring below the piston pin. Both oil-control rings have square gaps. All rings have parallel sides. Piston-ring grooves have quite sharp corners at the bottom and, for the oil-control rings, twelve ⅛-in. holes at the groove bottom return the oil to the inside of the piston. On the anti-thrust side of the piston at the oil-control ring adjacent to the compression rings, six 1/16-in. drain holes are drilled on a 60-deg chamfer adjacent to the skirt. At the piston-pin holes a slight relief is machined on the skirt through a subtended angle of approximately 90 deg.

The unusual feature of the piston is the means adopted to insure an adequate supply of lubricant to the piston-pin bearing surfaces in each boss which, in the case of an inverted engine, would seem to be almost self-sufficient. Each boss, however, has two 3/16-in. holes drilled to communicate with longitudinal grooves in the piston-pin bearing area and extending to within ⅛ in. of each end of this surface. Undoubtedly this precaution on piston-pin lubrication ties in with the service operation requirements which will be discussed later. The piston pin itself shows no discoloration from heat and in finish and design is comparable to the best American practice. Fig. 20 shows a sectioned pin which illustrates the straight and tapered wall sections. Machined dural plugs in each end with a spherical radius on the head slightly less than that of the cylinder bore permit the pin to float. These plugs are long for stabilization and fit snugly in both bores of the hollow piston pin.

Measurements of the piston show that the wide land adjacent to the combustion chamber has a diametral relief of 0.016 in. The minimum diametral clearance between the piston skirt and the cylinder wall is 0.017 in. Side clearances of the piston rings in the grooves were measured and from the head end of the piston were found to be 0.005, 0.003, 0.0025, 0.0015 and 0.001 in. respectively.

Supercharged Fluid Drive – One of the unique and perhaps the most interesting feature of the DB-601A engine is the supercharger fluid-drive system. Illustrated diagrammatically by Fig. 21 the system incorporates the following elements:

(a) Supercharger impeller and casing, *S*.

(b) Supercharger impeller drive shaft, *I*, integral with fluid coupling housing, *H*, and drain hole, *d*.

(c) Driving rotor *R* splined to hollow driveshaft *D*.

(d) Bevel drive gears *B* from crankshaft system.

(e) Oil supply from filter *F*.

(f) Primary engine-driven pressure pump *P* with safety valve *X*, supplying oil to hollow driveshaft *D*.

(g) Secondary engine-driven pressure pump *p* supplying oil under regulation from balanced bypass valve *V* to hollow driveshaft *D*.

(h) Sealed sylphon *C* with adjustment *A* controlling piston in balanced valve *V* and responsive in action to changes in atmospheric density through air induction scoop.

(i) Bypass line *L* returning oil from secondary pump *p* to engine sump *O* when so controlled by valve *V*.

The major parts in this system are shown by Figs. 22, 23, and 24. In Fig. 22 is illustrated the supercharger in a dismantled condition. The impeller is 10¼ in. in diameter, smoothly machined all over from an aluminum-alloy forging, no doubt, and finished by the anodizing process. The diameter of the entrance buckets is 5⅞ in. and the width

at the tip of the twelve blades is ½ in. A six-spline steel hub of approximately ⅞ in. O.D. with flanged faces is riveted in place similar to accepted practice in this country. There is no evidence of metal removal at the scallops for balancing purposes which indicates close machining tolerances. With a single outlet of 4¼-in. diameter, the scroll-type casing is cast in an aluminum alloy. The 6-in. diameter opening to the suction side of the impeller is protected by a 3/16-in. mesh steel screen. A vaneless diffuser is used on this particular engine, undoubtedly to reduce the overall diameter of the blower unit, with the long inlet manifold balanced between cylinder blocks serving to convert velocity of the induction air into pressure at the inlet ports. Two roller bearings support the supercharger impeller, its driveshaft and the fluid coupling housing shown in Fig. 23 in the upper left-hand corner. In the center of this figure is located the driving rotor having radial vane buckets on both faces, and which is splined to the hollow driveshaft integral with bevel drive gear (the lower right-hand corner). It will be noted that this shaft has a number of small oil feed holes which supply the driving fluid to each side of the rotor, thus eliminating end thrust from this source in the fluid-coupling unit. In the center foreground of Fig. 23 is the cover plate to the driven housing on the impeller shaft also fitted with radial vane buckets; and directly above is the bearing support housing.

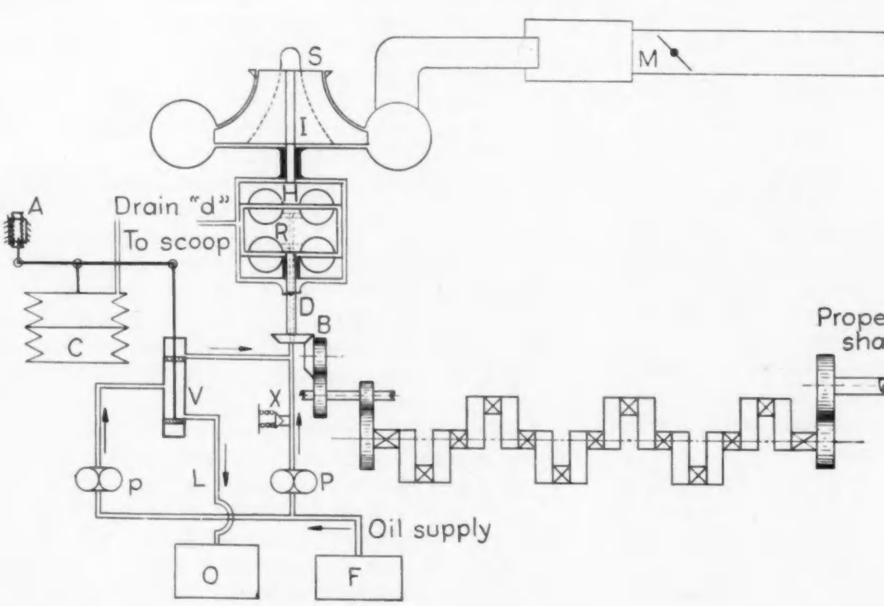
Running at approximately ten times crankshaft speed, the fluid-coupling unit is quite small with an overall diameter of 4¾ in. The radial vane buckets in both rotor and driven casing are aluminum castings riveted to their respective steel members. It is also noted that the rotor as well as the driven casing are discolored by heat which, it is believed, was caused by a shrink fit between the steel shell and the aluminum bucket casting. A slight amount of sludge was observed in the buckets and one oil feed hole was plugged with sludge, which indicates that the engine had relatively little service operating time or that the sludge is eliminated effectively through the bleed holes.

■ Fig. 22 – Dismantled supercharger of supercharger fluid drive system

Since the coupling itself makes an excellent centrifuge, the sludging possibility may well become a serious service problem on this type of supercharger drive. The clearance between the rotor and the driven housing was measured diametrically as 0.072 in. minimum. The axial clearance between rotor and housing is 0.102 in. In Fig. 24 is illustrated the drive assembly and housing which contains both primary and secondary pumps supplying oil to the fluid coupling. In conjunction with this unit is located also the driving flange for the fuel injection pump. The control sylphon for the secondary pump discharge is located at the top of the housing.

Returning to Fig. 21, the supercharger fluid drive and control system appears to function as follows:

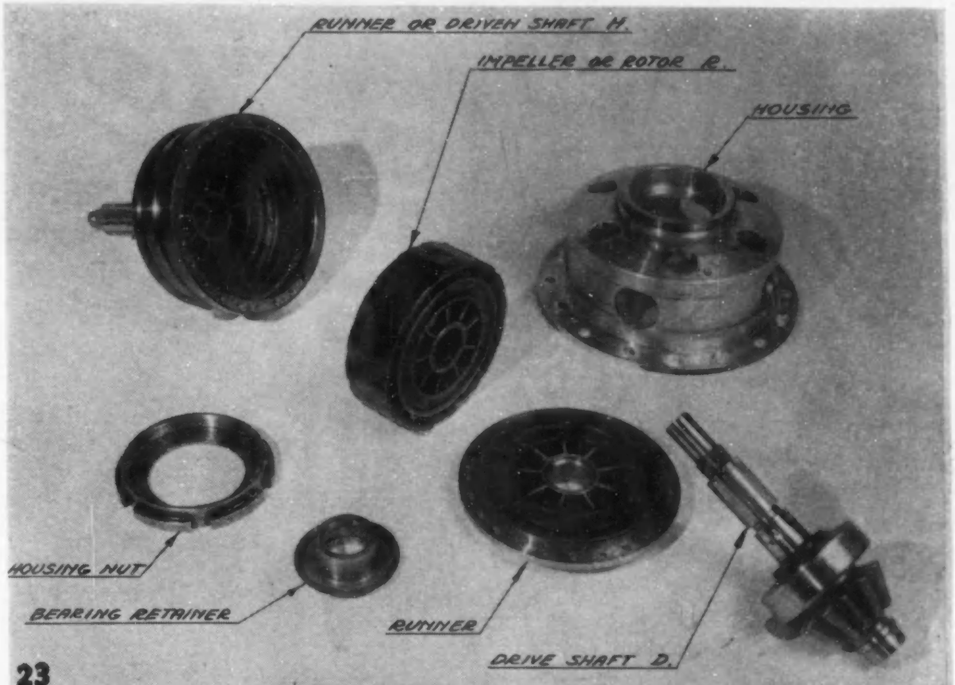
(a) Under sea-level operating conditions the fluid coupling is only partially filled with oil from the primary pump, *P*. The discharge from the secondary pump *p* is bypassed to the engine sump through valve *V*. With the resultant slip between rotor *R* and housing *H*, the net



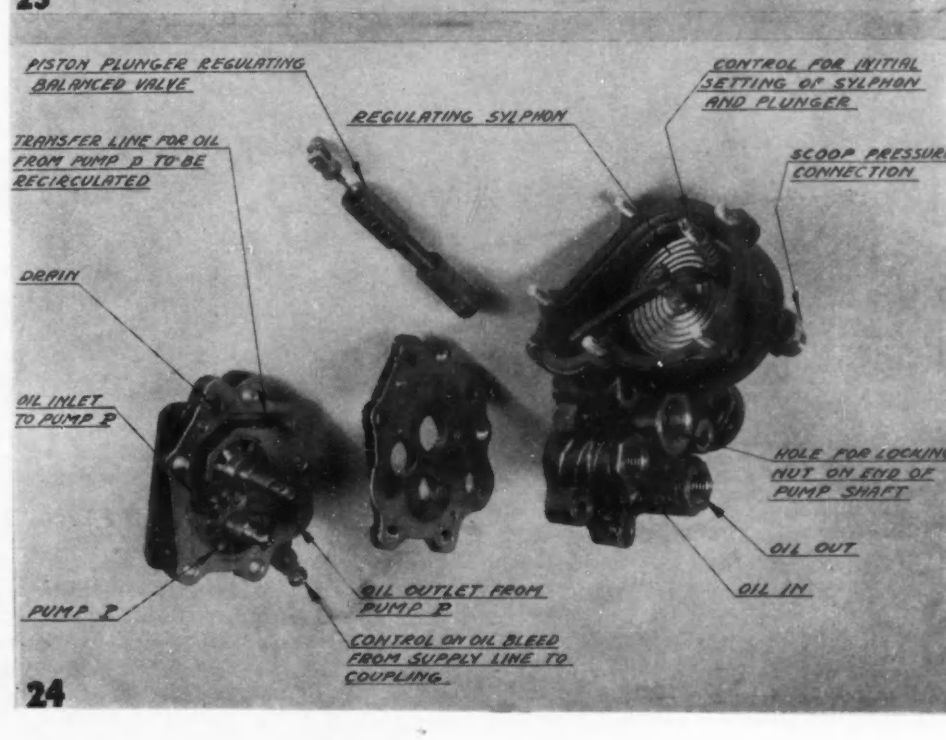
■ Fig. 21 – Supercharger fluid drive

effective driving ratio between crankshaft and supercharger impeller is of the order of 7:1. For take-off and climb this ratio provides ample manifold pressure as will be noted later in a discussion of the performance characteristics of the fluid drive.

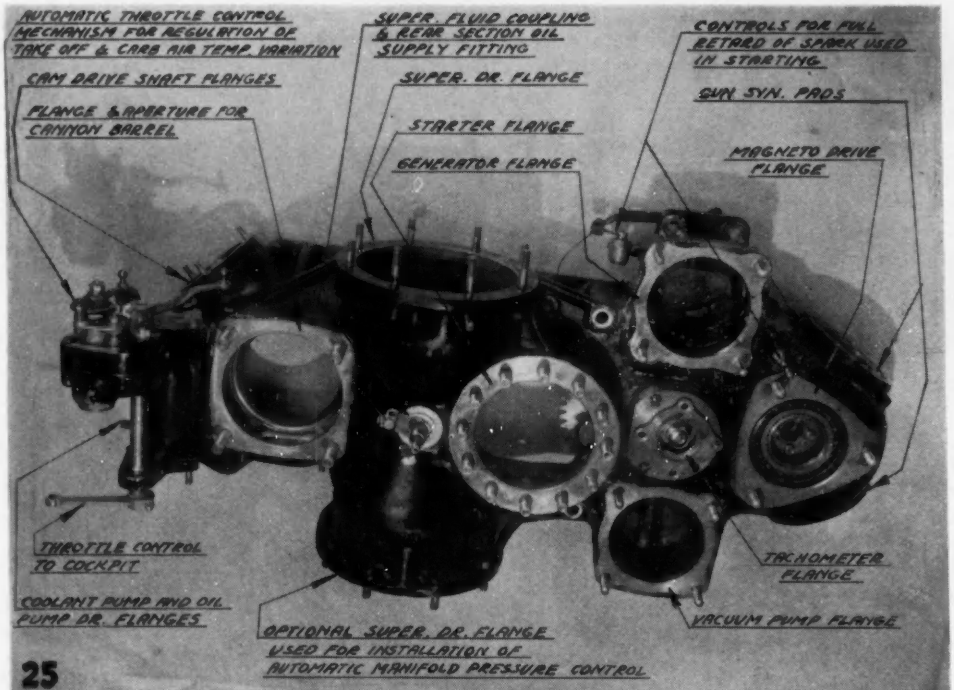
(b) With the discharge oil from the secondary pump p controlled through the bypass valve V, either to the hollow driveshaft D or to the sump, the degree of slip of the coupling and accordingly the net effective drive ratio of the supercharger impeller, is subject to the functioning of the sylphon control C which, in turn, is responsive to changes in altitude. With increasing altitude the flow is stepped up, the slip in the coupling decreases and the net effective ratio increases until, at approximately 12,000 ft the coupling is receiving the maximum oil delivery from secondary pump p as well as from pressure pump P. At this altitude the coupling is virtually locked hydraulically with a slip of but 2% and the effective supercharger drive ratio is 10.2. Further discussion of the fluid-drive characteristics based on test will be given later. With further reference to the sludging problem of the fluid-drive mechanism, careful examination of the control valve V disclosed that the travel of the piston was limited by a deposit of sludge which directly restricted the output of the supercharger and could not be overcome by the action of the sylphon.



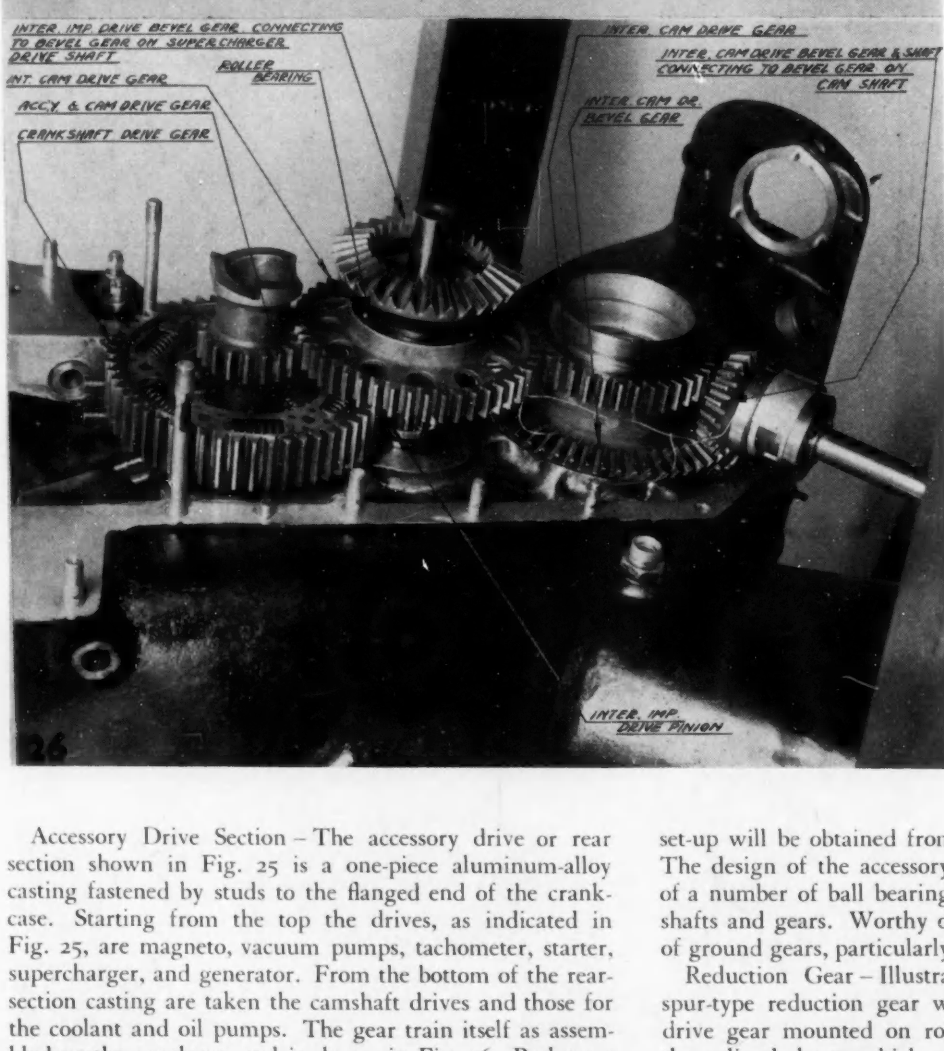
■ Fig. 23 – Supercharger hydraulic coupling



■ Fig. 24 – Supercharger hydraulic coupling oil-supply regulating pump



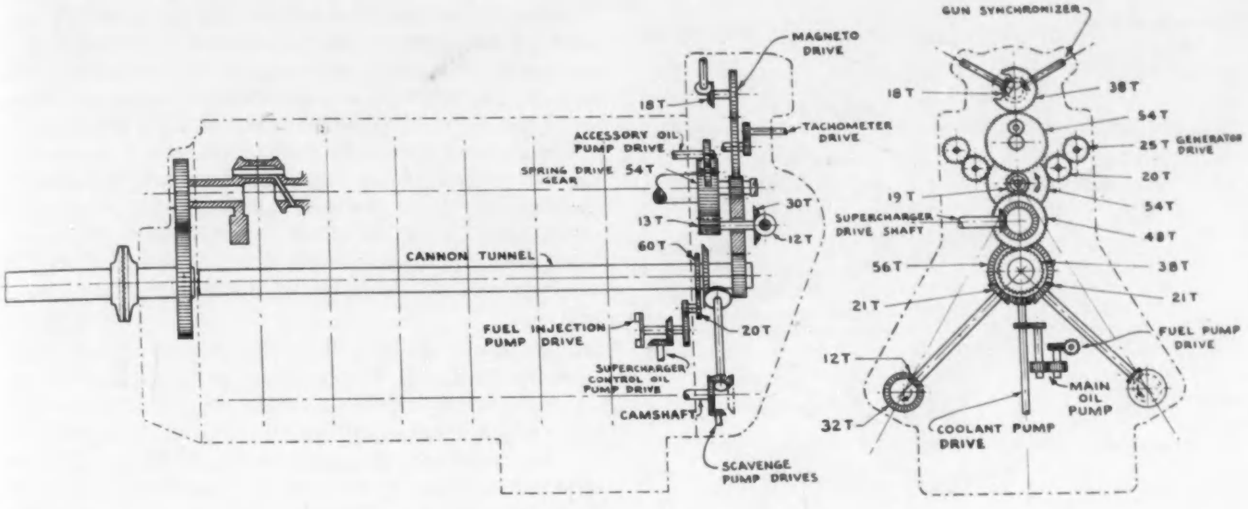
■ Fig. 25 – Accessory drive or rear section



■ Fig. 26 – Rear section gear train

Accessory Drive Section – The accessory drive or rear section shown in Fig. 25 is a one-piece aluminum-alloy casting fastened by studs to the flanged end of the crankcase. Starting from the top the drives, as indicated in Fig. 25, are magneto, vacuum pumps, tachometer, starter, supercharger, and generator. From the bottom of the rear-section casting are taken the camshaft drives and those for the coolant and oil pumps. The gear train itself as assembled on the crankcase end is shown in Fig. 26. Perhaps a better idea of the complete accessory drive and gear-train set-up will be obtained from the sketch shown in Fig. 27. The design of the accessory drive section features the use of a number of ball bearings to support the various driveshafts and gears. Worthy of note is the excellent example of ground gears, particularly of the bevel type.

Reduction Gear – Illustrated in Fig. 28 is the single spur-type reduction gear with both pinion and propeller drive gear mounted on roller bearings. On the right is the splined sleeve which connects the pinion and crankshaft. With a face of 2⅝ in., both gears are ground and



■ Fig. 27 – Front and rear section gearing

all edges are carefully rounded. The cast-aluminum gear case is internally ribbed for strength. An oil feed line, ending in a jet with two 1/16-in. holes, squirts oil on the disengaging side of the gears. On the extreme right side of the case will be observed the flange with Hirth-type serrated joint which secures the reduction gear and propeller shafts.

Induction System – As previously mentioned, the single outlet from the supercharger scroll discharges into the induction manifold which is balanced between cylinder blocks by virtue of its continuous loop construction. Induction air to the cylinders is regulated by two throttles:

(a) A pilot-controlled butterfly throttle adjacent to the loop manifold, and subject to the regulatory timing action of a clockwork mechanism for the wide-open position, governs the final admission of air to the inlet ports.

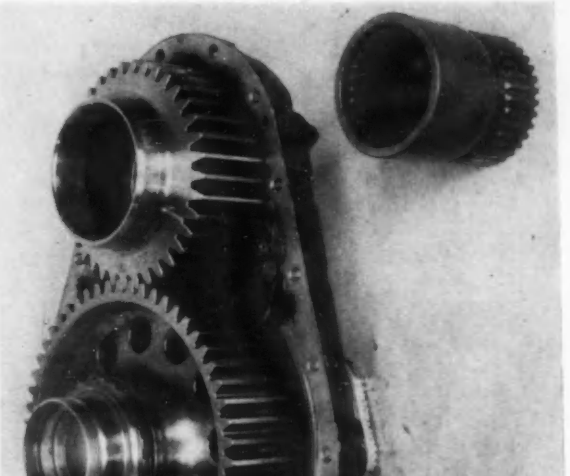
(b) An auxiliary butterfly valve located at the outlet of the supercharger and responsive to a boost control regulator maintains a constant pressure of 37.5 in. hg ahead of throttle (a) during part-throttle operation and somewhat increased pressure for full-open conditions.

Due to their streamlined design and aerodynamic balance, the throttle loading is considerably lower than that of the conventional carburetors.

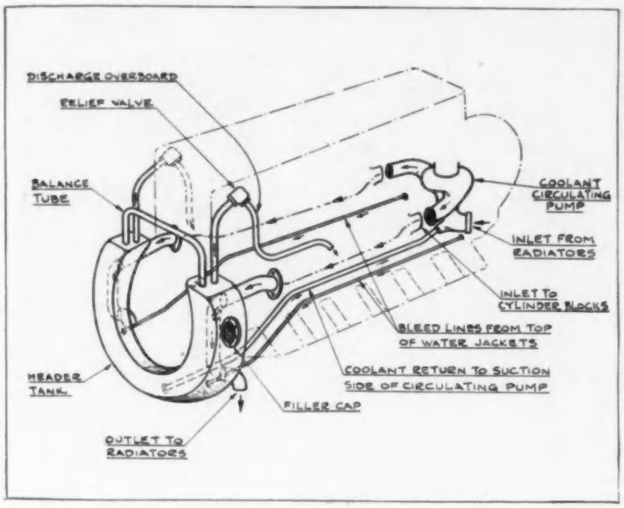
Cooling System – The cooling medium commonly used in the DB-601A is a mixture composed of 70% water and 30% ethylene glycol. In the case of fighters for high-altitude operation it has been reported that the ethylene glycol content is increased to 50%. The ethylene glycol is used merely as an anti-freeze and, so far as is known, the operating temperature limits are those of a water-cooled engine.

The cooling system is diagrammatically represented in Fig. 29. The circulating pump is a centrifugal type with double outlet, driven from the bottom of the accessory section and located in the cylinder vee between the camshaft drive shafts. The impeller of cast alloy with 12 blades, is riveted to the splined driveshaft which is supported on a ball bearing. The aluminum-alloy casting which houses the driveshaft and packing gland to form the rear end of the pump also embodies the oil pressure pump with integral strainer compartment. Due to its inaccessibility, the coolant pump packing gland is adjusted by means of a steel worm wheel meshing with a bronze wheel nut.

From the tunnel radiators on the bottom of the fuselage the coolant enters the pump and is discharged into the end of each cylinder block at a point adjacent to the rear inlet port. After circulating around the cylinder heads and



■ Fig. 28 – Spur-type reduction gear



■ Fig. 29 – Cooling system

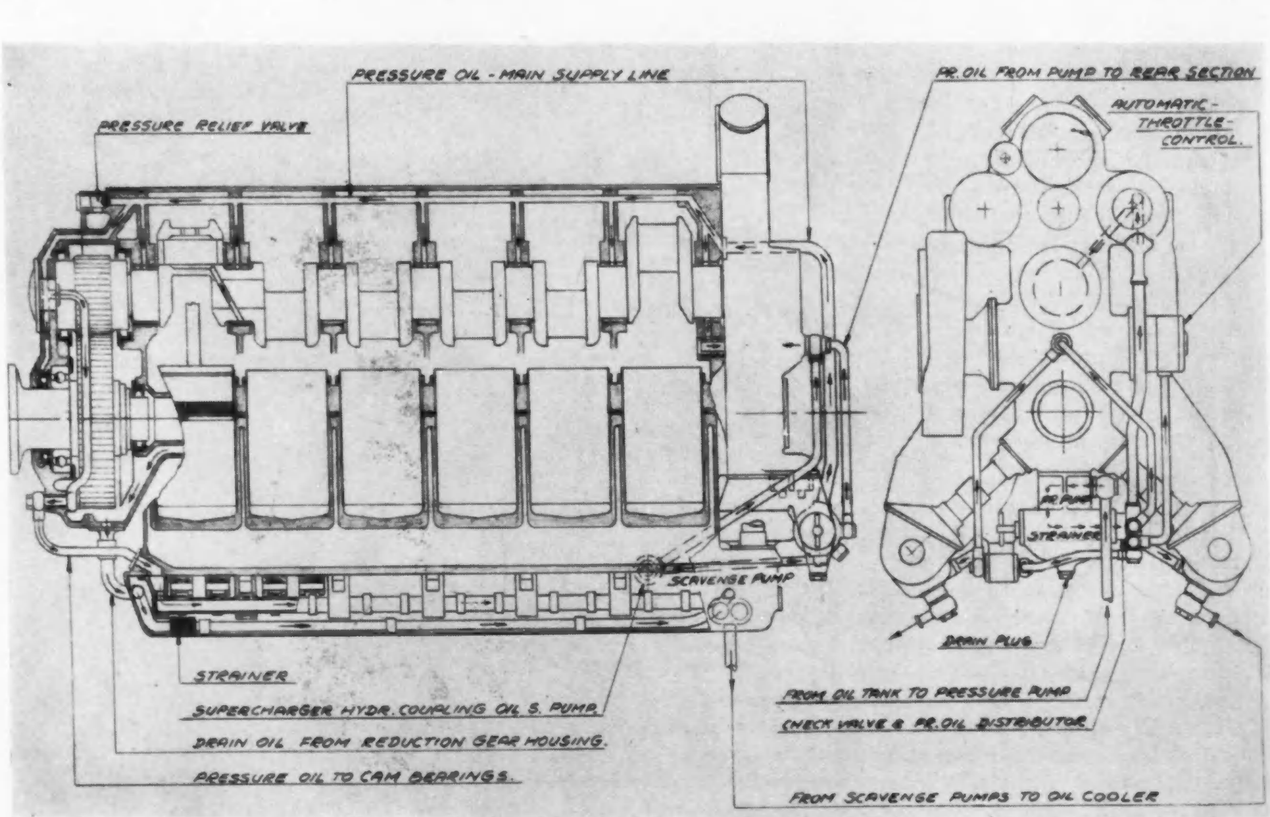
through the cylinder jackets, the coolant is then discharged from the front end of each cylinder block at a point adjacent to the crankcase deck into two pipes which enter the header tank. It will be recalled from the front-view photograph of the engine that the header tank is a horseshoe-shaped body, constructed of sheet aluminum, and attached by supports to the reduction gear housing. The two discharge coolant pipes curve through the header tank and emerge at the bottom where connections are made for the return of the coolant to the radiator. Along the inside radius of the two coolant pipes within the header tank are several small holes for the escape of vapor and air. The top ends of the horseshoe header tank are connected by a small balancing tube. There are also located at these points two lines which, after connection to relief valves mounted on each side of the crankcase, discharge overboard. The setting on the relief valves was checked at 5½ psi.

The coolant in the bottom of the header tank is returned through a 1-in. line passing along the intake manifold to the suction side of the coolant pump. To prevent the formation of vapor pockets at the upper rear end of each block a bleed line is conducted to the header tank. A filler cap is located on the upper left-hand part of the header tank.

Lubrication System – The lubrication system of the DB-601A engine is diagrammatically illustrated in Fig. 30. Oil from the supply tank enters the gear-type pressure pump which is located at the bottom of the rear section in a body integral with the gland section of the coolant pump as previously described. The oil is delivered from the pump to a cylindrical chamber with coiled wire strainer thence to several external pressure lines:

1. The main oil supply to the crankshaft journals.
2. Oil supply to the supercharger fluid drive pumps and coupling.
3. Oil supply to the rear section.
4. Oil supply to automatic controls.

The main oil supply passes from the external line through an internal passage for the length of the crankcase. Individual drilled leads are taken off at each main bearing journal for lubrication of the main bearings and connecting rods as previously described.



■ Fig. 30 – Lubrication system

At the front end of the main oil supply line is located a ball-type pressure relief valve which can be adjusted by the removal of a cap. Lubrication of the reduction gear is accomplished by a continuation of the main supply line with a branch terminating in a spray nozzle. From the reduction gear housing an external oil line delivers pressure oil to the end of each cylinder block for lubrication of the camshafts and valve gear.

Secondary oil pressure lines as indicated in Fig. 30 feed the supercharger fluid pump; the bushings, gears, and ball bearings of the rear section; and from one passage is supplied the oil for the spark-advance-control plunger. Another line as indicated supplies pressure oil to the automatic throttle control.

Scavenge oil flowing around the projecting cylinder barrels and through the cored holes in the main bearing webs, drains at the front of the engine, along with oil from the reduction gears, into two sumps formed by the cam covers. From the rear of the crankcase the scavenge oil drains via external lines to the same sumps. Accessory section scavenge oil flows to the two sumps through the camshaft driveshaft housings. As previously described, two scavenge pumps, each driven from the rear end of the camshaft, return the oil to the cooler.

The principal characteristic of the DB-601A lubrication system is the multiplicity of external pressure lines which, while inherently difficult to avoid in this type of powerplant, have been designed with unusual care to reduce potential failures either through gun fire or vibration. Flexible impregnated fabric multilayer tubing is used for these pressure lines with fittings rolled on the ends similar to the Weatherhead type. A protective glazed fabric cover surrounds the pressure tube about which is wound piano wire with a pitch of approximately ¼ in. For a final covering a heavy leather boot is stitched around the hose. Scavenge lines of larger diameter are similarly constructed except that they incorporate, in addition to the external wire, an internally wound wire coil to prevent collapse. The protective leather covering is omitted.

Fuel Injection System – A 12-cyl in-line Bosch fuel injection pump is used on the DB-601A. The injection system incorporates fuel-air ratio control.

Automatic Controls – A number of ingenious automatic controls are incorporated in the DB-601A to relieve the pilot as much as possible of the responsibility of making manual adjustments to obtain optimum engine performance; and to prevent abuse of the engine by inexperienced pilots.

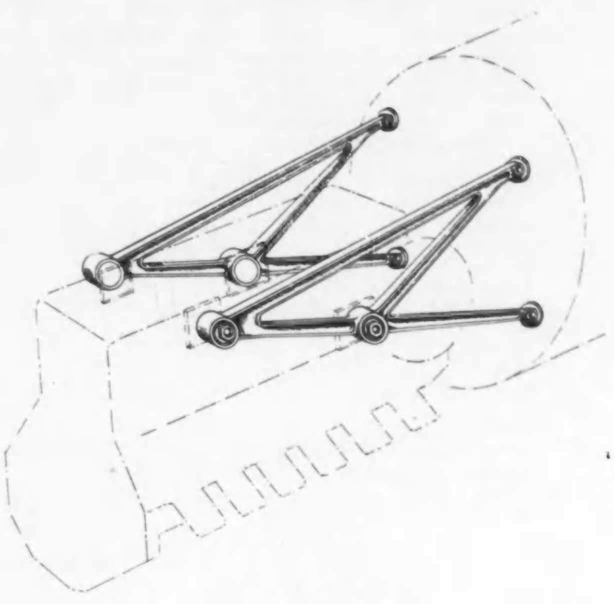
Exhaust System – For nacelle installations in bomber aircraft straight, short exhaust stacks are bolted individually to the three-stud elongated exhaust port flanges. No stacks, however, were received with the DB-601A under discussion. For fighter installations, such as the Me.109, exhaust jet propulsion stacks are utilized.

Ignition System – The complete ignition system of the DB-601A includes a Bosch double magneto, spark plugs, and radio shielding.

Engine Mount – The sketch in Fig. 31 illustrates the method of mounting the DB-601A engine in the Heinkel He.111K airplane. Two modified A-frames in forged duralumin I-section are attached through rubber bushings to the two mounting pads on each side of the crankcase. The legs of the frames terminate at the firewall in fittings secured to the standardized four-point suspension lugs. The engine is accessible yet quickly detachable from the airplane as a complete assembly.

■ Results of Magnaflux Inspection

Inspection of a representative number of highly stressed steel parts by the magnaflux method indicated that the material used in the DB-601A was comparable in standards



■ **Fig. 31 – Method of mounting the DB-601A engine in the Heinkel He.111K airplane**

of acceptance to American aircraft engine quality. The following parts were checked:

1. Blade connecting rods, caps, and bolts.
2. Forked connecting rods, caps, and bolts.
3. Connecting-rod roller-bearing races.
4. Crankcase tie-bolts.
5. Cylinder hold-down ring nuts.
6. Rocker arms and rollers.
7. Crankshaft.
8. Piston pins.
9. Lock-end valve-spring washers.
10. Accessory spring drive gear.
11. Intermediate supercharger drive-gear assembly.
12. Supercharger driveshaft and gear assembly.
13. Camshafts.

In the examination of the camshafts a few very small magnaflux indications were noted on the edges of the oil feed holes on the cam faces and bearing journals. Also several grinding checks were observed on the valve spring washers and a few indications were in evidence on the inside surface of the supercharger driveshaft.

■ Comparison on Quality of Finish

A number of representative DB-601A parts were inspected for surface finish on the Physicist Research Company (rms) profilometer. Table 3 shows the comparison with equivalent Cyclone parts and indicates a close degree of similarity in the standards for quality of finish.

Table 3 – Comparison on Quality of Finish

Part	Surface Finish in Micro-in. DB-601A	Cyclone (Similar Parts)
1. Cylinder Barrel	6	3
2. Crankshaft		
Main Bearing Journal	10	10
Main Bearings (Lead-Bronze)	27	16
Crankpins	7	4
3. Connecting Rods		
Surface Outside Piston-Pin Eye	8.5	16
Surface Outside Crankpin End	20	7
Roller Bearing at I.D. of Outer Race	5	7
Bearing Journal at O.D. of Roller-Bearing Outer Race	7	4
Rollers	7	4
Plain Bearing at Crankpin End (Lead-Bronze)	15	13
Clamp Bolt Body	7	....
4. Crankcase Tie-Bolt Body	15	....
5. Piston Pin		
Center Section	2	4
End Section	6	4
6. Camshaft		
Journals	12	10
Cam (Nose)	13	13
Cam (Heel)	8	13
7. Gears		
Oil Pump (Teeth)	45	15
Spring Accessory Drive (Teeth)	7	20
Intermediate Supercharger Drive Gear (Teeth)	18	20
Journals	7	8

DB-601A and Cyclone Materials

Laboratory analyses were made of a representative number of major parts of the DB-601A engine in order that the chemical composition, hardness, and structure of the material could be compared with similar parts used in the Cyclone engine. These data are outlined herewith as follows:

1. Piston –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 6242)	
Silicon	11.86	Silicon	0.90 max.
Copper	1.02	Copper	3.50 – 4.50
Nickel	1.02	Nickel	1.70 – 2.30
Magnesium	0.77	Magnesium	0.40 – 0.90
Iron	0.57	Iron	1.0 max.
Titanium	0.09	Titanium	0.20 max.
Manganese	0.06	Manganese	None
Aluminum	Remainder	Aluminum	Remainder

Remarks – The DB-601A material is a high-silicon alloy while WAC material is Y-alloy with high copper content which has been used for many years where strength at elevated temperatures is required. Although the grain sizes of the two are about the same, the high silicon content of the DB-601A piston produces a greater amount of silicon constituent in the micro-structure. While the hardness of both materials averages 109 Brinell as processed in a new piston, a check of the DB-601A piston at the center of the dome gave a reading of 69 Brinell. This indicates either considerable service operating time or, more likely, that the operating temperature is high as a result of the low specific fuel consumption.

2. Piston Pin –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 7330)	
Aluminum	0.80	Aluminum	None
Molybdenum	0.38	Molybdenum	None
Chromium	1.56	Chromium	1.25 – 1.75
Manganese	0.47	Manganese	0.30 – 0.60
Carbon	0.38	Carbon	0.38 – 0.45
Nickel	None	Nickel	3.25 – 4.00

Remarks – The DB-601A piston pin is of nitralloy while that of the Cyclone is of a case-hardened steel. The DB-601A pin is nitrided on the O.D. to a depth of 0.012 in. with a resultant hardness of 1097-1132 Vickers. Its core hardness is 40-41 Rockwell C. The German nitralloy material is similar to WAC No. 8190 nitralloy used for cylinder barrels. The Cyclone piston pin has a case hardness of 57-60 and core hardness of 35 Rockwell C.

3. Piston-Pin Bushing –

DB-601A Composition, %		Cyclone Composition, % (AMS 4520)	
Copper	90.82	Copper	86.0 – 88.0
Tin	8.85	Tin	3.5 – 4.5
Phosphorus	0.31	Zinc	3.0 – 5.0
Impurities	0.02	Lead	3.5 – 4.5
		Iron	0.10 max.
		Impurities	0.20 max.

Remarks – The piston-pin bushing of the DB-601A is a wrought bronze which in composition is quite comparable to AMS 4510. It has a Vickers hardness of 129-136 and in structure is comparable to our phosphor bronzes with the exception that it has a larger grain size. The Cyclone piston-pin bushing is of rolled strip stock and has a higher hardness range of 130-185 Vickers.

4. Crankshaft Main Bearing –

DB-601A Composition, %		Cyclone Composition, % (AMS 4820)	
Copper	78.61	Copper	67.0 – 74.0
Lead	21.13	Lead	26.0 – 31.0
Iron	0.26	Iron	0.25 max.
Silver	Trace	Silver	1.50 max.
Tin	None	Tin	0.05 max.
Zinc	None	Zinc	0.10 max.
Phosphorus	None	Phosphorus	0.01 max.
Nickel	None	Nickel	0.01 max.

Remarks – The structure of the DB-601A lead-bronze bearing material resembles that used by WAC in dentritic form and uniform lead dispersion in the copper matrix. It will be noted that the principal chemical constituents are considerably alike, the chief difference being the lack of silver in the German composition. The bond between the bearing lining and the steel back is quite comparable to that of WAC bearings. The hardness of the DB-601A lead-bronze material is 39-40 Vickers while WAC lead bronze is 25-30 Vickers. The steel backing of both bearings is 71-73 Rockwell B.

5. Piston Rings –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 8301E)	
Combined Carbon	0.79	Combined Carbon	0.45 to 0.75
Graphitic Carbon	2.74	Graphitic Carbon	2.70 to 3.20
Silicon	3.25	Silicon	2.60 to 3.10
Manganese	0.48	Manganese	0.40 to 0.80
Phosphorus	0.53	Phosphorus	0.30 to 0.80
Sulfur	0.12	Sulfur	0.10 max.

Remarks – The composition and structure of the cast iron used for the piston rings of the DB-601A is quite comparable to that of the Cyclone piston rings. The Rockwell B hardness of 101-103 of the rings examined will meet the WAC specification.

6. Inlet Valve –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 8160)	
Carbon	0.54	Carbon	0.25 to 0.40
Manganese	0.42	Manganese	0.65 max.
Silicon	2.91	Silicon	2.00 to 3.00
Chromium	9.10	Chromium	11.50 to 14.00
Nickel	Trace	Nickel	7.00 to 9.00
Molybdenum	0.36	Molybdenum	None

Remarks – The intake valve steel in the DB-601A is similar to the silchrome valve steels commonly used in this country. The structure of the material is uniform and it has a hardness of 26-27 Rockwell C. The CNS steel as just given and as used for many years in WAC engines contains nickel and has an austenitic structure with a hardness of 41-44 Rockwell C. This material is considered superior to the ordinary silchrome steels from the standpoint of strength, resistance to wear and fatigue, and ability to withstand corrosion.

7. Exhaust Valve –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 8164)	
Carbon	0.48	Carbon	0.40 to 0.50
Manganese	0.65	Manganese	0.70 max.
Silicon	1.57	Silicon	2.75 to 3.25
Nickel	13.50	Nickel	13.00 to 15.00
Chromium	15.30	Chromium	13.00 to 15.00
Tungsten	3.95	Tungsten	1.75 to 3.00
Molybdenum	None	Molybdenum	0.20 to 0.50

Remarks – The DB-601A exhaust-valve material has approximately the same hardness, 230-233 Vickers as the steel used in the exhaust valve of the Cyclone. In the case of the Cyclone exhaust valve, however, the stem is nitrided to produce an extremely hard wearing surface of the order of 725 Vickers. It will be noted that the German steel has a slightly higher chromium content and contains no molybdenum. The general physical characteristics are probably the same as WAC material with the exception of the case as noted. As previously mentioned, the seat is faced with a material equivalent to No. 6 stellite.

8. Valve Springs –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 7616)	
Carbon	0.70 to 0.73	Carbon	0.45 to 0.55
Manganese	0.55 to 0.56	Manganese	0.60 to 0.90
Phosphorus	0.18 to 0.20	Phosphorus	0.04 max.
Sulfur	0.45 to 0.50	Sulfur	0.05 max.
Vanadium	0.17 to 0.18	Vanadium	0.15 min.
Chromium	None	Chromium	0.80 to 1.10

Remarks – The hardness range of the DB-601A springs is 44-47 Rockwell C compared with 41-46 for the WAC valve springs and both materials are comparable in their fine sorbitic structure. The German valve-spring material is inferior from the standpoint of its high phosphorus and sulfur content, and from its entire lack of chromium which contributes to the high strength and anti-fatigue properties so essentially required.

9. Crankshaft –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 7320)	
Carbon	0.20	Carbon	0.35 to 0.45
Manganese	0.50	Manganese	0.60 to 0.80
Nickel	2.10	Nickel	1.50 to 2.00
Chromium	2.11	Chromium	0.80 to 0.90
Molybdenum	0.19	Molybdenum	0.20 to 0.30

Remarks – In composition the DB-601A crankshaft steel compares very favorably with the material used in the Cyclone crankshaft. With a somewhat higher chromium content, the DB-601A crankshaft has a hardness of 341 Brinell compared with the range of 290-320 Brinell for the Cyclone crankshaft. The hardened crankpin for the connecting-rod roller bearing tracks is of the order of 60 Rockwell C. Examination of the German crankshaft material structure showed a banded sorbite in a ferrite matrix.

10. Reduction Gear – The DB-601A reduction-gear material is the same type as that used in the crankshaft. (See the chemical analysis of the DB-601A crankshaft material for the approximate composition of the gear steel).

The DB-601A gear was case-carburized on the teeth to a depth of 0.052 in. The hardness of the case was 58 C Rockwell. In structure the case was similar to that obtained in the normal WAC carburizing practice although it contained larger dispersed particles of free carbide in the hypereutectoid zone. The core hardness of the DB-601A gear was 372 Brinell (40 C Rockwell).

In comparison to the DB-601A gear, the WAC reduction gears are made of AMS-6470 nitriding steel and have a specified nitrided case depth of 0.017 – 0.022 in. on the gear teeth with a hardness of 900 Vickers. The core hardness of the WAC reduction gear is 290-321 Brinell.

11. Cylinder Sleeve –

DB-601A Composition, %		Cyclone Composition, % (WAC No. 8190)	
Carbon	0.48	Carbon	0.38 to 0.45
Manganese	0.56	Manganese	0.40 to 0.70
Nickel	0.32	Nickel	None
Chromium	1.63	Chromium	1.40 to 1.80
Aluminum	None	Aluminum	0.95 to 1.35
Molybdenum	None	Molybdenum	0.30 to 0.45

Remarks – The steel used in the DB-601A cylinder sleeves is unlike any of our aircraft steels. In structure it shows a well dispersed uniform sorbite with some free ferrite. The sleeve is relatively soft with a hardness of 230-240 Vickers compared to the 900 Vickers hardness of the nitrided Cyclone barrel.

12. Connecting Rod –

DB-601A Composition, %		Cyclone Composition, % (AMS 6412)	
Carbon	0.47	Carbon	0.35 to 0.40
Manganese	0.67	Manganese	0.60 to 0.80
Nickel	0.28	Nickel	1.65 to 2.00
Chromium	1.02	Chromium	0.60 to 0.90
Molybdenum	0.25	Molybdenum	0.20 to 0.30
Silicon	0.32	Silicon	None

Remarks – In hardness the 313 Brinell of the DB-601A rod compares with the specification requirement of 331-375 Brinell for the Cyclone master rod material but, in the chromium and nickel alloying constituents, there is considerable difference. The structure of the German steel

shows a well-dispersed sorbite of a uniform nature with a small amount of free ferrite.

13. Camshaft and Cam Rollers –

DB-601A (Camshaft & Cam Roller)

Composition, %	
Carbon	0.26
Manganese	0.98
Nickel	0.45
Chromium	1.19
Molybdenum	0.30
Silicon	0.30
Case Hardness	62 Rockwell C
Case Depth	0.025 in.
Core Hardness	44-45 Rockwell C

	Cyclone Cam (WAC No. 7250)	Cyclone Roller (WAC No. 7501)
Carbon	0.08 to 0.13	0.17 to 0.22
Manganese	0.30 to 0.60	0.40 to 0.70
Nickel	4.75 to 5.25	1.65 to 2.00
Chromium	None	None
Molybdenum	None	0.20 to 0.30
Silicon	None	None
Case Hardness	58-63 Rockwell C	60 Rockwell C (min.)
Case Depth	0.035-0.065 in.	0.035-0.045 in.
Core Hardness	29-40 Rockwell C	30-45 Rockwell C

Remarks – The DB-601A camshaft and cam roller material differs from that used in similar parts on the Cyclone chiefly in its nickel and chromium content and also in the inclusion of silicon. Micro-structure examination shows its carburized case to be of approximately eutectoid composition at the surface with good gradation into a sorbitic core containing some ferrite. Due to its uniform appearance, it is believed that the case was obtained by good carburizing technique rather than by grinding off the hypereutectoid portion.

14. Rocker Arm –

DB-601A Composition, %		Cyclone Composition, % (AMS 6312)	
Carbon	0.33	Carbon	0.40 to 0.45
Manganese	0.41	Manganese	0.50 to 0.80
Nickel	2.18	Nickel	1.65 to 2.00
Chromium	1.19	Chromium	None
Molybdenum	0.23	Molybdenum	0.20 to 0.30
Silicon	0.27	Silicon	None

Remarks – The DB-601A rocker-arm steel in comparison with the Cyclone material is identified by a chromium as well as silicon content. It is similar to an American steel, AMS 6410, except for the silicon and slightly higher amount of chromium. In its Rockwell C hardness of 30 it is equivalent to the specification requirement of 29-34 for the AMS 6312 used in Cyclone rocker arms. The DB-601A material shows a micro-structure of finely dispersed sorbite, and contains slightly more oxide inclusions than in similar AMS steels.

15. Crankcase –

DB-601A Composition, %		Cyclone (Present model has forged steel crankcase. Older model has forged aluminum crankcase with following composition per WAC Specification No. 6225)	
Silicon	9.43	Silicon	0.70 max.
Copper	0.01	Copper	0.90 max.
Magnesium	0.36	Magnesium	0.75 to 1.00
Iron	0.57	Iron	1.00 max.
Manganese	0.08	Zinc	4.50 to 6.00
Titanium	0.11	Aluminum	Remainder
Cobalt	0.17		
Zinc	None		
Aluminum	Remainder		

Remarks – The DB-601A crankcase design does not adapt itself to forging and is, therefore, cast of an aluminum alloy with a high silicon content. While the German crankcase material corresponds in hardness, 96-100 Brinell, to WAC cast-aluminum alloys used for similar purposes, it differs in composition by the addition of titanium and cobalt which are probably added for grain refinement. The high-silicon aluminum alloy, as is well known, is used for its excellent casting properties as well as its resistance to corrosion, low specific gravity, and low thermal expansion.

16. Cylinder Block –

Remarks – The DB-601A cylinder block is the same high-silicon cast-aluminum alloy as used in the crankcase. The micro-structure, however, shows a much coarser grain size which may be due to a difference in heat-treatment. The cylinder block Brinelled 104-109.

The foregoing data on a comparison of materials between the major parts of typical high-production German and American aircraft engines are included not as a metallurgical study (which is quite beyond the scope of this paper) but to emphasize the following facts:

(a) The materials used in the DB-601A aircraft engine are quite on a par with those used for similar purposes in the Cyclone engine and, it is believed, in other American aircraft engines.

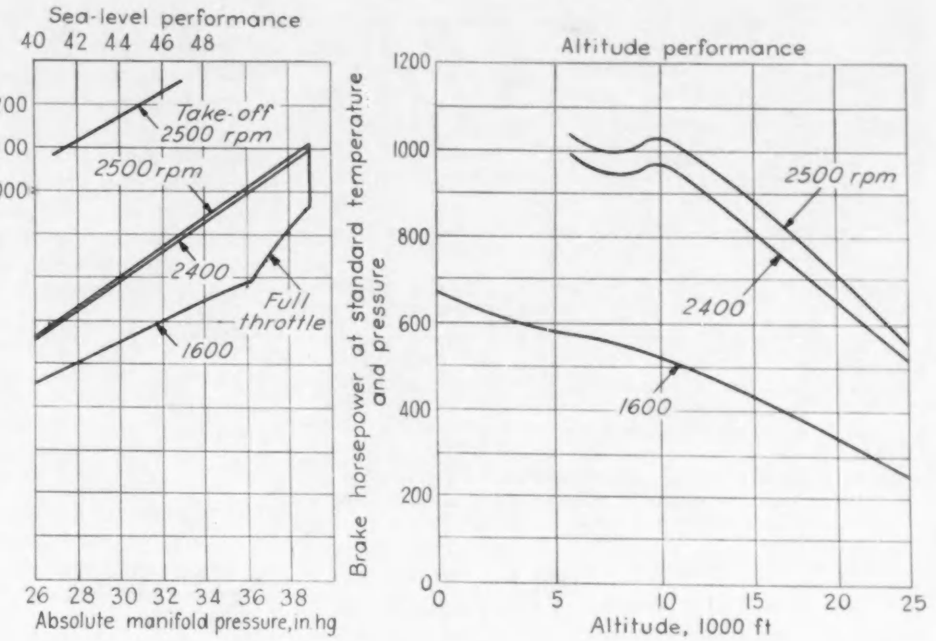
(b) There is no apparent sacrifice in the quality of material used in German aircraft engines up to the time, at least, when this particular DB-601A engine was produced, which is believed to be late in 1939.

(c) Comparison of the steel compositions shows a tendency toward elimination of nickel which may indicate a potential shortage of this strategic material.

(d) No effort has been made to lighten the DB-601A engine through the use of magnesium alloys in the crankcase and cylinder block. In view of the much-publicized development of "Electron" in Germany this is rather surprising. Perhaps, however, the fire-bomb requirement has a higher priority than the aircraft industry.

Performance Characteristics

1. Sea-Level Power Characteristics – In Fig. 32 are plotted the data obtained during a dynamometer calibration of the DB-601A engine. The family of curves on the left-hand side of the figure gives the sea-level power characteristics for various rpm in the operating range, plotted against absolute manifold pressure in in. hg. From these curves it will be noted that the automatic boost control takes charge to restrict the manifold pressure to a limiting value of 39 in. hg absolute at engine speeds above 1900 rpm. Below 1900 rpm the full-throttle manifold pressure is less than the boost limit established by the automatic regulator. The normal rating of the engine at 2400 rpm is thus established at 995 hp. For take-off the automatic regulator is overruled by the pilot's throttle for a time interval of 1 min established by an automatic clock mechanism, to produce an excess boost in the manifold which, at an engine speed of 2500 rpm, gives the take-off rating of 1150 hp at 43.2 in. hg absolute manifold pressure. The take-off power at 2500 rpm versus absolute manifold pressure characteristics are shown at the upper left-hand portion of Fig. 32. The sea-level power ratings are somewhat flexible since it is possible to adjust the oil flow into the fluid coupling of the supercharger and, by changing the slip, vary the effective impeller ratio. Thus it may be desirable for bomber installations involving take-off with heavy loads to increase the



■ Fig. 32 – Dynamometer calibration of DB-601A engine – standard atmospheric conditions, zero ram

effective impeller ratio for the maximum boost possible with the quality of fuel available.

2. Altitude Power Characteristics – The right-hand portion of Fig. 32 illustrates the variation of power output with altitude for two rpm ratings of the DB-601A. The characteristic shape of these curves is somewhat different from that produced by engines with conventional mechanical drive superchargers and therefore is worthy of comment. American aircraft-engine performance curves at altitude usually show power output versus altitude on the basis either of constant manifold pressure to the critical altitude at which full throttle is attained, or of constant horsepower to the full-throttle point at critical altitude. With mechanically driven superchargers the full-throttle point at critical altitude is defined quite sharply. In the case of the DB-601A characteristics, however, it will be observed that there is no very definite critical altitude point and that the power output fluctuates considerably in the altitude range where the automatic controls probably disturb each other with resultant hunting of the engine. Variations of the engine oil temperature also will affect the supercharger coupling slip and contribute to the unusual shape of the performance curves. Changes in the temperature of the induction air within the manifold will also vary the power output.

These test data show that the DB-601A produces on a military rating basis at 2500 rpm, 1000 bhp at 11,500 ft. This is approximately a 3000-ft reduction in altitude compared to the published performance previously given. However, ram at high speed will compensate considerably for this indicated loss.

3. Fuel Consumption Characteristics – The full-throttle specific fuel consumption characteristics of the DB-601A engine at various operating speeds and for three different altitudes are given in Fig. 33. It will be noted that two sets of values are shown:

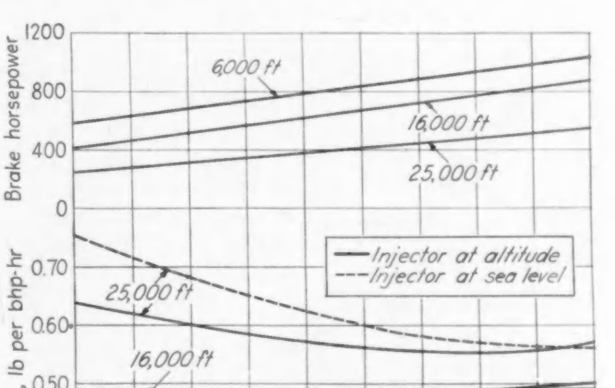
(a) Dotted lines for the fuel injection pump operating under sea-level conditions.

(b) Solid lines for the injector operating under altitude conditions with compensation for reduction in density to agree with airplane installation.

The effect of the compensator on power output was found to be negligible and, on fuel economy up to 16,000 ft, relatively small changes in specific fuel consumption resulted. The action of the compensator was most noticeable in the low-speed range. At 25,000 ft the compensator becomes much more effective particularly as the operating rpm of the engine is decreased.

The general setting scheme of the DB-601A fuel-injection system provides maximum economy for full-throttle operation at low altitudes over a wide range of engine speeds with specific fuel consumption values varying from 0.435 at 1800 rpm to 0.48 at 2400 rpm. For high-altitude operation at 25,000 ft the setting provides maximum power fuel flow with resultant specific fuel consumption of 0.56 at the normal rated speed of 2400 rpm. A check under sea-level take-off conditions gave a specific fuel consumption of 0.54 lb per bhp-hr.

4. Heat Rejection – In Fig. 34 are plotted the values of coolant flow and heat rejection to the coolant against en-



■ Fig. 33 – Full-throttle specific fuel consumption of DB-601A engine

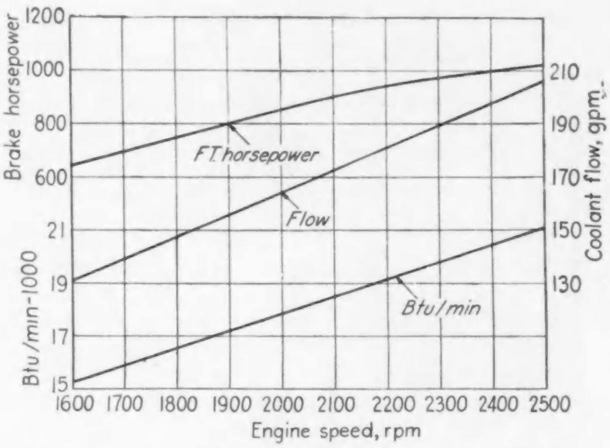


Fig. 34 – Heat rejection to coolant of DB-601A engine at sea level – coolant, 50% by volume of ethylene glycol in water; temperature out, 188 F ± 5 F

gine rpm. These data were taken under sea-level full-throttle operating conditions with a coolant out mean temperature of 188 F. Considering the values at 2400 rpm full throttle, the heat rejection to the coolant is 48% of the horsepower output while at 1900 rpm full throttle the coolant heat rejection is 51% of the brake horsepower. These values are quite conventional for aircraft engines operating at relatively low coolant temperatures.

The heat rejection to the oil characteristics for the DB-601A engine are illustrated in Fig. 35 together with the oil flow plotted against engine speed in rpm. These values represent full-throttle operation at sea level; and the heat rejection to the oil, therefore, includes the heat rejection from the fluid coupling oil which is operating under maximum slip conditions at sea level. Subtracting 1000 Btu per min for the coupling heat rejection, the resultant value of 2750 Btu per min heat rejection to the oil for the engine at 1000 hp is approximately 750 Btu per min greater than the Cyclone operating at the same power output.

5. Supercharger Fluid Coupling Characteristics – In order to obtain information on the actual performance characteristics of the DB-601A supercharger fluid coupling as well as of the supercharger itself, the complete unit with its accessory drive section was set up on a special test rig as shown in Fig. 36. Test data on the fluid drive coupling were taken over a considerable range of values for engine speed and power, oil temperature, pressure and flow; and compressor output. In Fig. 37 are shown the graphical relation between altitude on the horizontal axis and heat rejection to the oil in the fluid coupling; slip of the coupling; effect ratio of the impeller to crankshaft speed; coupling oil pressure, and coupling oil flow. The operating condition of the engine for these specific curves was the normal rating at 2400 rpm. One point for a take-off condition at 2500 rpm is also indicated.

From the results plotted in Fig. 37 the characteristics of the fluid coupling are as follows for the conditions indicated:

(a) The slip varies from approximately 28% at sea level to 2% at slightly over 12,000 ft (depending on the sylphon adjustment) when the full capacity of both oil pumps is delivered to the coupling. The slip characteristics may change somewhat with oil temperature, viscosity, and the amount of oil through the drain hole in the periphery of the coupling housing.

(b) During the slipping process, an appreciable amount of energy is dissipated into the oil within the fluid coupling in the form of heat. The heat-rejection curve shows that during maximum slip at sea level the oil carries away approximately 1000 Btu per min which must be dissipated through the oil cooler system. For take-off power and speed conditions, the supercharger fluid drive coupling rejects 1200 Btu per min to the oil. While this additional oil-cooling requirement is the price that must be paid for a hydraulic device which gives a smooth and variable speed drive for the supercharger impeller, there are several operating conditions which may compensate for the indicated increase in oil cooler capacity, particularly in the case of fighter installations. With cold engine take-offs and high rate of climb, the airplane quickly passes through the altitude at which maximum slip in the fluid drive occurs and attains its operating altitude before the oil in the entire system has had time to become thoroughly heated. Since a majority of flight missions in this war are reported to be carried out above 12,000 ft where coupling slip becomes negligible, the matter of increased oil cooler capacity may not involve the degree of heat dissipation indicated by laboratory tests.

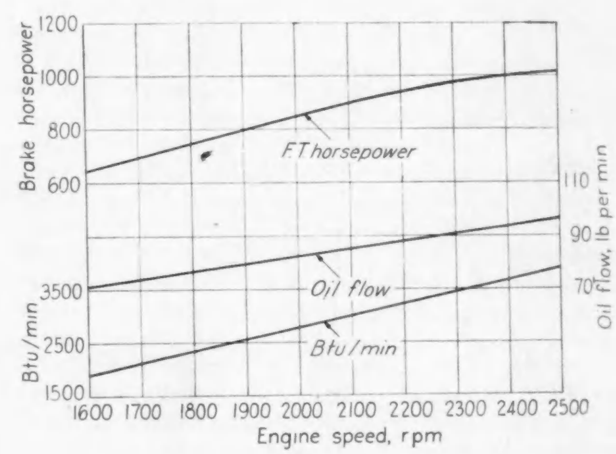
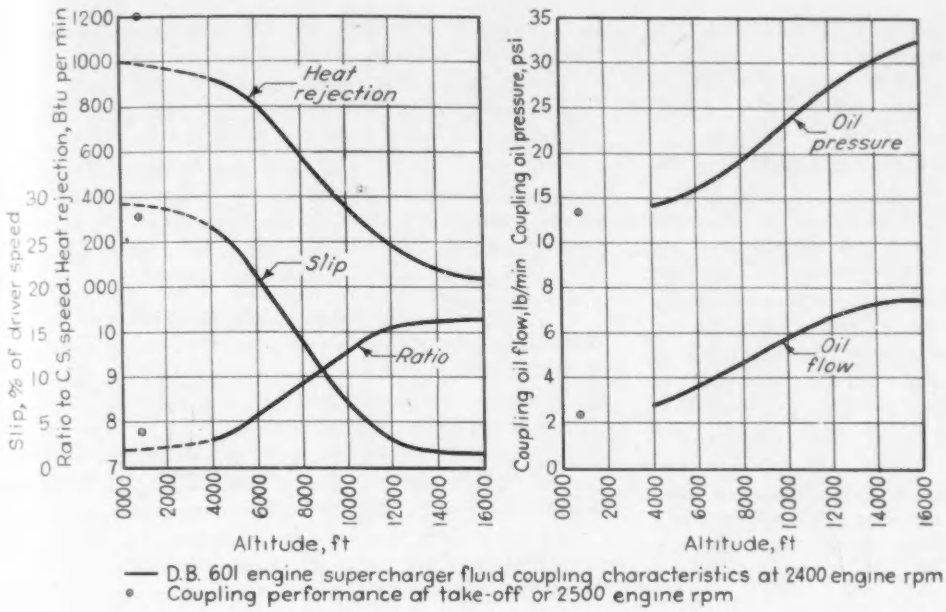


Fig. 35 – Heat rejection to oil of DB-601A engine at sea level – lubricating oil, Specification No. 9532, Grade 1120; oil temperature in, 155 F ± 5 F

(c) From the curve showing the ratio of impeller speed to crankshaft for various conditions of slip and altitude, it will be noted that, up to 4000 ft (and subject to adjustment of the automatic controls), the slip, and therefore the impeller ratio, remains substantially constant. The effective impeller ratio for take-off and initial climb is of the order of 7.5:1. With both oil pumps delivering full capacity to the fluid coupling above 12,000 ft, the impeller ratio at minimum slip is established at 10.2:1 with respect to the crankshaft speed.

In Fig. 38 are plotted against altitude three variables, that is, impeller tip speed, load coefficient, and supercharger outlet pressure, which indicate the functioning of the fluid coupling on supercharger performance under a constant engine speed of 2400 rpm. The secondary pump regulation is such as to supply the necessary amount of oil with increasing altitude to the fluid coupling as to increase the impeller



■ Fig. 37 – Relation between altitude and heat reduction to the oil in the fluid coupling, slip of the fluid coupling, ratio of impeller to crankshaft speed, coupling oil pressure, and coupling oil flow – DB-601A engine

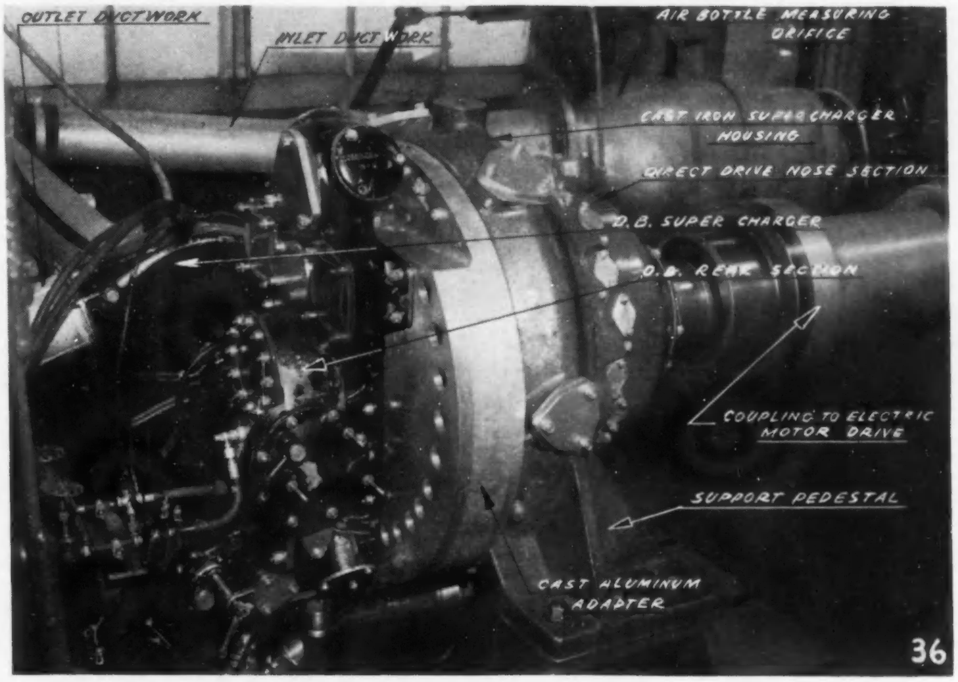
tip speed (lower curve) and maintain an almost constant discharge pressure up to 12,000 ft (upper curve). Beyond this altitude the coupling is hydraulically locked and higher tip speed of the impeller cannot be attained. It will be noted that the maximum impeller tip speed is slightly below 1100 fps. This value is approximately 150 fps lower than the current standards for American and British supercharger design.

The middle curve of Fig. 38 shows the characteristics of the load coefficient Q/N, at 2400 engine rpm, with the variations in tip speed of the impeller. The load coefficient Q/N is the quantity of air in cubic feet delivered per revolution of the supercharger impeller. The load-coefficient curve is rather flat considering the wide variations in tip speed of the impeller and that the throttling control is established at the outlet instead of the inlet to the compressor.

6. Supercharger Performance – In Fig. 39 are reproduced curves selected from rig test data on the DB-601A supercharger to illustrate its characteristics under two specific conditions as compared with the performance of the current Cyclone supercharger. These two conditions are defined by impeller tip speeds of 800 and 1100 fps which represent take-off, and altitude performance above 12,000 ft.

Supercharger characteristics are generally defined by the relationship between the load coefficient Q/N and two variables: (a) temperature rise ratio and (b) pressure coefficient.

(a) The temperature rise ratio is a measurement of the overall efficiency and is the ratio of calculated adiabatic



■ Fig. 36 – Rig for test of DB-601A supercharger

temperature rise based on observed pressure ratio to the actual observed temperature rise.

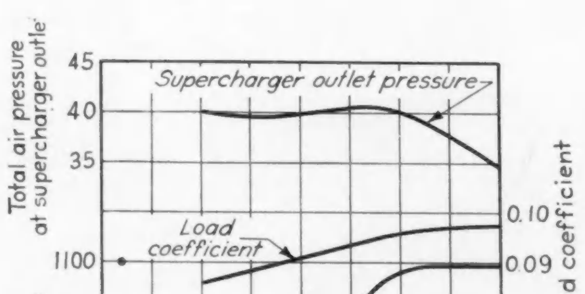
(b) The pressure coefficient is an indication of the ability of the compressor to provide boost at a given tip speed, and is the ratio of the energy from the pressure ratio actually obtained to the calculated energy imparted to the air stream when accelerated to the tip speed of the impeller.

The flat spread of the DB-601A supercharger performance curves indicates a characteristic of the vaneless diffuser which, however, is somewhat inferior to the vane type in efficient conversion of velocity energy to pressure. It has been reported that early models of the DB-601A engine incorporated a vane-type diffuser. The reason for a subsequent change to the vaneless diffuser may possibly have been made to decrease the supercharger diameter; facilitate production; or through the flatter performance curve to eliminate a possible surging condition. The reasonably satisfactory comparison with the Cyclone vane diffuser supercharger indicates that the DB-601A impeller is of good design and compensates to some extent for the deficiency of its vaneless diffuser.

■ Conclusions

Since no report on a subject of such current interest as the DB-601A aircraft engine is likely to be considered complete without some conclusions, the writer will undertake herewith to summarize, and in some cases re-affirm, the impressions gained and observations made during his rather limited study of this powerplant.

1. Workmanship and Quality of Finish – While the engine first presented a somewhat discouraging appearance upon being unpacked, with its dull black finish and a dangling mass of wires, controls, fuel, oil and coolant lines, and so on, it soon became evident upon dismantling that good design, high quality, and excellent workmanship lay beneath a somewhat homely exterior. No useless effort in man-hours or finish has been expended where there is not a direct return in increased reliability or performance. The general workmanship indicates the application of suitable machine tools, skilled operators, and efficient inspection personnel. Handiwork in polishing of stressed parts is of the highest order and consistent in extent to the requirements for high specific output, reliability, and long service life in a mass-production aircraft powerplant.



■ Fig. 38 – Relation between altitude and impeller tip speed, load coefficient, and supercharger outlet pressure – DB-601A engine

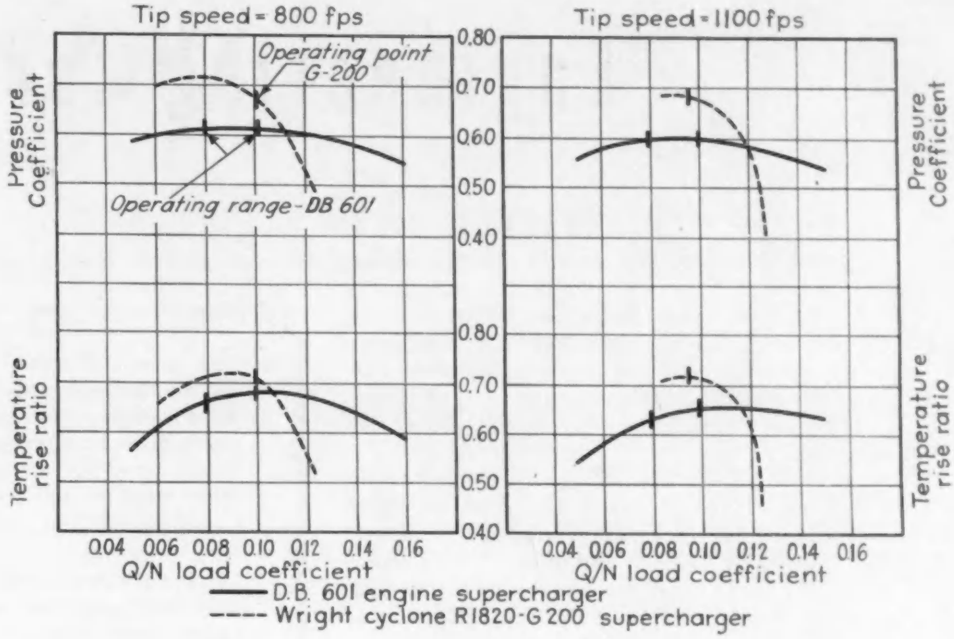
2. Design – The DB-601A general design reflects a ruggedness and reliability which has always characterized Mercedes-Benz products. The relatively low rating of the engine indicates conservatism in output for the sake of improved reliability and increased service life. This safety factor has been effected despite the standardization of mounts and attachments first employed by the Germans to facilitate engine changes. This latter procedure has greatly reduced the necessity for complicated maintenance operations on the powerplant while installed. However, routine maintenance operations on spark plugs, injection nozzles, strainers, and magneto are easily accomplished on the DB-601A.

Careful attention has been paid to seemingly unimportant details which may mean the difference between success and failure in a given design. Specific reference in this connection is made to the doweling practice and serrating of joints referred to throughout the foregoing detailed description for the purpose of eliminating chafing and ultimate trouble; also the surface-hardening procedure by means of shot blasting on stressed steel surfaces in intimate contact to reduce fretting and the possibility of fatigue failures.

3. Adaptability to Mass Production – From the manufacturing point of view, the design represents, with possibly several minor exceptions, good mass-production practice for military aircraft powerplants with the use of modern machine tools especially adapted for specific operations. While the grinding of gears is on a par with American practice, it is somewhat surprising to note that highly stressed bolts do not have ground threads.

4. Materials – The high quality of German aircraft steels is reflected in the marked absence of magnaflux indications. In so far as the strategic alloying constituents are concerned, with the possible exception of nickel, there is no appreciable evidence that at the time this engine was constructed, any shortage of tin, chromium, tungsten, and so on, existed for the German aircraft industry. There is also a noticeable similarity in the application and composition of many materials with those used in American aircraft engines. The one exception is the almost total lack of magnesium alloys which were exploited greatly prior to the war by the German light metal industry.

5. Operation – No little thought and considerable ingenuity of design have been directed to the simplification of engine operation through the use of automatic controls. For military powerplants this requirement in time of war is of utmost importance due not only to the burden of combat maneuvers which keep the pilot occupied but also to the inability of inexperienced pilots properly to control manually the many adjustments which are required for optimum performance of an aircraft engine under varying conditions of output and altitude.



■ Fig. 39 – Characteristics of DB-601A engine supercharger and Wright Cyclone engine supercharger at impeller tip speeds of 800 and 1100 fps

Another interesting phase of military operation which is reflected in the design and installation of the DB-601A is the requirement for an unfaltering take-off with a stone-cold engine. This is particularly required of fighter and interceptor powerplants. It has been reported from abroad that take-off with a cold engine in the German fighting planes is accomplished on a few seconds notice. A supplementary tank filled with a mixture of ether and gasoline is used for priming. A secondary tank containing fuel mixed with a small percentage of oil is used during take-off and climb. The take-off is made almost immediately after starting the engine and the switch to straight gasoline is not made until the normal operating temperatures of coolant and oil have been attained. Such high-output operation with a cold engine and resultant sluggish oil circulation may be the reason for the use of so many antifriction bearings in the connecting rods and accessory drives of the DB-601A engine.

6. Performance – Despite wishful thinking to the contrary, the performance of the DB-601A with respect to sea-level and altitude output, fuel consumption, and weight seems to be quite on a par with other contemporary powerplants of the same general type.

7. Potentialities of the DB-601A – During the normal course of development, a military aircraft powerplant will undergo an increase in power output averaging 6 to 8% per year for a given type. This higher performance may be accomplished by several or all of the following modifications:

(a) Increasing the degree of supercharging.

(b) Improving supercharger or induction system efficiency.

(c) Increasing crankshaft rpm.

(d) Increasing compression ratio.

In the stress of war-time necessity the degree of increasing power output yearly for a given type may attain 10 to 12% when proper emphasis has been established on the necessity of continuing development despite the urge for mass production of a current model. Assuming that the DB-601A engine under discussion is a model released for production in 1939, it is not unlikely that in 1940 this engine attained a military rating of 1175 hp and that the current rating in 1941 is 1400 hp. In view of the reports from abroad that German planes are using 92-octane fuel such an output is quite within reason.

Since the impetus of war with its ever-increasing demands for higher airplane speeds and heavier bomb and armament loads dictates the necessity for powerplants of greatly increased output, it is not at all unlikely that the basic design of the DB-601A has been incorporated into an X-type engine. Such a development would be a logical production set-up utilizing known and service-proved components, and accordingly may well be the 2400-hp powerplant recently reported to be under construction in Germany.

In conclusion and by way of acknowledgment, I wish herewith to tender my sincere thanks to the following:

(a) To my associates in Wright Aeronautical Corp. who conducted the tests; compiled the data; constructed the curves; drew the sketches; and made the photographs:

Messrs. P. Bancel
K. Campbell
J. H. Critchley
J. R. Douglass
P. Graef
O. Hollerith
H. Jones
W. G. Lundquist
E. F. Pierce
J. Spurling
J. Talbert
M. H. Young

(b) To W. H. Sprenkle and E. K. Von Mertens of Pratt & Whitney Aircraft, for the use of the sketch showing the accessory gear train of the DB-601A

(c) To R. M. Hazen of the Allison Division, General Motors Corp., for data on the V-1710-C-15.

Engineering for Better

IT is the purpose of this paper to review briefly the progress made over the past few years in improved fuel economy of our gasoline engines and to discuss the factors which may be expected to improve further the efficiency of the automobile.

THIS paper reviews the experience of one company over a period of years in developing its product to give better fuel economy in spite of the demand for increased performance and the rising trend of car weight which has been occasioned by advance styling and appearance items.

The data presented indicate how air-fuel ratios have been improved and compression ratios increased over a period of years, and some of the other factors influencing fuel economy. The effects of compression-ratio increase are discussed along with the use of higher-octane fuel and the necessity of proper choice of performance factors

We must bear in mind that each road to improvement is beset with problems which have yet to be solved, otherwise, we already would have incorporated such improvements into our products, rather than merely talking about them here. Doubtless, many of these problems can and will be solved when sufficient engineering attention is directed to them.

That gains in economy have been made during the past ten years, in spite of the trend toward heavier and heavier cars, and in spite of additional hill performance, is shown on Fig. 1, "Relation of Fuel Economy to Performance During Past 10 Years." Here, we have graphically plotted the fuel economy at 50 mph, the weight of the car, the performance factor, and the performance over the Proving Ground 11% hill in seconds of the Oldsmobile 6-cyl, Model 70 car. The major point here made is that, even though the weight and hill performance have increased, gains in miles per gallon of gasoline have been accomplished.

Enough encouraging results have been attained experimentally to assure us that continued progress can be made in gasoline consumption and that this miles-per-gallon curve will continue to rise during the years to come.

The use of automatic transmissions such as the Hydra-Matic drive, which permits of low engine revolutions at

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 4, 1941.]

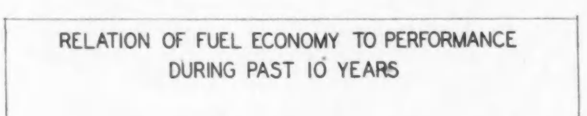


Fig. 1

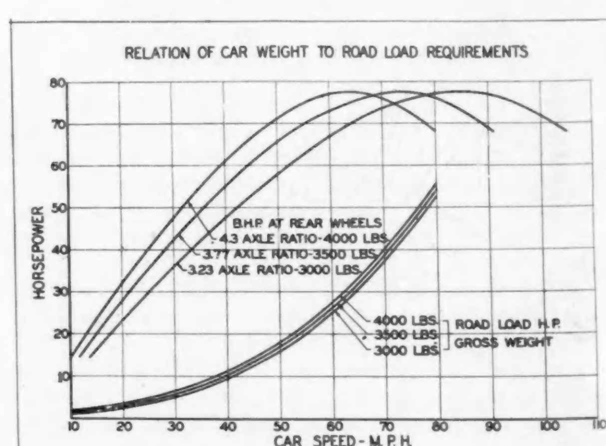


Fig. 2

FUEL ECONOMY

by H. T. YOUNGREN

Chief Engineer, Olds Motor Works Division, General Motors Corp.

to obtain maximum benefit of these factors to get the best economy. Among other things discussed are the effect of high versus lower performance factors and the effect of different rear-axle gear ratios.

A comparison of a small, high-speed engine versus a large, low-speed engine is made along with data on the effects of engine friction and mechanical efficiency, since it is evident that any fuel economy obtained by reducing engine speed has really been due to the reduction of engine friction along with higher throttle operation for a given car speed.

cruising speeds, has made further inroads on this economy problem. Yet the question at the moment is: Where do we go from here?

Basic Factors

In attacking the fuel-economy problem, let us classify its factors in three groups:

1. The horsepower-hours required to move the vehicle over a given distance.

2. The brake thermal efficiency of the powerplant at the particular speeds and loads used.

3. The amount of reserve power desired for rapid acceleration and hill climbing.

Weight of Car

Under group number one, horsepower-hours required to move the vehicle over a given distance, the first fundamental that comes to mind is the weight of the car itself.

The major effect of car weight on gasoline economy is not its influence on level rolling resistance at constant car speeds, but rather its high reserve horsepower requirements for satisfactory acceleration and hill climbing.

Since the ability of a car to accelerate is proportional to the ratio of its available excess thrust at the wheels to the weight of the car, reduction in weight alone will be accompanied by a fairly fast change in acceleration. If we select a maximum acceleration at some arbitrary speed, then reduction of car weight can be accompanied by change of axle ratio to reduce engine speed in relation to car speed. Thus, the mechanical efficiency of the powerplant may be

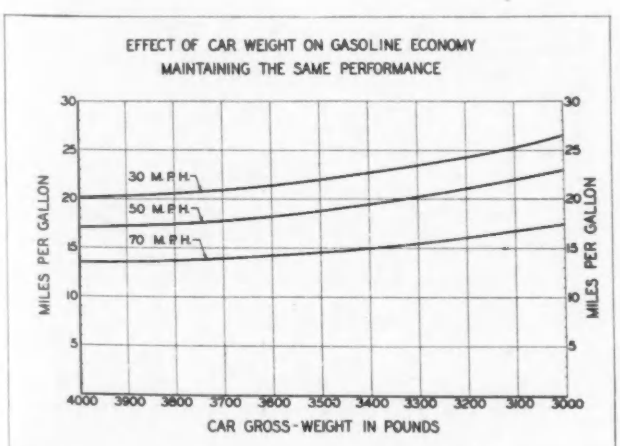


Fig. 3

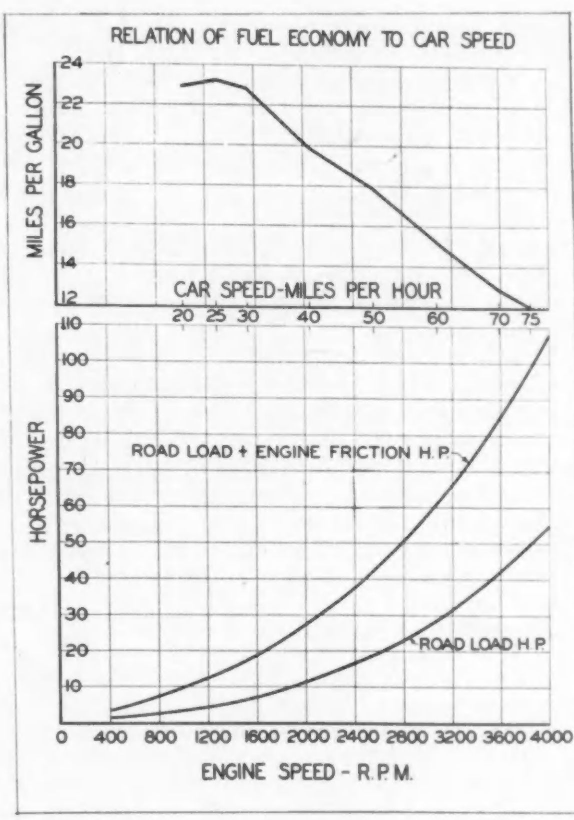


Fig. 4

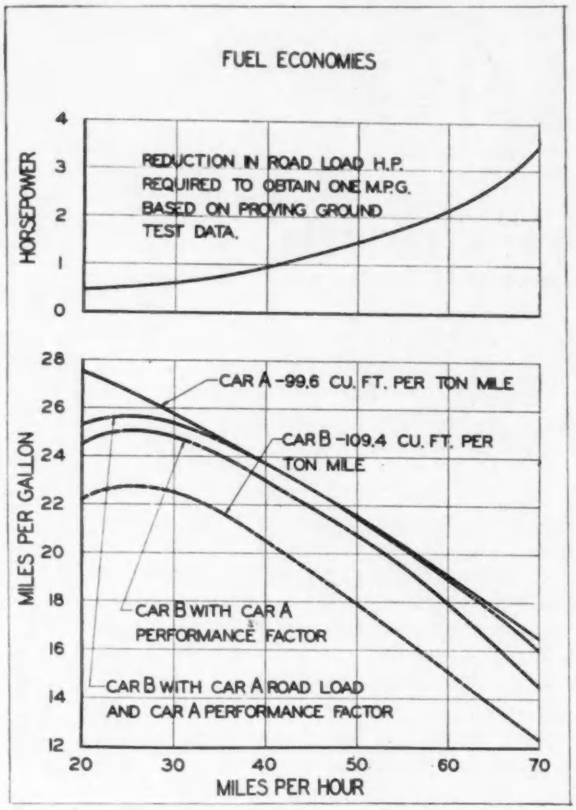


Fig. 5

improved over its whole operating range, and fuel consumption appreciably reduced.

Fig. 2 presents the conventional rear-wheel brake horsepower curve with axle ratios selected to give equal performance with various weights of car. Weights of 3000-lb, 3500-lb, and 4000-lb load are here studied.

Here, also, is shown the comparatively minor effect of car weight on rolling resistance.

Realizable economies are shown on Fig. 3 when the weight of a car of fixed proportions is varied between 3000 and 4000 lb, maintaining a constant performance factor by altering the axle ratio.

These curves were arrived at by cross-plotting chassis dynamometer data. Numerical values exhibited in this instance should not be taken too literally. The chart, however, does convey a fair approximation to fact and supports the contention that appreciable gasoline economies can be secured through weight reduction when accompanied by slowing down of the engine.

Perhaps future economic conditions will dictate weight reduction for reasons other than fuel economy.

■ Effect of Car Speed

Concurrently with the weight of the vehicle as a basic influence on economy, we have the major influence of car speed. Unlike weight, higher car speeds are essentially desirable for transportation; therefore, reduction of car speed cannot be considered as any practical help in the gasoline mileage problem. The factors, then, which cause high fuel usage at high speeds must be looked to for possible improvement. Car rolling and wind resistance plus engine friction are the causes of the drop in the miles per gallon chart as speed is increased.

The top curve of Fig. 4 shows miles per gallon versus car speed. The bottom curve shows power demanded to overcome rolling and wind resistance. However, the engine must overcome not only these resistances, but those of its own internal friction as well.

The variation of the sum of these demands, and particularly its rapid increase at higher speeds, accounts for the fairly rapid reduction in economy as speeds are increased. This is exhibited by the middle curve of Fig. 4.

■ Rolling and Wind Resistance

The rolling and wind resistance effect on economy is shown on the upper section of Fig. 5 in terms of the amount of road-load reduction necessary to gain 1 mpg of gasoline.

The lower set of curves is of a fuel-economy comparison between two different makes of cars, Car *A* having a very good economy curve and Car *B* more or less average result in this respect. Car *A* has a comparatively low performance factor of 99 cu ft per ton-mile, whereas Car *B* has a normal performance factor of about 110 cu ft per ton-mile. We see here that, by reducing the performance factor in Car *B* to that of Car *A*, a substantial gain in economy results; yet a difference still exists. This difference was then overcome over most of the speed range when Car *B* was run with Car *A* performance factor and Car *A* road-load power requirements. Obviously, Car *A* had less wind and rolling resistance than Car *B*.

Rolling resistance and car performance factors in terms

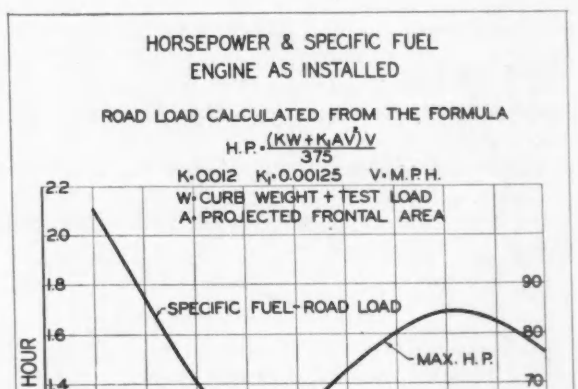


Fig. 6

of cu ft of engine displacement per ton-mile are major factors in economy. The latter effect is discussed in greater detail later.

Thermal Efficiency

While the weight of the car and power required to move it at various speeds are of much importance to fuel economy, most progress has been made during the past years, and hope for future improvement lies in the factors affecting the brake thermal efficiency of the engine.

Under this heading, we will deal with the effect of engine friction, compression ratio, engine size, exhaust scavenging, and induction systems.

Specific Fuel at Full and Part Throttle

The specific fuel consumption at road load is much higher at part-throttle low car speeds than at wide-open throttle. For instance, in Fig. 6, at 50 mph road load, a specific fuel consumption of 0.9 lb of gasoline per bhp-hr is observed whereas, at full throttle, we see a specific fuel consumption of 0.58. In terms of brake thermal efficiency, this means about 14.9% at road load and 23% at the same speed with wide-open throttle.

Offhand, one might suspect this comparative inefficiency at light throttle to be due to some actual combustion or adverse thermal condition existing because the engine is operated at light throttle and under low compression pressures. However, analysis shows that the poorer efficiency at part throttle is due to the lower percentage of power output in proportion to engine friction. In other words, the exaggerated case of this condition occurs when the engine is run idle with no load output, all the gasoline burned being used to overcome engine friction. In such a case, the efficiency is zero.

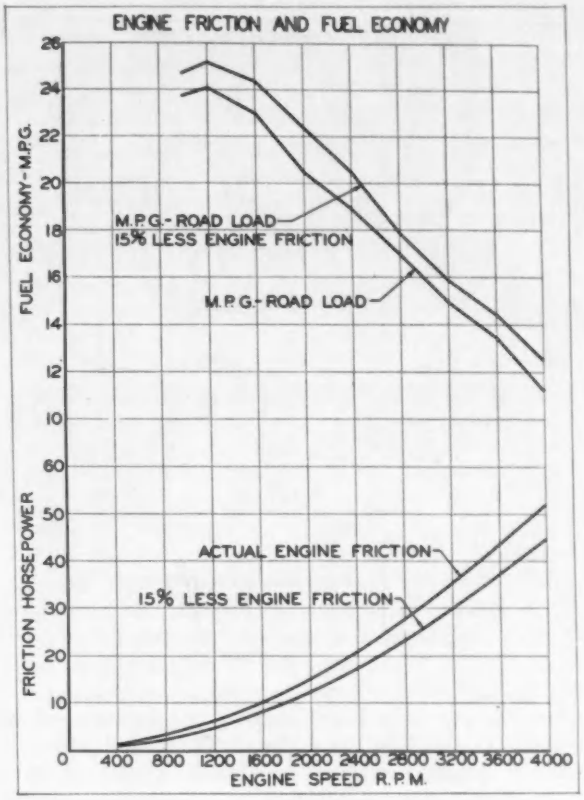


Fig. 8

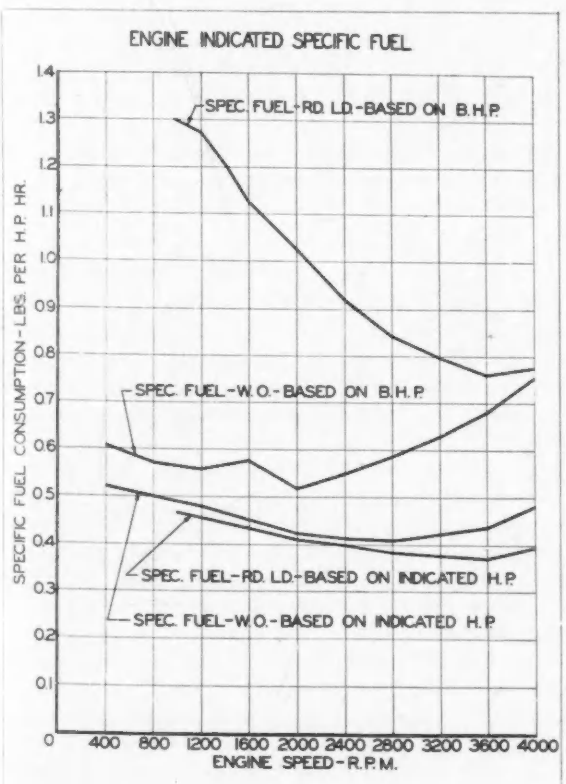


Fig. 7

Effect of Engine Friction

That this low efficiency at road load is due to friction is shown further on Fig. 7 where the specific fuel at full and part throttle is plotted in terms of indicated horsepower.

Here, then, based on indicated horsepower, the light load specific fuel assumes a position below the wide-open indicated specific fuel; the maximum-economy mixture ratio shows some of its advantages over the maximum-power mixture ratio used at wide-open throttle.

Fig. 8 graphically illustrates the results of a test wherein the engine friction was arbitrarily reduced 15% and the road-load economies observed. At 40 mph we see a gain of nearly 2 mpg when the friction was reduced 2¼ hp.

It is interesting to note that this gain is substantiated by calculations:

$$\text{Mpg} = \frac{W\,V}{F \times \text{Ihp}} \quad \text{where}$$

W = Weight of 1 gal of fuel, lb
V = Car speed, mph
F = Indicated specific fuel consumption, lb per ihp-hr
Ihp = Road load indicated horsepower

At 40 mph the road-load ihp was 28½ and the indicated specific fuel, 0.42. Inserting these values above gives 20 mpg. Then, by reducing the engine friction 2¼ hp using an ihp of 26¼, the result becomes 21.8 mpg, a calculated

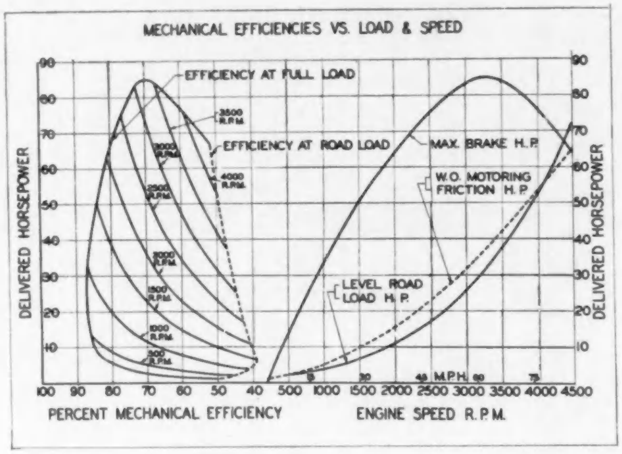


Fig. 9

gain of 1.8 mpg of fuel, fuel weighing 6 lb per gal.

The subject of friction and its influence on economy is stressed so much because, based on an indicated thermal efficiency of 33%, we then pay about 3 hp in terms of gasoline heat units for every 1 hp of frictional loss.

Here then, a study of the mechanical efficiencies of the engine under various power-output loads and speeds is shown graphically on Fig. 9. These curves emphasize the idea of minimum engine speed in the interest of maximum efficiency, and are further interesting in that they form a ready reference for determining the mechanical efficiency under varying loads and speeds.

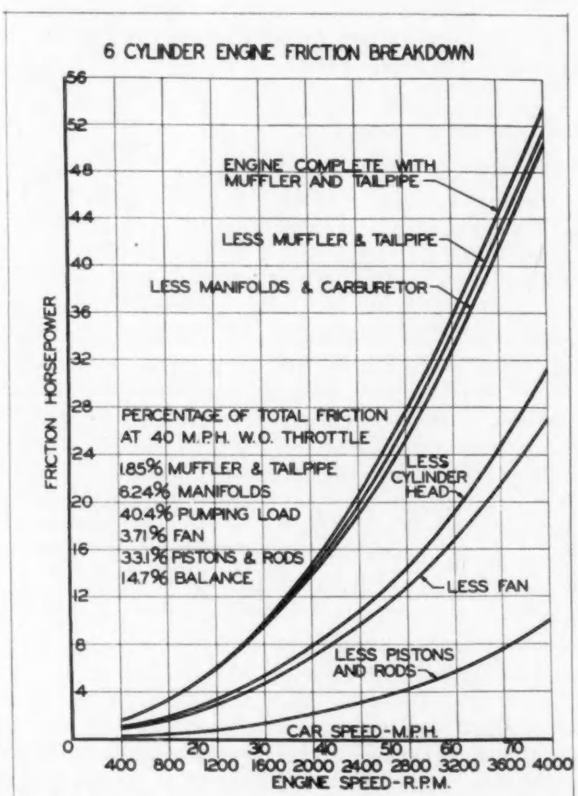


Fig. 10

Referring to this chart, in the field bounded by the full-throttle horsepower and level-road horsepower, any chosen point represents some condition of operation, coasting excepted. To illustrate, let us arbitrarily choose a speed of 40 mph and discuss the mechanical efficiency at various loads.

At full load, projecting to the left from the maximum-horsepower curve to the curve of efficiency at full load, thence down to the efficiency scale, we get 81%. At road

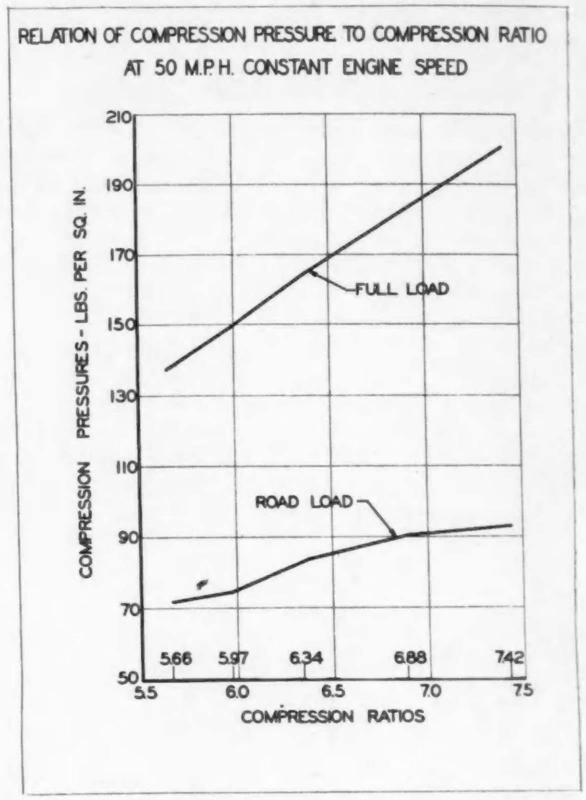


Fig. 11

load we project from the level-road horsepower curve to the efficiency scale we get 41%, which is unfortunately low from an economy standpoint. Likewise, for any given load conditions between the maximum power curve and the road curve, the mechanical efficiency can be estimated closely.

It appears then that an attempt to reduce engine friction deserves much engineering effort. While it is admitted that, as far as our investigations have gone, there is nothing accurately known about friction in engines when operating under power, we can use motoring friction as a guide to point out the units which contribute the largest percentage of loss.

To that end, Fig. 10 shows a breakdown of engine motoring friction and the percentage of total friction chargeable to the various major units. In the case of this engine, the apparent pumping loss is by far the largest contributor and, as such, offers a fairly large field for investigation. This pumping loss exists in spite of a comparatively high volumetric efficiency of 82% at 1200 rpm and 63% at 4000 rpm.

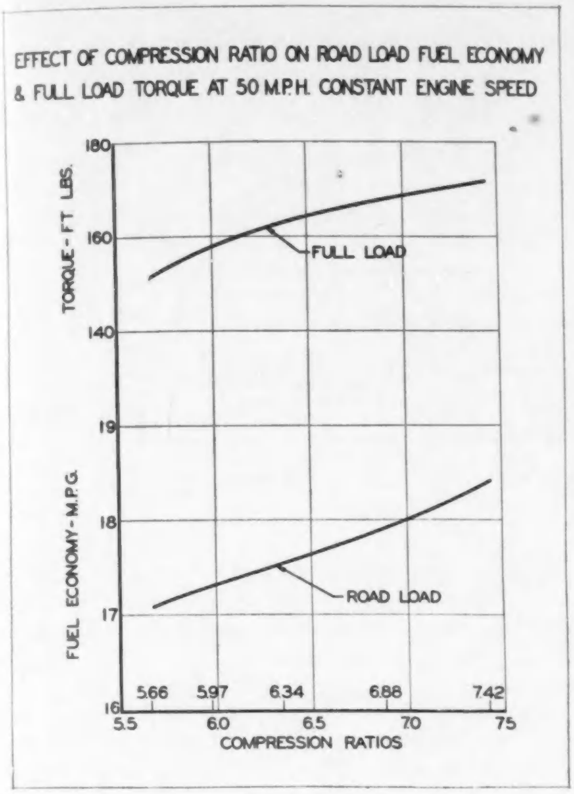


Fig. 12

Pistons and rings are a large contributor to friction and, again in the case of the engine under discussion which uses four rings to insure oil economy, the elimination of just one oil ring on each piston resulted in a gain of 1 mpg of gasoline at 50 mph.

If it can be accomplished, a reduction of engine friction will pay big dividends in the way of light-load thermal efficiency and additional miles per gallon of gasoline.

Compression Ratio

Increased compression ratio and its effect on gasoline economy has been the subject of many discussions and has been covered in great detail by previous papers presented to the Society. Higher compression ratios will continue to be contributing factors to better fuel economy.

Doubtless others are better qualified to tell just when gasoline of the necessary octane ratings for 8:1, or higher ratios will be commercially available as standard fuel. At the present time, considering eventual carbon build up, 6.7:1 is as high as we, ourselves, operate commercially on 74-octane fuel.

On Fig. 11 the relationship of compression pressure to compression ratio is pictured. These data were observed on an Oldsmobile 6-cyl car at both full- and road-load throttle. It is interesting to note the change in compression pressure at road load compared with full throttle as the ratio is increased. The part-throttle result is influenced by the additional throttling necessary to maintain car speed power requirements. Fig. 12 shows a substantial gain at 50-mph road load in economy with increased compression ratio, all other factors in the engine as well as the fuel being held constant. It is contended that, by increasing compression ratio, the simultaneous increase in expansion ratio is more responsible for the improved engine efficiency than is the resultant higher compression pressure.

Compression ratio is accepted as being the sum of piston displacement plus combustion-chamber volume divided by combustion-chamber volume.

Expansion ratio may be defined as the volume swept by the piston up to the time of exhaust release, plus the combustion volume, divided by the combustion volume.

In the engine tested, the exhaust valve was off its seat 0.010 in. when the piston had traveled 3.746/4.126 or 0.908 of its stroke.

Thus the 7:1 compression ratio on this engine has an expansion ratio of 6.45; likewise, the 6:1 compression ratio has a 5.54 expansion ratio.

Further increase in pressure occurring after the exhaust valve has opened has not been taken into account here because it is believed to be a negligible factor at light-throttle operation.

If increased expansion ratio is a major contributor to light-load economies then, at least on the engine observed, a study of later exhaust-valve opening is indicated. This would tend to increase the expansion ratio and may offer some additional gasoline economy.

Increasing the expansion ratio by means of higher compression ratio, given suitable high-octane fuel, improves specific economy and performance at the same time. To get the fuller realization of potential economy, advantage of the power gain must be taken to reduce either engine size or engine speed. As has been shown, the mechanical efficiency of the powerplant can be increased appreciably by reducing engine speeds for given car speeds.

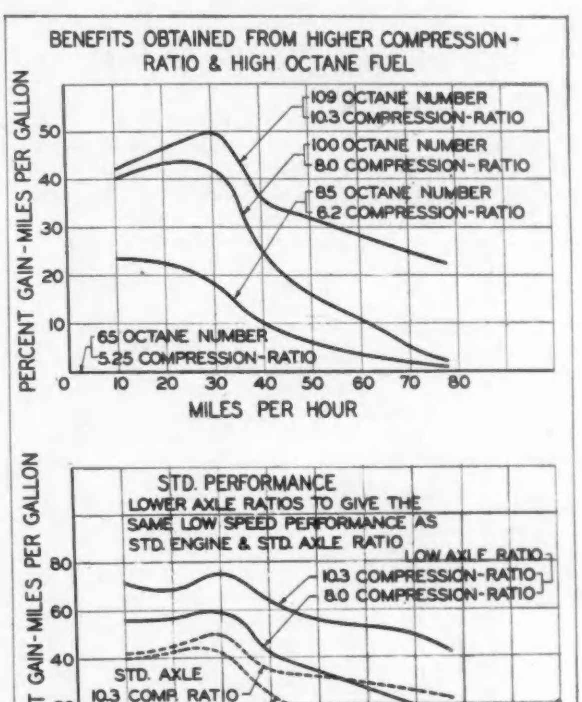


Fig. 13

As applicable to the higher compression ratios, this point is substantiated in Fig. 13, which shows in the upper half the percentage economy gain made throughout the speed range when the compression ratio is increased successively from 5.25 to 6.2 to 8.0 to 10.3, without an axle change. The lower section of Fig. 13 shows the combined gain because of increased compression ratio and reduction of axle ratio to give the same low-speed performance as the original lower compression job. Here we show a maximum gain in economy of 60% at 30 mph with a suitable axle ratio and an 8:1 head. With the head only changed, we gained about 40%. At 70 mph the 8.0:1 compression ratio alone showed only a 5% gain but, with the lower axle ratio, a 20% gain was experienced.

This comparison is of particular interest in that it indicates that increased compression ratio alone does not result in important economies at high speeds. To get the real benefit in economy from increased compression ratio, keeping the same engine displacement, it is necessary to change the drive ratio to cause lower engine speed, thereby reducing the indicated road-load horsepower requirements. This is permissible without loss in performance because of the increased output of the high compression engine.

■ Large Engine Vs. Small Engine

Suppose that, rather than reduce the axle ratio, we take advantage of the increased power output of the higher compression ratio and reduce the size of the engine. Fig. 14 illustrates such a comparison made by the General Motors Research Laboratory. The results indicate a marked superiority in economy for the larger engine running slower. This is shown to be particularly emphasized at the higher vehicle speeds. Each engine was equipped with an axle ratio to give the same low-speed car performance, yet at 70 mph, the larger engine having a 10:1 compression ratio gave a 45% increase in power output, whereas the smaller engine running at higher speed gave only 20% gain in power output.

In terms of economy the same engine with a 10:1 compression ratio showed a 30% gain at 70 mph with the smaller engine, and a 50% gain with the larger engine running at low speed.

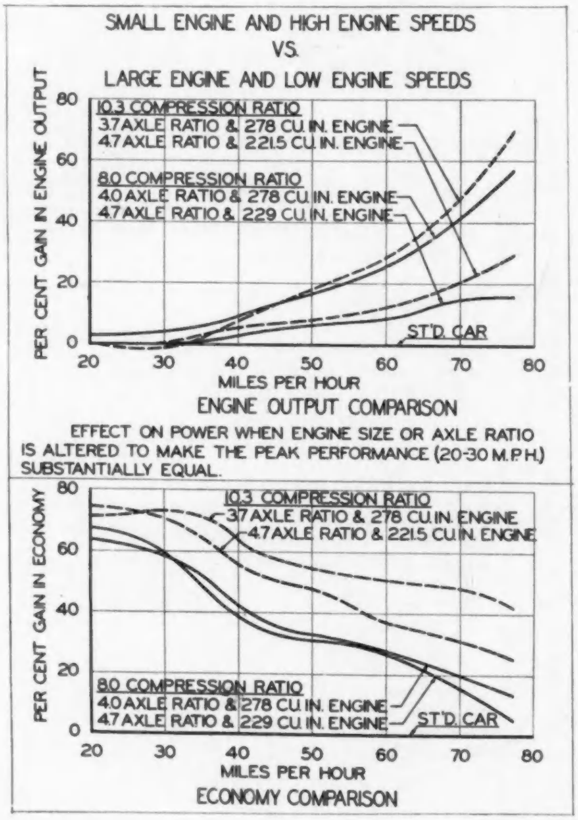


Fig. 14

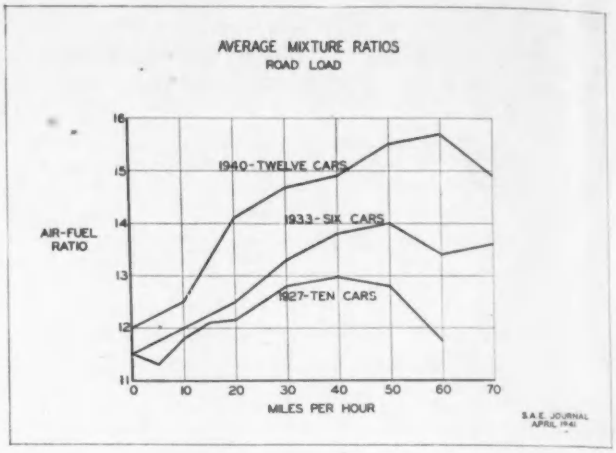


Fig. 15

■ Mixture Ratio

Another well-known factor in any fuel economy program is the gasoline and air mixture ratio. It will be recalled that, during the 1941 Annual Meeting of the Society, data were presented showing the trend during the past years towards leaner mixtures. A reprint of the history of mixture ratios is shown here in Fig. 15, whereon we see that air-fuel ratios of 12:1 were common in 1927 and currently we are burning 15 lb of air to 1 lb of fuel.

Quoting from a General Motors Research Laboratory report: "The lean limits for representative hydrocarbons as determined in laboratory vessels are between 25 and 26:1." Continuing from this report: "The limiting lean mixture for steady operation with gasoline in a single-cylinder engine at loads between 50% and 100% of full load and at 800 rpm appears to be about 19:1." Fig. 16 shows the per cent gain in specific fuel consumption as mixture ratios are increased between 13.5 and 19:1. These results were obtained on a single-cylinder engine with spark setting for maximum economy.

From the foregoing tests, the economy gain in going from a 17:1 air-fuel mixture ratio to a 19:1 ratio was comparatively small, and does not give much promise of large mile-per-gallon gain. However, from our own viewpoint, this field needs more exploration. Yet, we must not overemphasize the notion of a leaner mixture, as the result that we are really seeking is the procurement of the maximum of useful power per unit of fuel and, when this is obtained, the matter of mixture ratio is one of no importance. Because of limiting conditions in our engines, such as the dilution of the fresh gasoline and air charge with residual

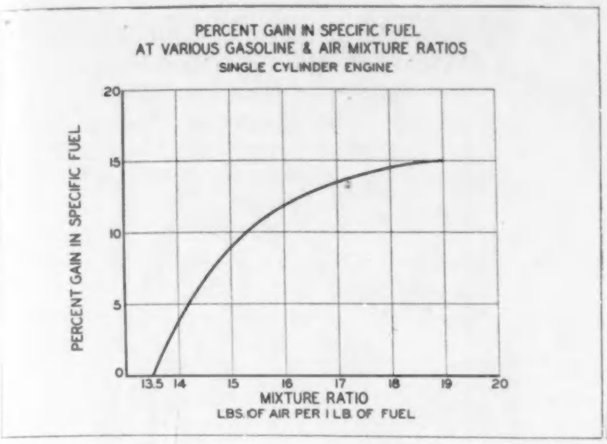


Fig. 16

exhaust gas in the combustion chamber, and the slow rate of burning of these lean mixtures, we have not been able to operate sufficiently satisfactorily on a multicylinder engine to obtain economy comparisons in the field of 18 or 19:1 fuel-air mixture ratios.

While, theoretically, complete combustion can occur with 15.4 lb of air to 1 lb of fuel, the burning of leaner than normal mixture ratios might result in a rise in thermal efficiency inasmuch as the temperature range of the cycle is reduced as the result of the lower mean specific heat of the gas. The heat loss to the water jacket will be reduced by the lower gas temperature, provided we can cause the combustion of such lean mixtures to be completed in the normal time period. Other than a very early spark, we have no further suggestions of how to accomplish the burning of lean mixtures in a normal rate of time in the conventional manner.

It would seem that the fuel-injection method, whereby the fuel-air mixture is localized in a comparatively small area at the point of ignition, rather than the mixing of the fresh charge as in our present turbulent system, may offer a means of running on leaner mixtures with gains in economy. Again, our lack of experience along these lines limits expression here to conjecture, rather than to factual data.

■ Spark Timing

In the course of these experiments it was found advantageous at the lean mixtures to advance the spark to timings ordinarily considered impracticable. Thus, at wide-open throttle with a mixture ratio of 19.2:1, the spark was advanced to 80 deg BTC for maximum power. This is indicative of comparatively slow burning of the lean mixtures and suggests that improved operation at lean mixtures might be obtained in practice with spark timings more advanced than the conventional settings at part throttle. Incidentally, the engine knocked at 19:1 with an 80-deg spark timing about as much as at 14:1 with 30-deg spark timing which was the maximum-power setting for that mixture ratio.

■ Exhaust Scavenging

Evidences of gains in road-load economy by means of improved scavenging are shown by the data on Fig. 17. In this case the better scavenging was obtained over at least a part of the speed range by using the so-called Y or split manifold which incorporates double exhaust pipes of lengths calculated to minimize resonance effects.

The residual exhaust mixture within the combustion chamber is one of the limiting factors for utilizing part-throttle lean mixtures to improve fuel consumption. This factor can be affected appreciably by the exhaust system as these experiments show. Exhaust gas present in the chamber displaces what might have been air and poisons or reduces the burnability of the mixture. Better scavenging reduces this dilution by the exhaust gas. Scavenging is affected, of course, by many other engine components – combustion-chamber and port shapes, valve timing, type of muffler, and so on.

■ Spark-Plug Gap Width

The effect of a wider than normal spark-plug gap in the conventional engine in promoting economy and steady operation over a wider range of air-fuel ratios is pictured on Fig. 18. Data presented here indicate the miles per gallon gain versus gap width – the effect decreases as the speed increases, there being practically no gain at 60 mph.

Oscillograph records show that the discharge across a wide spark-plug gap exists two to four times as long as the discharge across a narrow gap. Also, the type of discharge is multiple or oscillatory for the wide gap instead of dying out quickly as a single arc for the narrow gap. Under conditions of part-throttle engine operation when mixtures are lean, stratified, and varying in inflammability from cycle to cycle, the wide gap and its consequent type of dis-

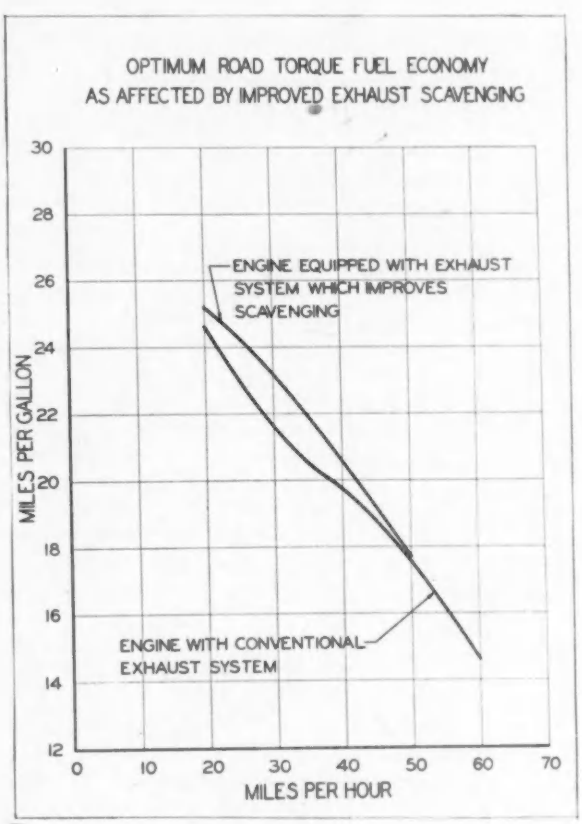


Fig. 17

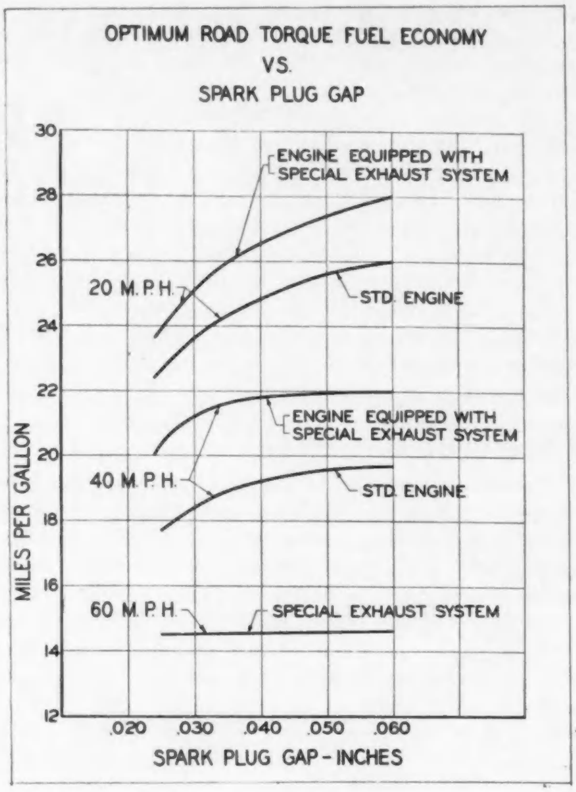


Fig. 18

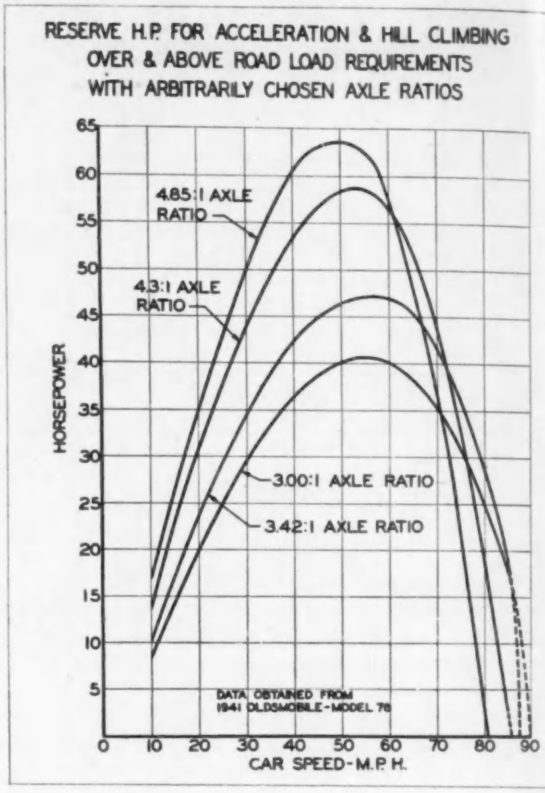


Fig. 20

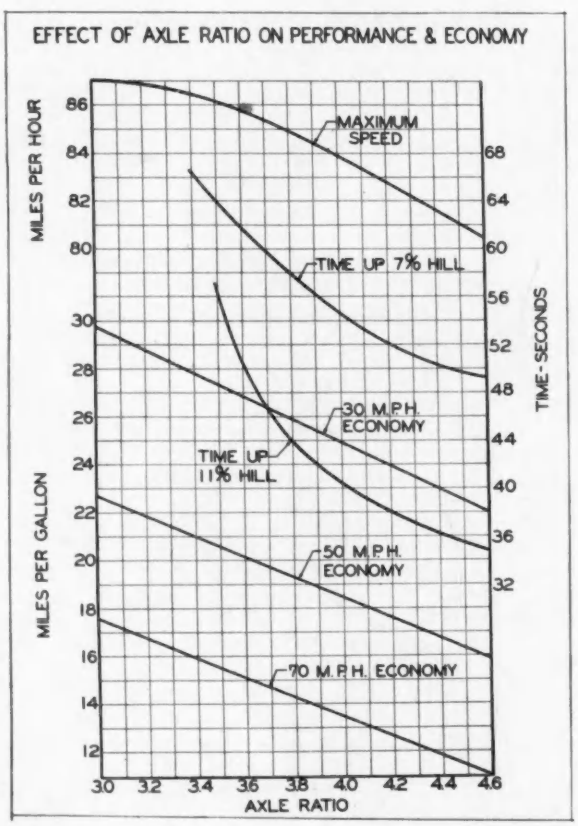


Fig. 19

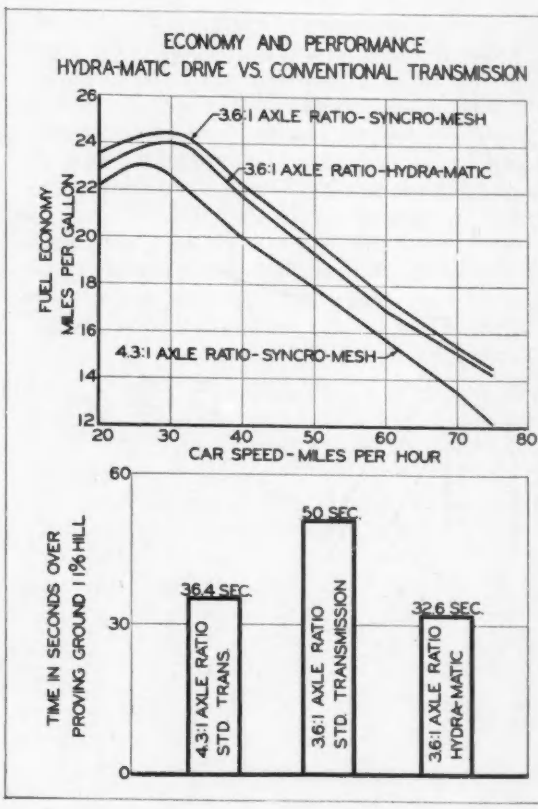


Fig. 21

charge provides a margin to increase the probability of ignition and to ignite not only the mixture in the gap at the beginning of discharge but any mixture that might appear in the gap space during the life of the discharge. In other words, the target is fired at with a shot gun instead of a rifle. Inasmuch as engine design still must effect a compromise between full-throttle power and part-throttle economy, there do exist conditions of part-throttle operation wherein lean mixtures, valve timing, exhaust back pressure, improper scavenging, stratification, and so on, seriously affect part-throttle economy. For these conditions, therefore, the wide-gapped spark plug holds a potential economy gain, at least on the particular engine under study.

The limiting influence here is the capacity of ignition systems to fire plug gaps of increased initial dimension, allowing for a normal further increase with usage. We are using 0.040-in. gap on our 6-cyl engine which, even then, necessitates a higher capacity coil than is generally used on 6-cyl engines.

The outcome of additional tests will determine the desirability of improved ignition capacity to handle a 0.060-in. spark gap.

Credit for much of the data presented here concerning mixture ratios, exhaust scavenging, and spark gaps is due to the General Motors Research Laboratory from whose reports this information was obtained.

■ Performance Factor

The relationship of reserve power for hill performance and rapid acceleration to gasoline economy is shown on Fig. 19, whereon is plotted the hill performance and economy with axle ratios of from 3.00 to 4.60. Here again is seen the detrimental effect on economy of high engine speeds obtained with such axle ratios as give sufficient hill performance or reserve performance factor for our present weight of car. A curve which we found interesting is shown on Fig. 20. Here is plotted the reserve horsepower available with different axle ratios over and above level-road requirements for acceleration and hill climbing. Automatic transmissions, such as the Hydra-Matic drive which permit of low engine speeds for cruising and yet a high performance factor without conscious manual or mental effort on the part of the driver, go a long way toward solving this phase of the economy problem. Fig. 21 shows the economy and performance comparison of an Oldsmobile 6-cyl conventional transmission car and the same car equipped with a Hydra-Matic transmission. Here, either increased economy or increased performance is available as the occasion warrants, without violating the American public demand for non-manual gearshift performance.

■ Summary

In general fuel consumption can be improved by:

1. Eliminating extraneous losses such as improved engine design or reduction of engine speed to improve its mechanical efficiency, and reduction of rolling resistance.

2. Higher compression and expansion ratio when suitable fuels are economically available, using the additional power thus gained to reduce engine speed.

3. Optimum exhaust scavenging to prevent excessive dilution of the comparatively small fresh charge obtained at light-throttle operation.

4. Adequate energy of spark to promote the firing of lean mixtures.

5. Best possible fuel atomization, delivering at part throttle such mixture ratios as are found to give maximum economy.

6. Ignition occurring sufficiently early to develop a maximum explosion pressure at such time during the pressure stroke as to be most effective.

The conclusion of this analysis points to reduction of engine speed, lighter-weight cars, and the economical and general availability of higher octane fuels to permit of higher compression ratios, as the real economy factors.

Aircraft Spotwelding Problems

THE spotwelding of aircraft materials (mainly aluminum alloys) is fundamentally more difficult than the spotwelding of automotive materials (mainly steels). The reason is that steel has high electrical resistance and low heat conductivity; hence passage of a moderate current will generate sufficient localized heat to make a good spotweld.

Aluminum alloys, on the other hand, have low electrical resistance and high heat conductivity; hence very heavy currents and pressures must be used. Even then, most of the generation of heat occurs at the surface of contact between the sheets being welded.

These conditions have imposed difficulties which have somewhat retarded the application of spotwelding in the aircraft industry. However, of recent years development has come very rapidly.

The increase in requirements for spotwelding equipment in turn brought to the fore a very serious problem – the magnitude of the power supply necessary for such equipment.

At this point a saving factor appeared in the form of the "stored-energy" type spotwelder first introduced into this country two years ago by a French firm. The "stored energy" spotwelder introduces a new principle – that of charging the transformer up slowly with magnetic energy, then discharging it quickly to make the weld. The charging is usually done through a three-phase rectifier. The demand on the power line imposed by the slow charging process is much lower than that which would correspond to the quick welding discharge. It is like filling a bucket from a small faucet and then throwing the contents suddenly on a fire.

Consistent maintenance of the proper pressure is essential to successful spotwelding because pressure controls the surface contact resistance. In many machines pneumatic force is used to produce electrode pressure and, if the air lines to the welder are too small, the air pressure will drop when the machines are operated rapidly, and mysterious troubles will ensue. It was found necessary to install a large air receiver near the welders to overcome this trouble. Hydraulic pressure systems should be a helpful improvement.

Probably the greatest single cause of spotwelding trouble is improper surface preparation. Oil, grease, or paint left on the surfaces to be welded produce cracked, burned, weak, or even "blown" spots. Careful cleaning is essential, and, in most cases, it is also necessary to remove part of the oxide film which always covers the surface of aluminum alloys.

Excerpts from the paper: "The Development of Aircraft Spotwelding," by Mabel M. Rockwell, production research engineer, Lockheed Aircraft Corp., presented at a meeting of the Southern California Section of the Society, Santa Monica, Calif., April 18, 1941.

STATIC FATIGUE

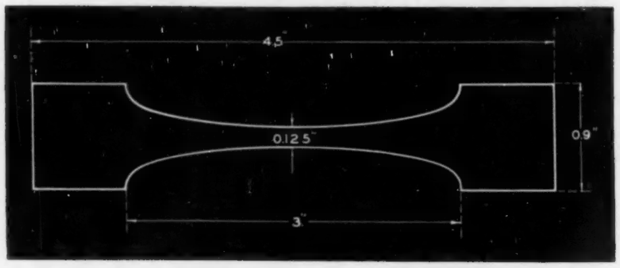
STATIC fatigue of rubber is defined by the authors as a progressive breakdown under the influence of a static load, whereas dynamic fatigue is defined as the progressive loss of strength due to successive cycles of stress. The static fatigue life is the time required for rupture under a static load.

Test data presented on the tension static fatigue of rubber indicate that the static fatigue lives of the samples are functions of the stresses acting on them; that the static fatigue lives fall off rapidly with increasing stresses; and that the dependents of static fatigue life on the stress is a function of the stock, among other things.

Curves of reduction of tensile due to static fatigue show that the tensiles of samples under load actually decrease and that the decrease is greater, the greater the time under load.

Referring to the effect of degree of cure, the data indicate that resistance to static fatigue decreases as degree of cure is increased beyond a certain optimum value which varies with the stock.

Other data show that large flow during cure in a particular region reduces the static fatigue resistance of that region in comparison with the remainder of the body of the stock; and that lateral pressure is highly beneficial in increasing the static fatigue lives of bonded rubber parts used in shear.



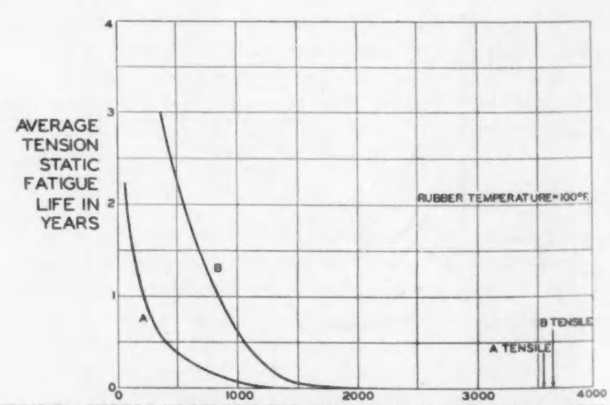
■ Fig. 1 – Sample used for determining tension static fatigue of rubber stocks

RUBBER is subject to two types of fatigue – dynamic and static. Its dynamic fatigue is the progressive loss of strength due to successive cycles of stress. Its static fatigue is, we *believe,* a progressive breakdown under the influence of a static load. We have already discussed the dynamic fatigue of rubber.[1] This paper deals with the static fatigue of rubber.

The load required to break a rubber sample in a test machine is a function of the speed at which the rubber is loaded to break, being greater, the greater the speed. If a rubber sample is kept under a constant load slightly less than its ordinary test-machine breaking load, the sample will break in a very short time due to the high static stress. If various smaller loads are placed on similar samples, it is found that the static fatigue life – the time required for rupture – varies with the load. In other words, the load required at any particular time to break a sample decreases with time as that sample is kept under a static load. Ultimately the load required to break the sample decreases until it becomes equal to the load which is acting on the sample, and rupture results. The phenomenon of rupture after a period of time under static load is referred to as static fatigue.

■ Tension Static Fatigue of Rubber

The way in which the tension static fatigue of rubber depends upon loading can be indicated more clearly by a few tension static fatigue curves. Fig. 1 shows the type of sample we used in obtaining the greater part of our data on tension static fatigue of stocks. The sample has a uniform width of 1/8 in. for a length of approximately 1/2 in. at the center. This shape of sample was chosen to make sure that the great majority of samples would break at or near the center. The samples were cut from cured slabs of rubber by means of a die. In what follows, stresses on the samples are expressed in pounds per square inch of *original*



■ Fig. 2 – Tension static fatigue data for two typical U. S. Rubber stocks at 100 F

[This paper was presented at a meeting of the Detroit Section of the Society, Detroit, Mich., March 31, 1941.]

[1] See *Industrial and Engineering Chemistry,* Analytical Edition, Vol. 12, Jan. 15, 1940, pp. 19-23: "Dynamic Fatigue Life of Rubber," by S. M. Cadwell, R. A. Merrill, C. M. Sloman, and F. L. Yost.

LIFE of RUBBER

by S. M. CADWELL, R. A. MERRILL, C. M. SLOMAN, and F. L. YOST
United States Rubber Co.

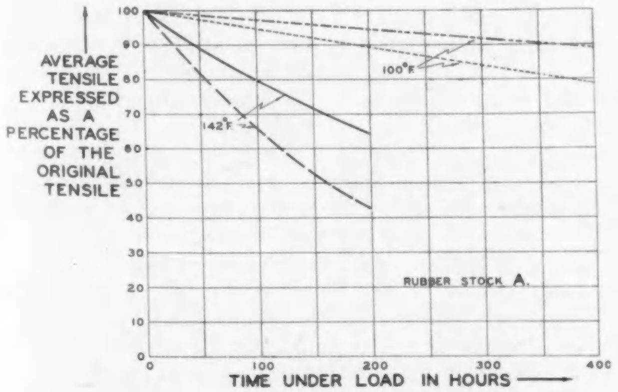
area at the center of the sample, the stress for each sample being corrected for the actual gage of the sample at its center. In a tension static fatigue test each wide end of a sample was clamped in a special jaw. One jaw was then attached to a support and the other jaw supported a load. The sample was kept at a constant temperature and the time to failure was noted.

Fig. 2 shows tension static fatigue data for two typical U. S. Rubber stocks at 100 F. Stock A is a 54 shore durometer (type A) stock whose average tensile for the cure used is 3540 psi of original area. Stock B is a 39 durometer stock whose average tensile is 3650 psi of original area. The stresses at the centers of the samples in pounds per square inch of original area are plotted as abscissae. The average tension static fatigue lives of the samples in years are plotted as ordinates. These curves indicate that the static fatigue lives of the samples are functions of the stresses acting on them; that the static fatigue lives fall off rapidly with increasing stresses; and that the dependence of static fatigue life on the stress is a function of the stock, among other things. These curves do not show what happens beyond stresses of 1500 to 2000 psi of original area. Actually the static fatigue lives become smaller and smaller for the higher stresses, down to minutes or seconds in the region of the tensile value; and in the complete course of the curves B is superior to A.

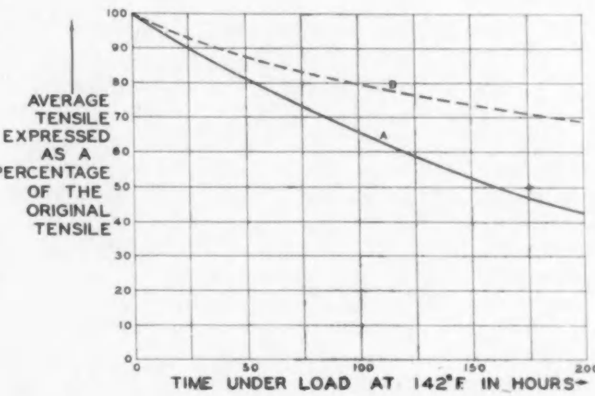
■ Reduction of Tensile Due to Static Fatigue

This sort of dependence of static fatigue life on stress is to be expected from the fact that average tensiles of samples which have been supporting loads for some time are lower than the average tensile of control samples which have not been supporting loads. Fig. 3 shows how the average room-temperature tensiles of samples of stock A are affected when the samples are kept under load at 100 F or 142 F for different periods of time. In getting these data groups of samples were subjected to the same loads. At different time intervals a group was unloaded and the average room temperature tensile of that group was determined. Times under load previous to tensile testing are plotted as abscissae in Fig. 3. The average room-temperature tensiles of the samples expressed as percentages of the corresponding tensiles of samples which were not subjected to stress are plotted as ordinates. The upper of each pair of curves corresponds to a stress of about 15% of the original average tensile of the samples; the lower, to a stress of about 30%. The actual loads on the samples were 5 lb and 10 lb. Average tensiles of control samples subjected to the same treatment except not strained did not show any significant changes.

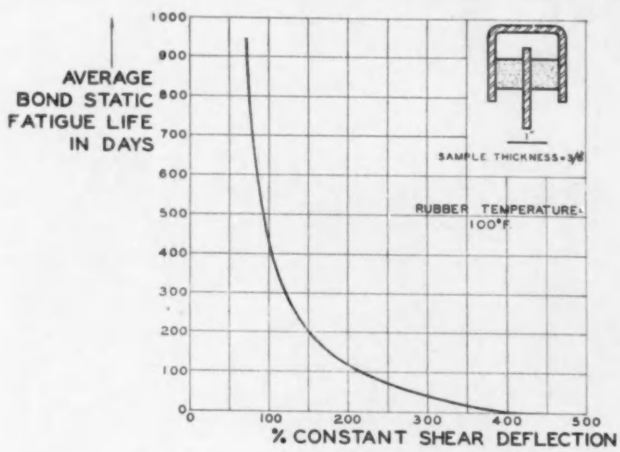
These curves show that the tensiles of samples under load actually decrease and that the decrease is greater, the greater the length of time under load. They also show that static fatigue takes place much more rapidly at high temperatures than at low. No curves are given in this paper showing static fatigue as a function of temperature, but Fig. 3 shows how important temperature is in static fatigue. We have many other comparison tests showing that static



■ Fig. 3 – Effect on average room-temperature tensile strength of samples of Stock A when the samples are kept under load at 100 F or 142 F for different periods of time



■ Fig. 4 – Relative loss in tensile strength for Samples A and B stressed at about 32% of their tensiles at 142 F



■ Fig. 5 – Static fatigue of rubber-to-metal bond for a stock with a shear modulus of about 80 psi

fatigue life falls off rapidly with increase of temperature above room temperature.

Fig. 4 shows the relative loss in tensile for samples of U. S. Rubber stocks A and B stressed at about 32% of their tensiles at 142 F. The comparative behavior of the two curves is in agreement with the actual static fatigue curves already shown, that is, A is worse than B.

■ Rubber-to-Metal Bonds in Shear

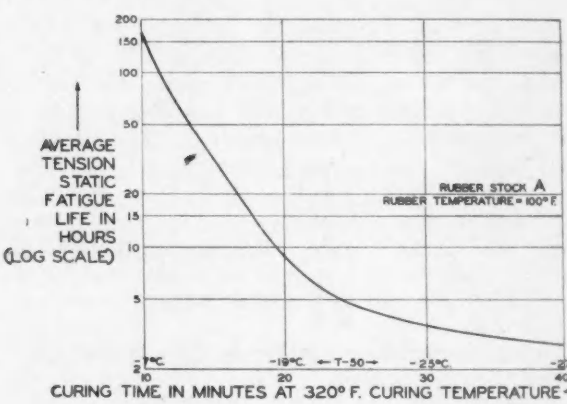
Static fatigue occurs not only in rubber stocks but also in the neighborhood of rubber-to-metal bonds – in the region where we have the transition from body stock to skim coat to cement to brass-plated metal. We have been accustomed to refer to a failure in this region as a "bond" failure, even though a thin layer of rubber remains coating the metal; and that is the sort of break we have in mind when talking about "bond" failures in what follows. Fig. 5 is a plot of data for static fatigue of the bond for a stock with a shear modulus of about 80 psi used in the type of shear sandwich shown. During the tests the samples were kept at 100 F and at various constant shear *deflections*. The average static fatigue lives of the bonds in days are plotted versus the percentage constant shear deflection.

The curve for static fatigue life of the bond in Fig. 5 resembles those for static fatigue life of the stocks. The static fatigue curves for either rubber stocks or rubber-to-metal bonds and for all types of deformations have this general form whether plotted as functions of constant stress or of constant deflection. For any particular case the actual magnitudes of the static fatigue lives – which may be greater or less than those shown in Fig. 5 – depend on many factors: the shape of the unit, the stock, the cure, the rubber-to-metal bond, the temperature of the rubber and the type and amount of deformation. For example, there was no lateral pressure on the rubber in the units from which data were taken to plot the curve in Fig. 5. As will be pointed out later, if the rubber had been under 10% to 20% lateral compression strain during test, the average static fatigue lives of the bonds would have been very much greater.

■ Effect of Degree of Cure

Fig. 6 shows how the static fatigue of Stock A varies with the degree of cure. Times of cure in minutes for a curing temperature of 320 F are plotted as abscissae; and the T-50's – which are a measure of the degree of cure – are indicated for certain cures. The average static fatigue lives of the samples in hours are plotted as ordinates along a logarithmic scale. Under test these samples were kept at 100 F and were stressed at about 1570 psi of original area at the center, which corresponds to 44% of the tensile for normal cure (maximum tensile). Resistance to static fatigue decreases as degree of cure is increased beyond a certain optimum value (this optimum value varies with the stock). In particular, the long-time static fatigue lives for slight under-cures are many times greater than those for normal cures or over-cures. Of course, for degrees of cure low enough for the stock to be practically raw, the static fatigue lives would fall off to negligible values. This dependence of static fatigue on cure is an important consideration, because it is often the practice to slightly over-cure stocks in order to reduce their drift in use. Such over-cures unquestionably reduce the static fatigue resistances of the stocks.

This is also shown by Figs. 7 and 8. Fig. 7 shows static



■ Fig. 6 – Effect of degree of cure on static fatigue of Stock A

fatigue curves for three U. S. Rubber Stocks A, B, and C. When normally cured, C is of about the same durometer and tension modulus as Stock B; and, when normally cured, C would be expected to have about the same static fatigue lives, or slightly better. However for these tests C was very seriously over-cured (T-50 ~ 33 C). As a result, its static fatigue resistance is much inferior to that of B. The static fatigue of Stock C, which is an over-cured 40-durometer stock, is a little better than that of normally cured stock A, which is a 54-durometer stock. Fig. 8 shows that C loses tensile more rapidly under stress than does B. Actually C is stressed to only about 19% of its tensile at test temperature whereas B is stressed to about 32% of its tensile at test temperature. C lies well above A in Fig. 8 due to the fact that the degree of loading for C is considerably less than that for A.

■ Effect of Flow During Cure

Another feature which can affect static fatigue of the stock is excessive flow during cure. To test this factor we

cured large slabs in molds designed so that flash flow during cure took place either through 1/32-in. flash holes down the midline of each slab and on both faces or through 1/32-in. flash slits along the midline of each slab and on both faces. Slabs cured had either slight projections or a long ridge along one centerline. Test samples were cut so that the projections or ridges were at the center of the sample. Control samples were cut from other portions of the slab not near the flow vents. Table 1 shows for Stock A the effect of flow introduced in this manner.

Large flow in a particular region reduces the static fatigue resistance of that region in comparison with that of the rest of the body of the stock. It is impossible to cure rubber without some flow, but regions near flash vents will represent volumes of greatest flow and parts should be designed so that such regions will not be subjected to the maximum stresses.

Effect of Lateral Pressure

These considerations carry over into the stock in bonded parts; but there are other things which can affect the static fatigue of bonds. One of them is lateral pressure on shear units (that is, pressure normal to the bond faces). Fig. 9 shows one type of sample used in these tests. One annular plate was held fixed and, by means of a weight attached to a lever arm on the other plate, the unit was put in rotational shear. The effect of lateral pressure on these units is shown by the data in Table 2 which are for tests which were run at 100 F.

For stock A the static fatigue life of the bond at a stress of 290 psi – which is enormous compared to stresses in practice – is manyfold greater at high lateral pressures. The data for synthetic stock D show increases in static fatigue life of the bond resulting from lateral pressure. The synthetic data partially answer another question which is frequently asked: Does occasional vibration improve static fatigue? The samples marked (v) were vibrated 500 times each week through 25 deg to either side of their equilibrium deflections. In this case as in all our other trials we find that the vibrated samples fail first.

Table 3 shows the same general results for another U. S. Rubber stock, E, used in large-scale annular rotational units.

TABLE 1

Test Temperature = 78 F
Average Stress = 2000 psi
= 56% of average tensile of controls.

Feature	Average Static Fatigue Life, min
Control	52
1/32-in. flash hole	16
1/32-in. flash slit	15

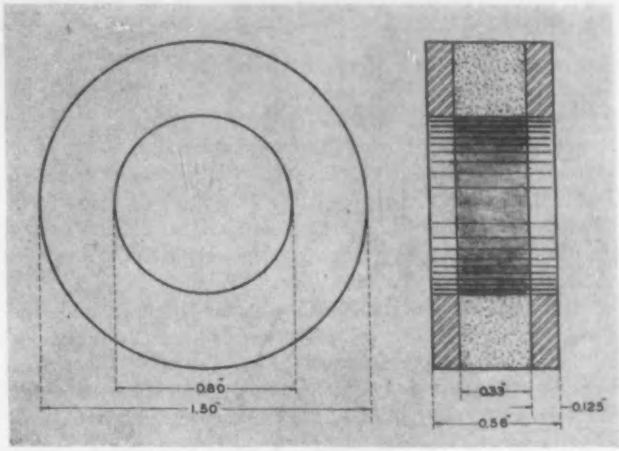


Fig. 9 – Sample used in tests of effect of lateral pressure on static fatigue of rubber-to-metal bonds in shear

This stock shows poor static fatigue of the bond compared to others that we have tested, but that does not detract from this particular test. In this as in all of our tests we find that lateral pressure is highly beneficial in increasing the static fatigue lives of bonded rubber parts used in shear.

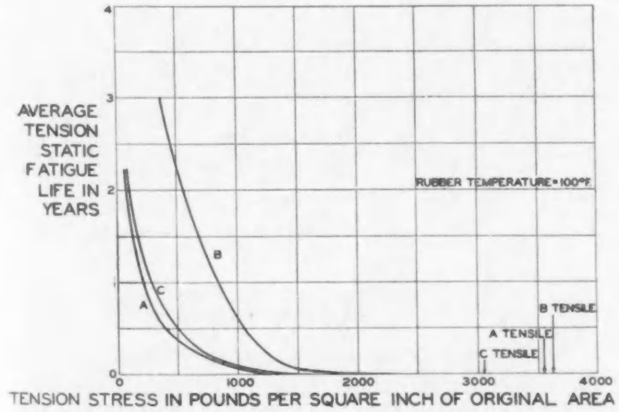


Fig. 7 – Static fatigue curves for three U. S. Rubber Stocks, A, B, and C

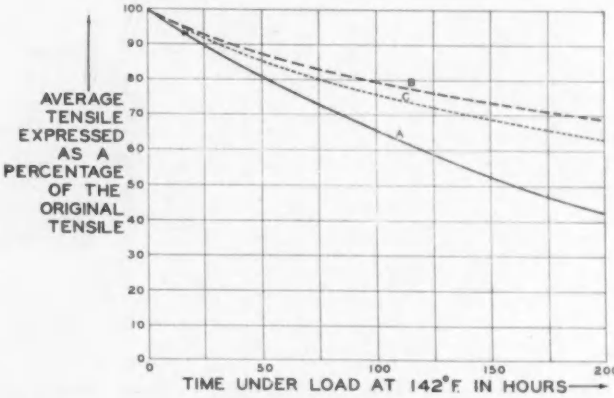


Fig. 8 – Average tensile expressed as a percentage of the original tensile versus time under load at 142 F for Stocks A, B, and C

TABLE 2

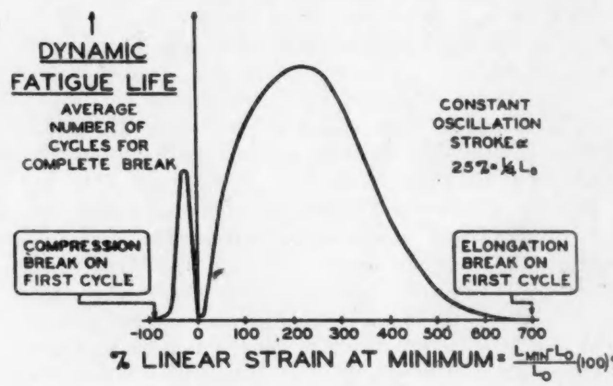
Type Number	Stock	Torque, lb-ft	Maximum Shear Bond Stress, psi	Average Initial Deflection, deg	Average Percentage Lateral Compression on Stock Thickness	Average Static Fatigue Life
1	A	14.8	290	93	9	10 hr
2	A	14.8	290	87	15	> 250 d.
3	A	14.8	290	80	21	> 250 d.
4	A	14.8	290	80	27	140 d.
5	D (synthetic)	3.5	70	24	0	139 d.
6	D (synthetic)	3.5	70	22	0 (v)	24 d.
7	D (synthetic)	3.5	70	22	10	175 d.
8	D (synthetic)	3.5	70	23	10(v)	85 d.

Tension Static Fatigue of a Synthetic Stock

Synthetic rubbers, like natural rubber, show the phenomenon of static fatigue. Fig. 10 shows the curve of reduction of tensile with time under stress at 142 F for synthetic stock D. The samples were stressed at about 26% of their tensile at 142 F. It is important to remember that, for these synthetic samples, such an actual stress is much smaller than would be the case for rubber. This is due to the fact that, in going from room temperature to 142 F, the tensile for rubber will not decrease below 80% to 90% of its room-temperature values whereas the tensiles of the synthetic samples with which we worked fell to about 10% of their room-temperature values. Accordingly the reduction-in-tensile curves shown previously for rubber were for samples supporting loads of 5 or 10 lb, whereas this curve is for synthetic samples supporting loads of about 10 oz. This radical reduction in tensile with increase of temperature must be of importance in the fatigue of units designed to work at high temperature. In cases of use at room temperature where cracking could lower the fatigue life, this synthetic would be expected to be somewhat better than rubber. Otherwise, it should be about the same at room temperature.

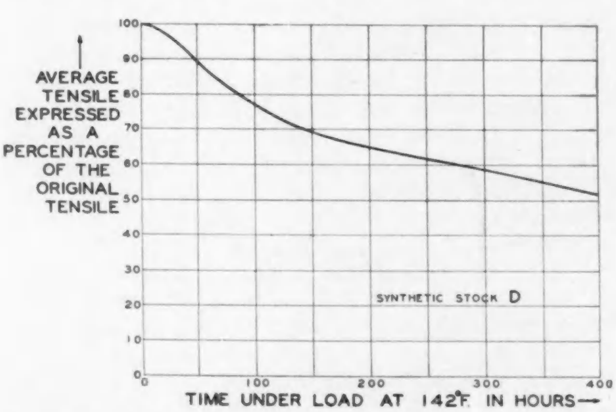
Fatigue Life of Rubber – Static Plus Dynamic

Fig. 11 shows how the dynamic fatigue life of rubber being vibrated linearly with a constant oscillation stroke depends on the percentage linear strain in the sample at its minimum length in the oscillation cycle. The important point is that, near the condition of zero minimum strain, the dynamic fatigue life has a minimum which is bounded by maxima in both the compression and extension regions. In this particular curve the dynamic fatigue life is expressed as a certain number of vibrations required to produce failure.



■ Fig. 11 – Dynamic fatigue life of rubber being vibrated linearly versus percentage linear strain in sample at minimum length in oscillation cycle

In practical applications both static and dynamic fatigue must be considered. The considerations involved can be shown by Fig. 12 in which the static and dynamic fatigue life curves for a hypothetical stock are superimposed. The stresses on the samples in pounds per square inch of original area are plotted in arbitrary units as abscissae. It is to be understood that the same scale is used for both the static and dynamic cases. The type of dynamic fatigue

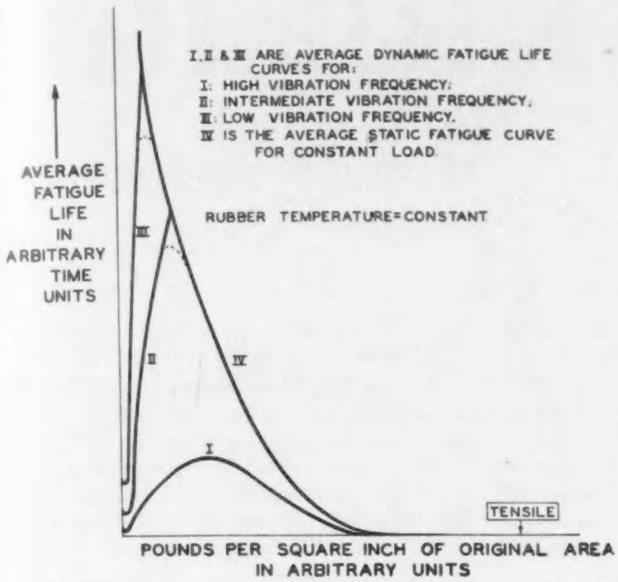


■ Fig. 10 – Reduction of average tensile with time under stress at 142 F for Synthetic Stock D

TABLE 3

Bond Stress = 78 psi
Test Temperature = 100 F

% Lateral Compression on Rubber During Test	Average Static Fatigue Life
0	68 d.
10	> 370 d.
20	> 370 d.



■ **Fig. 12 – Superimposed static and dynamic fatigue life curves for a hypothetical stock**

considered is that in which constant loads are suspended from one end of a sample and the other end is rapidly vibrated. The fatigue lives for either the static or dynamic case are plotted in arbitrary time units as ordinates.

The static fatigue curve is of the type already discussed. The three dynamic fatigue curves represent cases in which, for any loading condition, the conditions of vibration are identical *except* that each curve refers to a different vibration frequency. Therefore, points corresponding to the same loading condition represent the same dynamic fatigue lives in *cycles* but different lives in point of *time*. The combined curves show that there is a limit to the effective fatigue life which can be obtained by increasing the minimum stress to get good dynamic fatigue. That limit is represented by the junction of the dynamic and static fatigue curves. Actually, the combined effect of static and dynamic fatigue would cause the curves to follow the indicated dashed lines. It is clear that there is ordinarily a limit to the time fatigue life which can be obtained for rubber in vibration by increasing the minimum stress; that this time limit is roughly the time corresponding to the intersection of the dynamic and static fatigue curves; and that, the lower the frequency of vibration, the lower is the stress at which this intersection takes place.

The Automotive Engineer in Britain

THE work of the automotive engineer in Britain goes on and his problems are increasing. Miles per gallon became a demand with the rationing of fuel to the public. There rose a howl for more miles per drop of fuel. We got down with the carburetor, spark plugs, compression ratio, axle ratio, and so on, but the most important method was by teaching folks how to drive a car. This was not as difficult as it sounds. We put an electric contact on the carburetor and a red light lit up on the dash and, if you drove by the light, you could save 15% and sometimes more. Then the army demanded better consumption for military trucks. It was not possible for a while because certain high ranking officers had picked out the carburetors for everyone to use. If that happens here, you had better get your shotguns out quick. When these gentlemen were given other things to do and civilian experts were put on the Mechanization Board, it was a simple matter to bring the carburetor up to date and save 18%. Since this affected many thousands of military trucks, the saving was enormous – 15,000 gal per day, and yet it took a fight to see it through.

The winter of 1939 and 1940 was the coldest Britain has had for 50 years. Then it was we were able to make those men understand you could not use SAE 60 oil in engines for road vehicles. It sounds funny, doesn't it? It wasn't a bit funny trying to sell it. But we found ourselves saddled with the problem of obtaining zero starting for all tanks and trucks. This was an unknown feature for English vehicles. This job fell in the laps of the civilian members of the Board. Only because of our American experience we could step up to it and modify all carburetors to a predetermined formula and obtain the required result.

Then we were asked to meet for trucks, the problem of fuel restrictions. Here we were told to develop gas producers. In this field we found more liars per acre than in any endeavor that we tackled. We were supposed to pick out existing units and make it. We eventually had to develop a gas producer, even to burn coke. The chemical activated the coke so the producer could be started. Then that ate up the spark plugs and valves in 10 hr. Then we had to develop many filters to stop sodium gas. It was done.

All that work was being done in strides by engineers with many other things to do and under bombing conditions. Quite often work papers were picked up and carried to shelters where the work continued. The number of problems that each one must work on at one time is enormous and you cannot look over a group of technical applicants to see who has the necessary experience and hire him. You have got to learn as you go, consult where you can, and believe nobody. Reduce your problem to practice, test it as thoroughly as possible and move on. Thus you might find an acoustic engineer, with an engineering journalist as a partner, carrying on the development of a gas producer and filters and radios and telescopes and gun mountings. Sure, they make mistakes. It is like asking for a construction that could not be made. That is the job of supervision. Department heads have to assume a few additional responsibilities too. They must see to it that the proper foundry, forge, and machine shop information is available. Thus everybody grows together and no problem is beyond the scope of a live organization. Intelligence and confidence combined does solve problems. Intelligence and confidence must be a two-way stretch between engineering and engineering management. The engineer must engineer and the engineering management must manage. Each has enough to do without trying to do the other fellow's job.

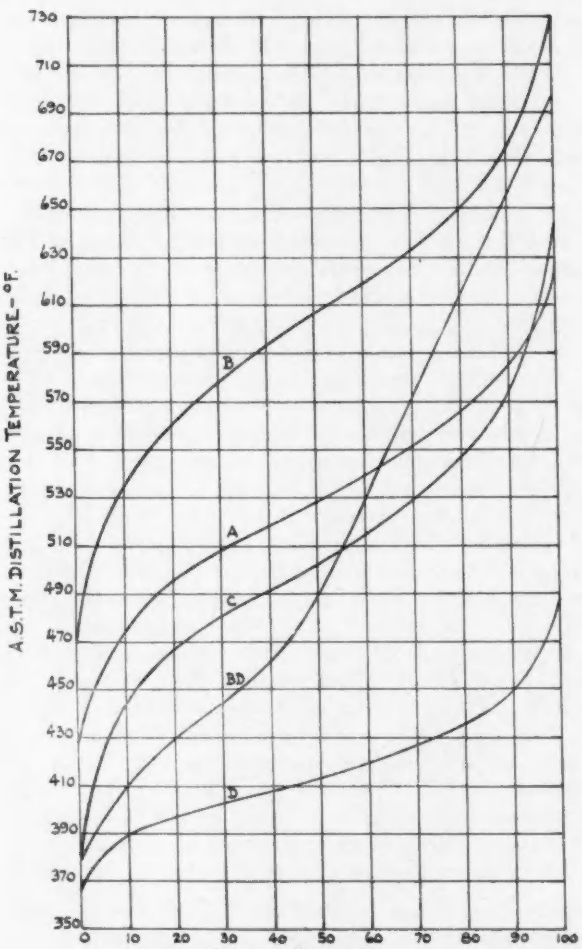
Excerpts from the paper: "Inside that Fortress with Britain's Automotive Engineers," by Alex Taub, British Purchasing Commission, presented at a meeting of the Metropolitan Section of the Society, Detroit, Mich., May 19, 1941.

Evaluation of DIESEL FUELS

(Report of the Cooperative

THE Automotive Diesel Fuels Division of the CFR Committee was organized in September, 1937, at the request of The Volunteer Group for Compression Ignition Fuel Research, and has continued the activity of that group. Though some fuel testing had been done with full-scale diesel engines[1] in the development of the CFR Diesel test method, it was felt that a separate group was necessary in order properly to carry out the investigation of fuel properties as they affected full-scale engine performance.

The Full Scale Engine Group was organized for the purpose of clarifying the issue of automotive diesel fuel specifications by determining those properties of analyses which influenced engine performance and engine deposits.



■ Fig. 1 – ASTM distillation – Full scale on test fuels

THE CFR Full-Scale Engine Group was organized by engine men representing the major classifications of automotive-type diesel engines. This original group, with the cooperation of the representatives of the petroleum industry, the government agencies, and colleges, has carried on extensive series of tests on commercial diesel engines.

The purpose of the investigation was to determine the influence of fuel properties such as cetane number, viscosity, volatility, and gravity on the engine performance. The following relationships are indicated:

Starting and engine smoothness are dependent upon the ignition quality of the fuel.

Smoke and engine deposits vary with the volatility and viscosity.

Exhaust odor varies with ignition quality and cetane number.

Power output and fuel consumption vary with the heat value of the fuel.

At a given pump setting, viscosity may have an independent effect on the power output because of plunger leakage.

The limited number of fuels used in this series of tests made it impossible to isolate all fuel properties and their effect on fuel performance, but the results obtained will guide future investigations.

The group was formed with the primary membership consisting of engine men who represented the four essential classifications of automotive-type diesel engines.

At least two engine men were selected to represent each engine type as well as a representative from the U. S. Naval Experiment Station at Annapolis. These engine men met at Peoria on July 26, 1939, and agreed to conduct prescribed tests on multicylinder diesel engines of their own manufacture selected with characteristics within the following limits: bore 3½ to 4½ in.; number of cylinders 4 to 6; speed 1500 to 2000 rpm. The factors selected as essential to the investigation of fuels were: (1) cetane; (2) volatility; (3) viscosity and (4) gravity, and their effect on (1) starting; (2) smoothness; (3) smoke; (4) power output;

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 5, 1941.]

[1] See SAE Transactions, January, 1938, pp. 27-36: "Report of the Volunteer Group for Compression-Ignition Fuel Research," by C. H. Baxley and T. B. Rendel.

in Full-Scale Engines

Fuel Research Committee)

presented by W. G. AINSLEY

(5) fuel consumption; (6) smell and (7) engine deposits.

The fuels required for this investigation were to be straight-run products to cover a range of cetane number, volatility, viscosity, and gravity. Four fuels subsequently were specified, and a Fuels Group appointed to arrange for the manufacture and distribution of these reference fuels. The laboratory inspection data on those fuels are given in Table 1, and ASTM distillation curves are shown in Fig. 1.

An Instrumentation Group was selected for the purpose of setting up instrumentation and procedure in order to standardize the methods of test employed by the various laboratories. This group functioned throughout the entire investigation.

A secondary group, consisting of representatives from the petroleum industry who have engines in their laboratories of the types of manufacture as represented by the engine men of the primary group, either single or multi-cylinder structures, was invited to participate. The petroleum men were to run tests as prescribed for and by the engine men and to contribute supporting data to the respective representatives of the engines manufactured by the Engine Men Group.

Representatives from colleges and government agencies were invited to test the reference fuels according to the prescribed tests and to contribute data to the group.

Table 1 – Laboratory Inspection Data on Reference Fuels

Fuel	A	B	C	D	50% B 50% D
Fuel Characteristic	High Cetane No.	Low Volatility	Low Cetane No.	High Volatility	Blend
Ignition Qualtiy					
(A) Diesel Index Number	61.5	37.6	36.9	55.6	46.0
(B) Cetane Number	56.1	40.5	33.2	41.3	40.8
Viscosity at 100 F					
(A) Kinematic, centistokes	3.24	7.81	3.33	1.54	3.04
(B) S.S.U., sec	37	51	37.5	31	36
Distillation, F					
(A) Initial Boiling Point	427	468	380	367	379
(B) 10% Boiling Point	475	535	445	390	410
(C) 50% Boiling Point	527	607	502	413	489
(D) 90% Boiling Point	584	672	574	449	650
(E) End Point	625	729	642	487	704
Gravity, deg API	37.6	26.6	29.1	40.8	33.4
Flash, PM, F	206	238	168	157	170
Pour Point, F	0	20	B-10	B-10	−5
Water and Sediment, %	0	Trace	0	0	Trace
Corrosion					
(A) 3 hr at 130 F	Neg	Neg	Neg	Neg	Neg
(B) 3 hr at 212 F	Neg	Neg	Neg	Neg	....
Total Sulphur, %	0.13	0.60	0.11	0.07	0.36
Carbon Residue, % (On fuel)	0.014	0.023	0.01	0.014	0.027
Carbon Residue, % (10% Btms.)	0.03	0.21	0.04	0.027	0.139
Ash, %	0	0	0	0	0
Color					
(A) ASTM	1	5−	1½−	....	....
(B) Saybolt	−10	....	....	+19	
Heat Content					
(A) Btu/Gal	137,945	143,347	142,186	135,452	
(B) Btu/Lb	19,763	19,164	19,345	19,571	
* Modified ASTM Gum	32.0	925.0	66.0	6.0	
* Hercules Residue (Mg/100 G)	16.0	139.5	26.5	10.0	

* Average of two laboratories

The 16 companies which participated in the full-scale program are listed below:

A. Engine Companies
   1. Caterpillar Tractor Co.
   2. General Motors Corp.
   3. Fairbanks, Morse & Co.
   4. Chrysler Corp.
   5. Mack Manufacturing Corp.
   6. Waukesha Motor Co.

B. Refining Companies
   1. Shell Oil Co.
   2. The Texas Co.
   3. Standard Oil Co. (Indiana)
   4. Pure Oil Co.
   5. Standard Oil Development Co.
   6. Sinclair Refining Co.
   7. Atlantic Refining Co.
   8. Socony-Vacuum Oil Co.

C. Associated Groups
   1. University of Wisconsin
   2. The Pennsylvania State College

A progress report was presented at the Summer Meeting of the Society of Automotive Engineers at White Sulphur Springs, West Va., June 11, 1940, by Chairman C. G. A. Rosen. The data used as a basis for that report, along with additional data submitted after that date, were used as a basis for this report.

■ Personnel of Analysis Group

An analysis group was selected from those members most familiar with the data submitted. The personnel of this group was made up of the five representatives of the diesel engine manufacturers who submitted reports and five representatives of the oil refiners active in the program. As instructed by the Full-Scale Group at the meeting in Detroit, Jan. 9, 1941, this group held a two-day meeting and: (1) prepared recommendations on the automotive diesel fuel characteristics as indicated by the full-scale engine research test data and the laboratory and field experience of those present; (2) made a preliminary study of the data available and formulated a general plan and program for the more detailed analysis. A detailed report of the recommendations on the required diesel fuel characteristics is given in Appendix A.

In carrying out the actual work of studying and of evaluating the data it was necessary to enlarge the original

group. The personnel of the group responsible for the formulation of this report was as follows:

W. G. Ainsley, Sinclair Refining Co.
W. F. Aug, Mack Manufacturing Co.
A. J. Blackwood, Standard Oil Development Co.
K. M. Brown, Caterpillar Tractor Co.
W. H. Browne, Caterpillar Tractor Co.
J. M. Campbell, General Motors Corp.
C. E. Cummings, The Texas Co.
W. E. Drinkard, Chrysler Corp.
J. O. Eisinger, Standard Oil Co. (Ind.)
A. H. Fox, Standard Oil Co. (Ind.)
H. M. Gadebusch, Detroit Diesel Engine, Division, General Motors Corp.
L. W. Griffith, Shell Oil Co.
L. E. Hebl, Shell Oil Co.
S. L. Henry, Fairbanks, Morse & Co.
C. G. A. Rosen, Caterpillar Tractor Co.
G. C. Wilson, University of Wisconsin
H. D. Young, Sinclair Refining Co.
G. W. Zabel, Fairbanks, Morse & Co.

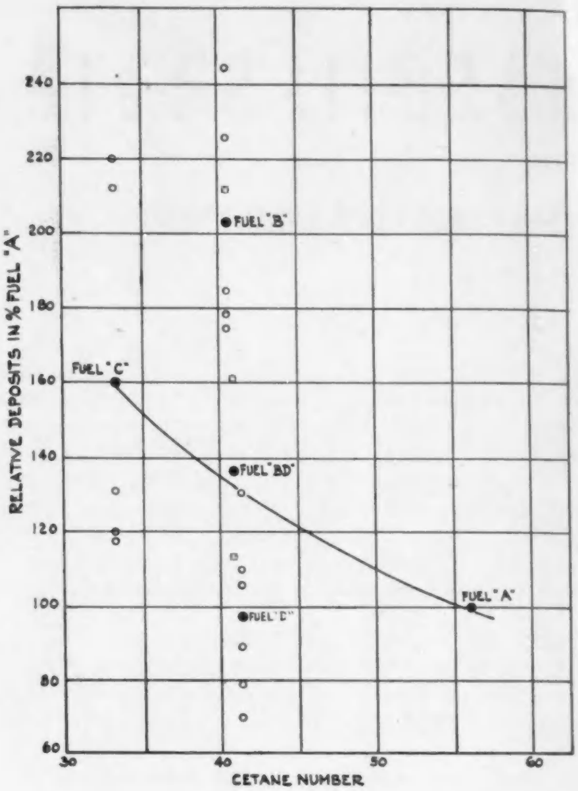
Reports were received on engine testing as conducted in 10 laboratories employing 15 individual engines of 7 different makes. The data on the engines used in these tests are given in Table 2.

Since data available in this report were gathered from several engines by different contributing laboratories, the problem of correlation was a difficult one. The reported results had to be brought to a common basis before any fuel property influence could be analyzed.

The fact that the fuel properties are interrelated and that only four basic reference fuels and cross blends were investigated made the interpretation of the results in terms of fuel properties a task which in some cases was impossible to solve at this time. To facilitate the correlation of engine test data with fuel properties, all specific values were converted into percentages relative to one of the fuels.

■ Engine Deposits

The engine combustion-chamber-deposit test method was intended to be used as a means of comparing fuels by determining the combustion-chamber deposit-forming tendencies under operating conditions simulating 5000 ft elevation. The engine was operated at no load and rated speed continuously for 50 hr on the fuel under test. The intake air was throttled to 5 in. hg depression during the entire test (this is the equivalent to operation at approximately 5000 ft altitude). Water and oil temperatures were maintained at normal levels.



■ Fig. 2 – Relative deposits versus cetane number

Table 2 – Reports on Engine Testing – 15 Engines, 7 Makes

Engine	Company Reporting	Bore, in.	Stroke, in.	Number of Cylinders	Compression Pressure, psi	Compression Ratio	Combustion System	Injection System	Displacement, cu in. Total	Displacement, cu in. Per Cylinder	Rated Brake Mean Effective Pressure, psi	Speed Range, rpm
General Motors Corp. 3-71	Texas	4¼	5	3	500	19/1	Open	Own	213	71	57	500-2,000
General Motors Corp. 3-71	Atlantic	4¼	5	3	500	19/1	Open	Own	213	71	57	500-2,000
General Motors Corp. 3-71	Standard Oil Development	4¼	5	3	500	19/1	Open	Own	213	71	57	500-2,000
General Motors Corp. 1-71	Shell	4¼	5	1	500	19/1	Open	Own	71	71	57	500-2,000
Hercules DKXB	Standard Oil Development	3½	4½	6	...	...	...	...	...	...	...	...
Fairbanks, Morse & Co.	University of Wisconsin	4¼	6	1	580	16.8/1	Precombustion chamber	Bosch	85	85	77	500-1,200
Fairbanks, Morse & Co.	Shell	4¼	6	1	580	16.8/1	Precombustion chamber	Bosch	85	85	77	500-1,200
Fairbanks, Morse & Co.	Fairbanks-Morse	4¼	6	4	580	16.8/1	Precombustion chamber	Bosch	340	85	77	500-1,200
CFR	Standard Oil (Ind.)	3¼	4½	1	...	Variable	Precombustion chamber	Bosch	37.5	37.5	..	600-2,000
Dodge T106	Chrysler	3¾	5	6	450	14.7/1	Turbulence chamber	....	331	55	88	500-2,600
Dodge	Standard Oil Development	...	..	..	...	...	...	...	...	...	..	...
Dodge TKD	Atlantic	3¾	5	6	...	14.5/1	Lanova	....	331	55	..	...
Mack	Mack	4	5⅜	6	...	14.6/1	Lanova	Bosch	405.3	67.5	100	...
Caterpillar E48	Caterpillar	3¾	5	4	745	18.5/1	Precombustion chamber	Own	221	55.2	75.2	600-1,650
Caterpillar 1A138	Caterpillar	4¼	5½	1	695	17/1	Precombustion chamber	Own	78	78	80	600-1,500

Table 3 – Engine Deposit Data

Total Weight, g

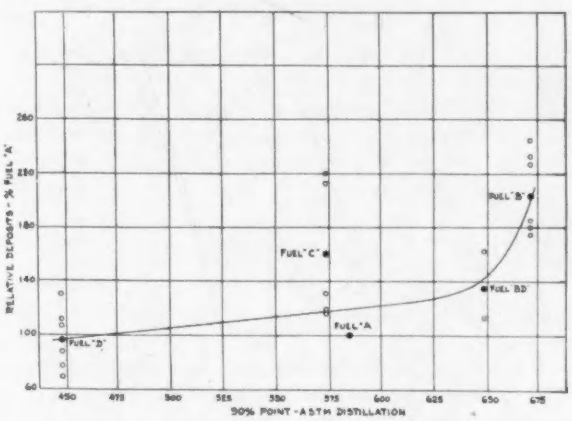
Code	A	B	C	D	½B+ ½C	½A+ ½C	¼A+ ¾C
1	8.06	14.29	9.57	8.45	9.10	...	...
3	94.8	173.7	111.3	73.8	...	...	...
5	7.71	18.81	10.02	10.0	...	...	...
6	27.6	58.6	60.9	24.6	44.5	...	...
7	*42.8	75.2	...	30.0	...	53.0	20.2
13	11.39	25.70	24.07	12.56	...	...	...

* At 2500 rpm for 24 hr only.

% of Fuel A

Code	A	B	C	D	½B+ ½D	½A+ ½D	¼A+ ¾C
1	100	178	119	105	113	...	...
3	100	184	118	78	...	...	...
5	100	244	130	130	...	...	...
6	100	212	220	89	161	...	...
7	100	175	...	70	...	124	47
13	100	226	212	110	...	...	...
Average	100	203	160	97	137	124	47

At conclusion of the test, photographic records were made of fuel valves, exhaust and intake valves, cylinder walls, and under-side of cylinder head and valve ports. The deposits were scraped carefully from fuel injection valves, exhaust valves and exhaust ports, air inlet passages, cylinder walls and pistons above top rings, and cylinder heads, and weighed separately to the nearest 0.01 g.



■ Fig. 3 – Relative deposits versus 90% point – ASTM distillation

Engine deposition, as herein discussed, concerns only that portion of material commonly designated as combustion-chamber deposits. The influence of fuel characteristics upon this specific condition involves a detailed study of deposits occurring in various typical engines under conditions set forth in the method of test.

A tabulation of engine deposit data is given in Table 3.

Fuels A and C possess slight differences in their volatility characteristics, but have wide differences in ignition qualities (33.2 to 56.1 cetane number). Fig. 2 shows a curve which indicates that cetane number is related to combustion-chamber deposits, though the distribution of points (Fig. 2) representing tests on fuels of approximately 40 cetane suggests that there may be other factors which have a greater effect.

Figs. 3 and 4 show relative deposits plotted against the 90% point and end point (ASTM distillation) respectively with curves drawn through the averages of relative deposits. These curves indicate that volatility has an effect on combustion-chamber deposits which becomes more pronounced in the high boiling range.

Figs. 5 and 6 show relative deposits plotted against viscosity (Saybolt sec, Universal) and API gravity respectively. These curves indicate that these fuel characteristics have an effect on engine deposits. Again it should be noted that the spread of average points where more than one fuel was tested of the same viscosity, shows that other factors probably have an effect.

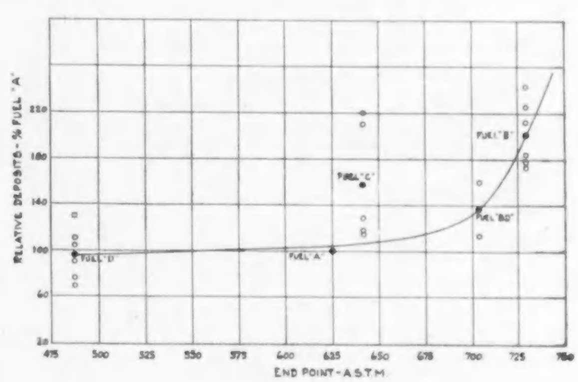
Curves on Figs. 7 and 8 indicate that carbon residues on 10% bottoms and the Hercules residue test show fuel properties which have an effect upon combustion-chamber deposits for the types of fuels investigated. The co-plots of the results of the residue of the 10% bottoms against Hercules residue test and the modified ASTM gum test, as shown in Figs. 9 and 10, show linear relationship which indicates that modified ASTM gum tests, carbon residue on 10% bottoms, and the Hercules residue test appear to measure the same fuel property. Since the fuels employed in this work contained little cracked stock, it is desirable to confirm these findings by further investigations.

■ Engine Smoothness

The pressure rise per degree of crank angle was employed as an indication of engine combustion smoothness. In most cases data were obtainable from combustion oscillograms. One laboratory reported results based on an aural method and its relative ratings agreed quite well with those obtained from oscillograms.

Five companies submitted data on five engines, representing four different makes. The data showing the effect of cetane number on relative engine roughness are given in Table 4 and a plot of these data against cetane number is shown in Fig. 11.

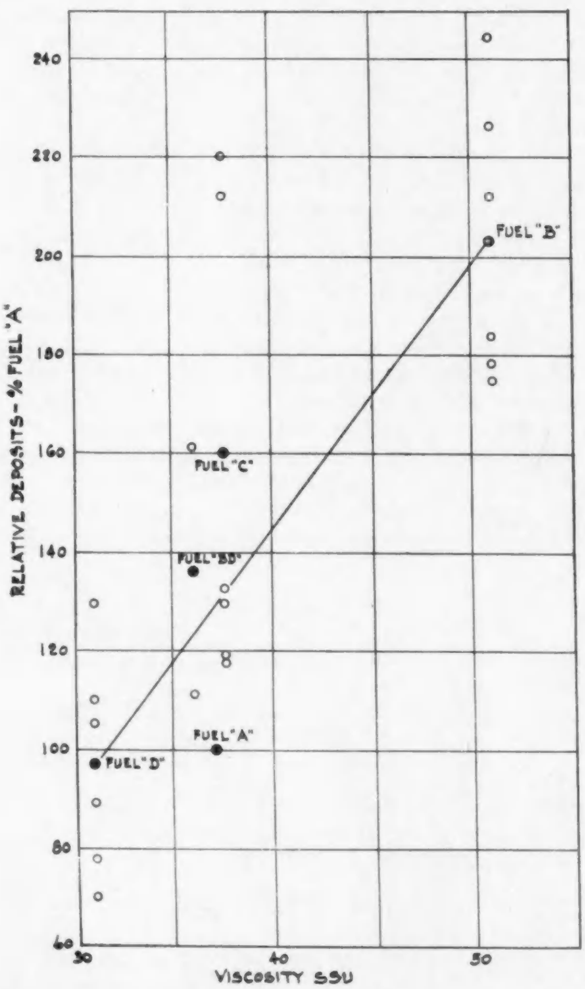
It is needless to say that Fuel A produced different rates of pressure rise in the different engines (see Table 4) which ranged from 21.0 to 49.0 psi per deg. Since roughness in one engine apparently was not affected greatly by cetane number in these tests, the data on this engine have been considered separately and a dotted line drawn through those points on Fig. 11. The results of the other four engines have been averaged, and these are indicated by the "solid square" symbols. It is apparent that cetane number of a diesel fuel over the range investigated (that is, from



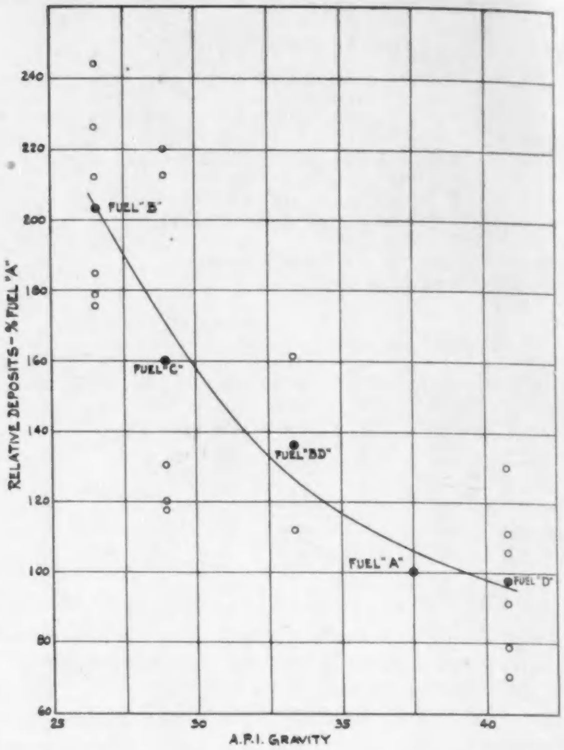
■ Fig. 4 – Relative deposits versus end point

Table 4 – Effect of Cetane Number on Relative Engine Roughness

Engine No.	Fuel A	Fuel B	Fuel C	Fuel D	50% B 50% D
	Pressure Rise, per Degree of Crank Angle, psi				
1	21.0	26.5	22.5	24.0	23.0
4	35.0	75.0	70.0	60.0	....
6	49.0	90.0	144.0	77.0	99.0
	Aural Ratings				
9	3.5	6.5	7.5	4.5	....
11	4.0	7.5	8.5	8.0	....
	Relative Roughness (Fuel A = 100%)				
	Fuel A	Fuel B	Fuel C	Fuel D	50% B 50% D
1	100	126	107	114	110
4	100	214	200	171	....
6	100	184	294	157	202
	Aural Ratings				
9	100	186	214	128	....
11	100	188	213	200	....



■ Fig. 5 – Relative deposits versus S.S.U. viscosity



■ Fig. 6 – Relative deposits versus API gravity

33.2 to 56.1) has a marked effect on engine combustion roughness.

The ASTM 90% point, end point, API gravity, and viscosity (Saybolt sec Universal) over the ranges studied, show no substantial correlation with engine combustion roughness.

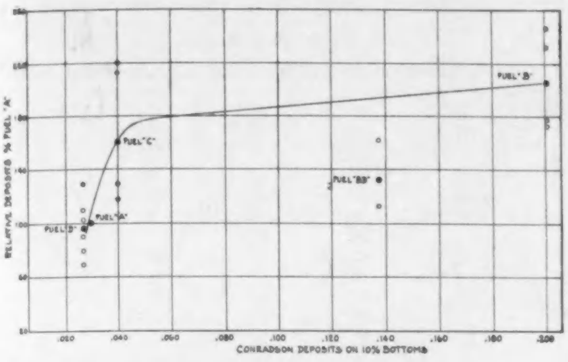
■ Starting

The four laboratories used different methods of evaluating the starting characteristics of the fuels. One laboratory determined the starting time for the various fuels at room temperatures and jacket-water temperatures of about 70 F. The electric dynamometer was used as a cranking medium of 350 rpm.

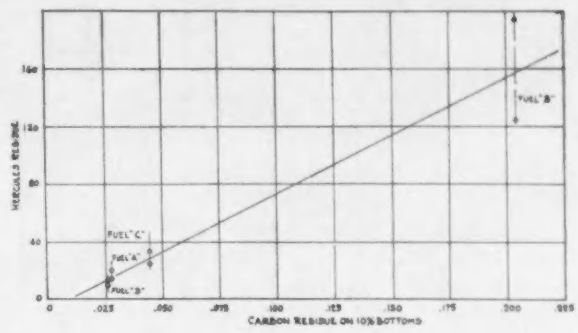
One laboratory artificially cooled the engine by using for the intake, cold air provided by the exhaust from an air motor and by circulating a coolant through the jacket. Air temperature was maintained at approximately 15 F, and jacket temperature was maintained at approximately 17 F. The cranking time was determined for each fuel with the use of a starter with a cranking speed of approximately 170 rpm.

One laboratory artificially cooled the engine by placing it in a cold room and allowing sufficient time between starting attempts for the engine and surroundings to reach a temperature of 35 F. The engine was cranked at constant cranking speed by means of a starter and batteries. Cranking speed was constant at 260 rpm. The time for a start was determined for the various fuels.

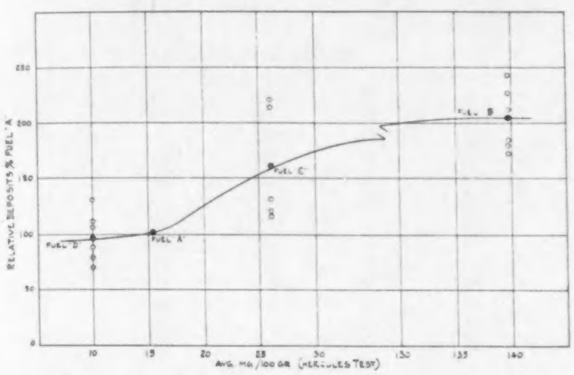
The other laboratory reported the minimum cranking speed in rpm necessary to start the engine at a jacket temperature of 80 F and air temperature of 86 F for the various fuels. The starting data are given in Table 5.



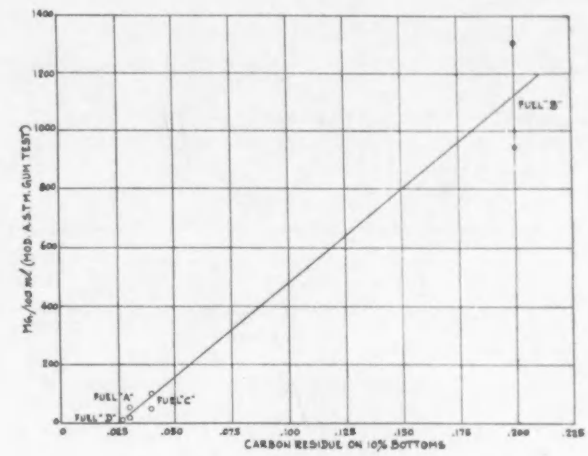
■ Fig. 7 – Relative deposits versus 10% bottoms



■ Fig. 9 – Hercules residue versus carbon residue on 10% bottoms



■ Fig. 8 – Relative deposits versus average mg per 100 g (Hercules test)



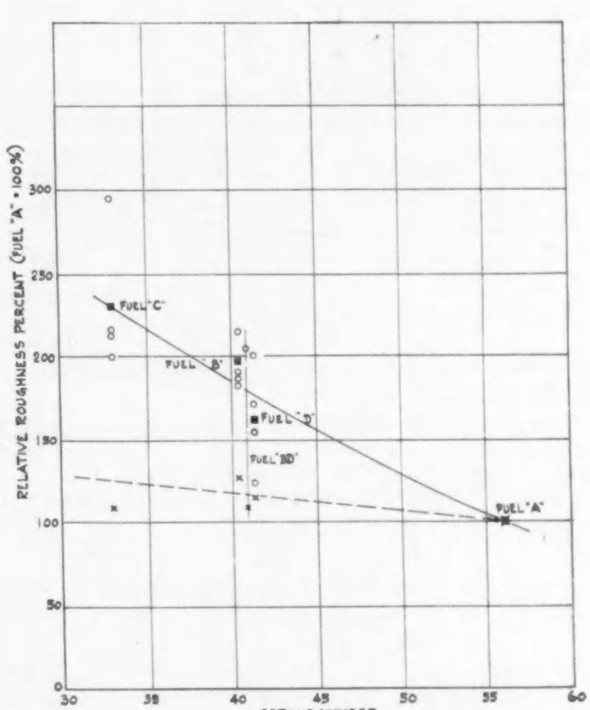
■ Fig. 10 – Mg per 100 ml (modified ASTM gum test) versus carbon residue

Table 5 – Starting Data

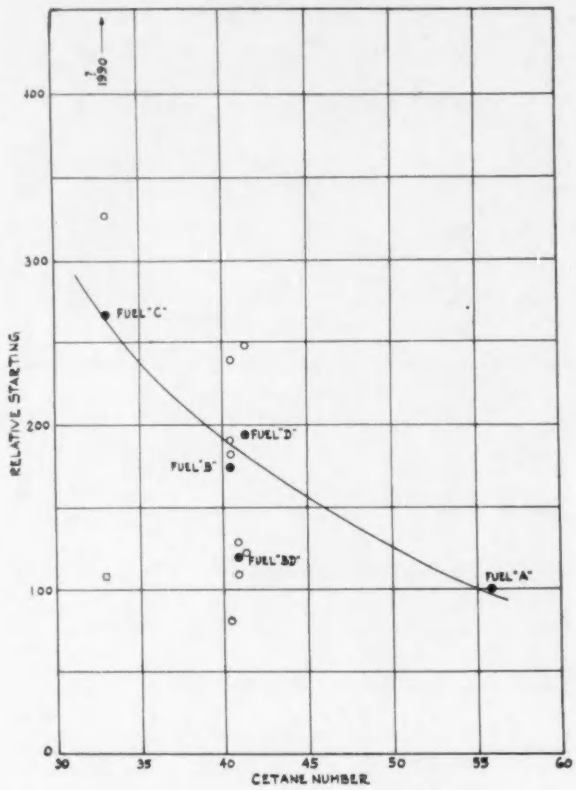
Code	A	B	C	D	½B+½D	
	Starting					
2	70 rpm	56 rpm	76 rpm	85 rpm	76 rpm	86F Air 80F Water
3	38.3 sec	72.6 sec	138 sec	80 sec		15F Air 17F Jacket 168 rpm
6	17 sec	31 sec	55 sec	42 sec	22 sec	35F Air 35F Water 260 rpm
9	31 sec	73.5 sec	618 sec	58.7 sec		70F Room 74F Jacket 350 rpm
	% of Fuel A					
2	100	80	109	121	108	86F Air 80F Water
3	100	190	360	209		15F Air 17F Jacket 168 rpm
6	100	182	324	247	129	35F Air 35F Water 260 rpm
9	100	237	1990*	189		70F Room 74F Jacket 350 rpm
Avg.	100	172	264	192	119	

* Omitted from average

For the three laboratories reporting in seconds, the starting conditions varied from 15 F air, 17 F jacket, and 168 rpm to 70 F air, 74 F jacket, and 350 rpm; the conditions used for the results in minimum rpm were 86 F air and 80 F jacket.



■ Fig. 11 – Relative engine roughness versus cetane number



■ Fig. 12 – Relative starting versus cetane number

The curve on Fig. 12 indicates that, for the majority of the engines reported, the cetane number affects the ease of starting. The higher the cetane number, the easier.

The other fuel properties failed to show any independent influences.

■ Smoke

The smoke data presented here were determined in nine laboratories with nine different smokemeters. These were of two general types – full-flow and sampling. The former was used by two laboratories, and the latter was used by the other seven. The effective lengths of the meters varied approximately from 8½ to 28½ in. The results were re-

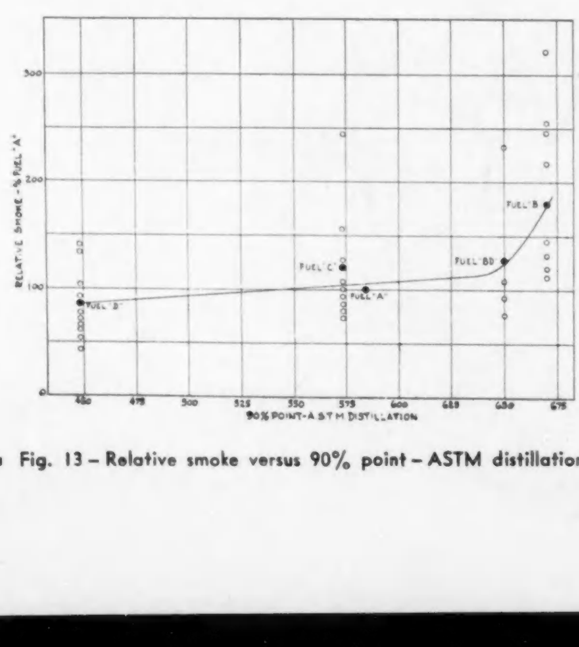
Table 6 – Smoke Data

	% Smoke at 85% of Maximum Load at Rated Speed				
Code	A	B	C	D	½ B+½ D
1	17	20	21	17	13
3	31.4	66.2	47.6	22.6	33.3
5	58	63	51	70	....
6	28	52	56	37	54
7	10	12	8	6	9
8	8	34	42	7	....
9	36	66	50	33	....
10	14.5	24	20	10	....
11	56	69	51	41	....
12	23	30	21	11.5	....
13	32	40	34	15	....
14	5	4	52	37	....
15	35	75	34	26	....

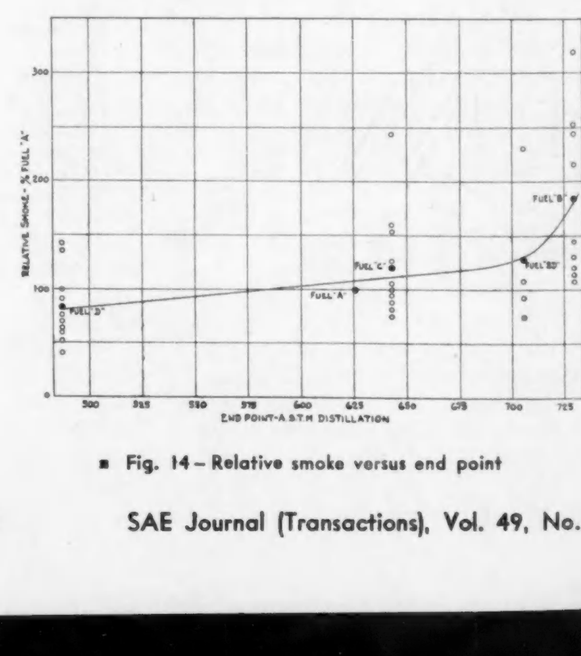
	Specific Smoke Density				
Code	A	B	C	D	½ B+½ D
1	16	19	20	16	12
3	87	220	139	60	94
5	106	122	87	147	....
6	41	89	100	57	95
7	53	63	44	36	49
8	22	88	104	19	....
9	54	132	85	50	....
11	100	144	87	65	....
12	58	76	54	31	....
13	48	63	51	20	....
14	13	10	134	94	....
15	53	170	51	37	....

	Relative Specific Smoke Density Based on % of Fuel A				
Code	A	B	C	D	½ B+½ D
1	100	119	125	100	75
3	100	253	160	69	108
5	100	115	82	139	....
6	100	216	244	139	232
7	100	119	83	68	92
8	100	400	473*	86	....
9	100	245	158	93	....
11	100	144	87	65	....
12	100	131	93	53	....
13	100	131	106	42	....
14	100	77	1030*	723*	....
15	100	320*	96	70	....
Average	100	177	123	84	127

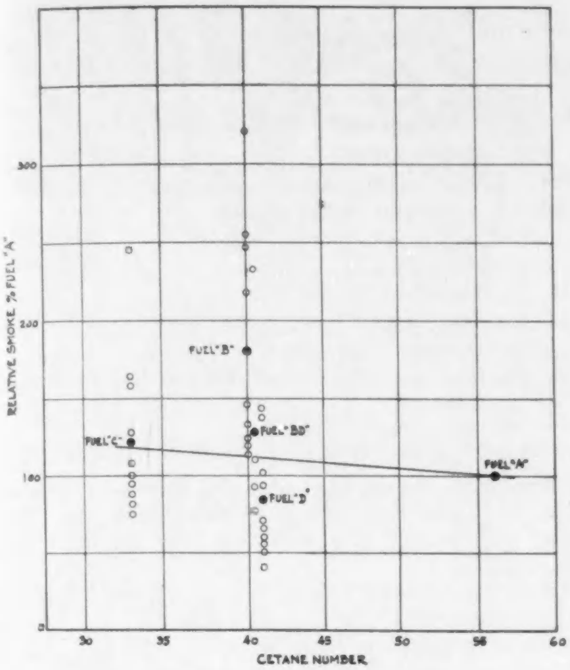
* Values omitted from average



■ Fig. 13 – Relative smoke versus 90% point – ASTM distillation

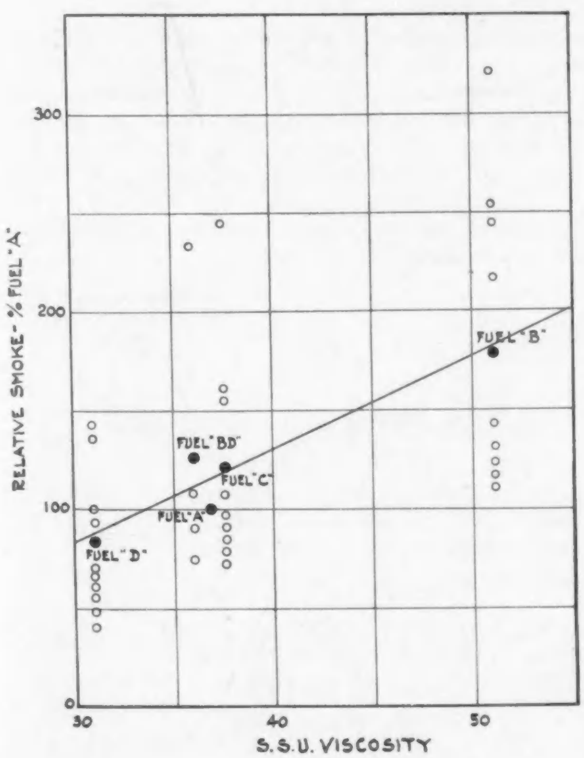


■ Fig. 14 – Relative smoke versus end point

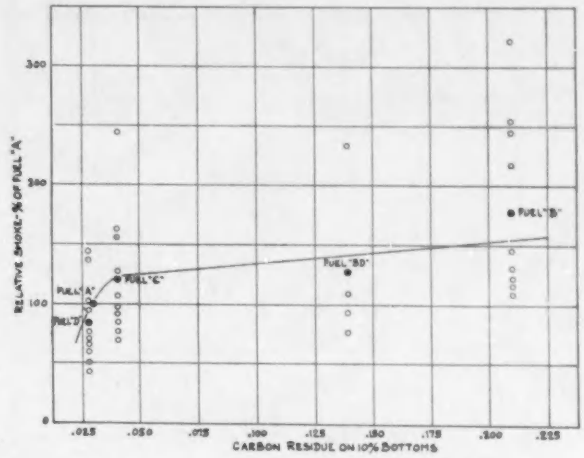


■ Fig. 15 – Relative smoke versus cetane number

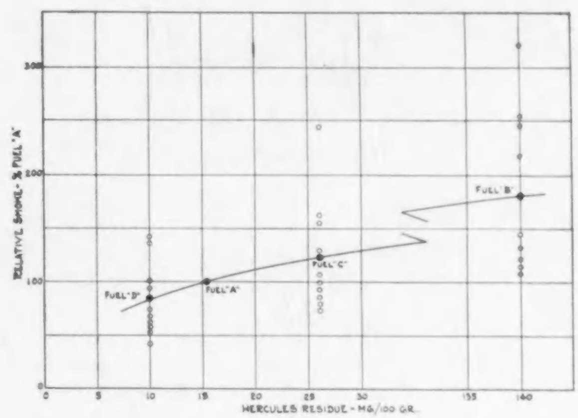
ported by the laboratories on these different bases – per cent smoke, micro-amperes decrease, and micro-amperes. In order to correlate these results it was thought desirable to convert the readings to a common standard. The method is that outlined by K. M. Brown.[2] The scale used in this method is specific smoke density or density of a column



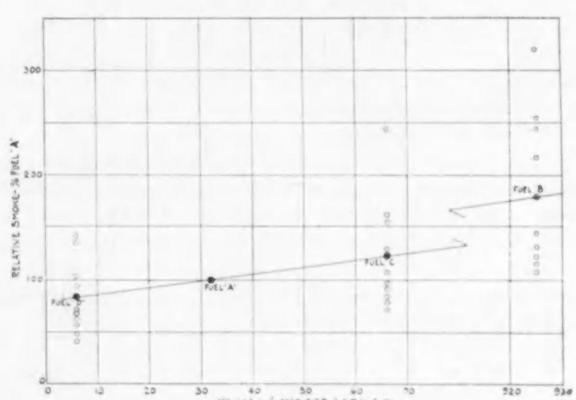
■ Fig. 16 – Relative smoke versus Saybolt Universal Viscosity in sec



■ Fig. 17 – Relative smoke versus carbon residue



■ Fig. 18 – Relative smoke versus Hercules residue

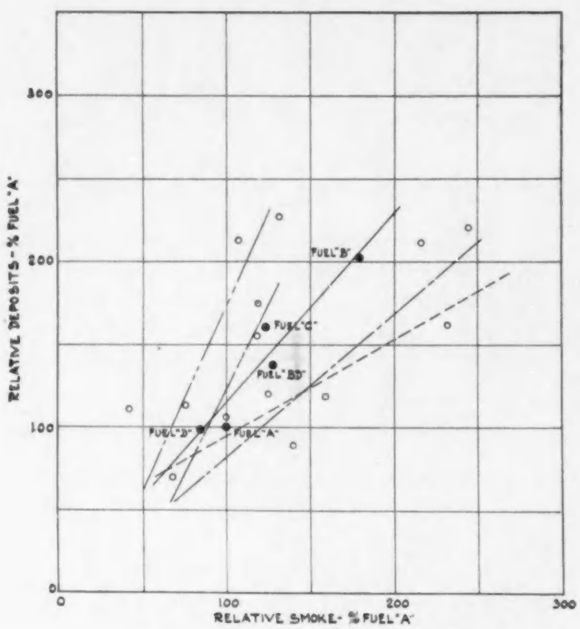


■ Fig. 19 – Relative smoke versus modified ASTM gum

of the smoke 100 m long. The values reported and those so obtained are given in Table 6.

The curves on Figs. 13 and 14 indicate that the smoking tendency of an engine is related to the volatility of the fuel. The limited number of fuels investigated makes it

[2] See SAE Transactions, May, 1941, pp. 188-192: "Requirements of a Smokemeter," by Kenneth M. Brown.



■ Fig. 20 – Relative deposits versus relative smoke

difficult to say whether the break in the curves at 650 F at the 90% point and 700 F end point is a true indication of critical volatility points as far as exhaust smoke is concerned.

The plot of relative smoke versus cetane number (Fig. 15) indicates that smoke decreases with increase in cetane number, though the spread of average values for the several fuels about 40 cetane number indicates that some other characteristic may have a greater effect than the total effect of cetane within the range covered.

The curve shown on Fig. 16 would seem to indicate that the smoking tendency is increased by high viscosities, but it must be remembered that the fuel with the highest viscosity was also the fuel with the lowest volatility. Consequently, it does not necessarily follow that viscosity, within the range of the fuels tested, affects smoking as an independent fuel property.

Curves shown on Figs. 17, 18, and 19 indicate an increase in smoking tendency with carbon residue on 10% bottoms, Hercules residue, and modified ASTM gum respectively. Below 0.04 of carbon residue on 10% bottoms, the smoke appears to drop off decidedly.

From the curve shown on Fig. 20 it would appear that there is a correlation between the amount of engine deposits and the smoking tendency of the engine; the more the engine smokes, the greater will be the deposits found.

■ Power Output and Fuel Consumption

The following nine laboratories participated in the test work to determine the influence of the four CFR reference fuels on power and consumption:

1. University of Wisconsin
2. Shell Oil Co.
3. Fairbanks, Morse & Co.
4. The Texas Co.
5. Atlantic Refining Co.
6. Standard Oil Development Co.
7. Chrysler Corp.
8. Caterpillar Tractor Co.
9. Mack Manufacturing Co.

Tests were run on 1-cyl and 4-cyl Fairbanks-Morse engines, 1-cyl and 3-cyl General Motors engines, the Dodge diesel, the Caterpillar 4-cyl engine, and the Mack 6-cyl diesel engine.

Maximum Power – Two of the laboratories determined the influence of the test fuels on maximum power output. One type of engine was exclusively used for this work, it being the only one provided with a positive maximum fuel stop. The results were as follows:

Reference Fuel	Maximum Power, Output, %		
	Engine No. 10	Engine No. 13	Average
A	94.0	96.5	95.2
B	100.0	100.0	100.0
C	97.0	97.0	97.0
D	91.6	94.7	93.2

These values show that the maximum obtainable power of this engine drops as much as 6.8% over the range of the investigated fuels.

Fuel Consumption – All laboratories investigated one change of volumetric fuel consumption occurring when maintaining constant power output with use of the four fuels. The results of these tests are shown in Table 7 on a percentage basis. This method was selected in preference to presentation of the absolute values, since difference in consumption of the same engine in different laboratories exceeded the maximum variations encountered between all engine types.

Averaging the values given in Table 7, the following summary is obtained:

Reference Fuel	Average Volumetric Fuel Consumption
A	103.4
B	100.0
C	101.5
D	105.5
50% B + 50% D	104.7

Table 7 – Fuel Consumption (Per Cent)

Reference Fuel	Engine Numbers										
	No. 4	No. 5	No. 6	No. 10	No. 11	No. 12	No. 13	No. 7	No. 9	No. 1	No. 3
A	104.5	102.5	101.5	106.0	105.5	103.0	107.0	103.0	100.0	103.0	101.5
B	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
C	99.0	99.5	102.0	106.0	98.0	100.0	103.0	102.0	105.0	101.0	100.5
D	105.0	103.5	104.0	107.0	106.0	103.5	110.0	104.0	104.5	107.5	106.0
50% B and 50% D	.....	.....	103.0	.....	.....	.....	.....	108.0	.....	103.0	.....

Of the physical and chemical fuel oil properties, the thermal value must be expected to be of foremost influence on the amount of power produced in the engine.

The heat values of the four reference fuels were checked by three different laboratories and the averages follow:

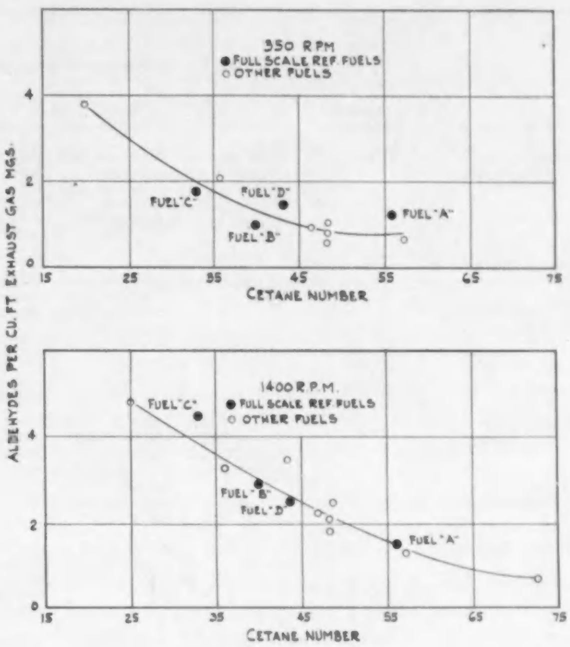
Reference Fuel	Average Volumetric Heat Value (Higher)	
	Btu/Gal	%
A	137,945	96.1
B	143,347	100.0
C	142,186	99.0
D	135,452	94.4

A comparison of the percentage difference of the thermal values with the decrease of maximum power and the increase of the fuel consumption is shown in the following table:

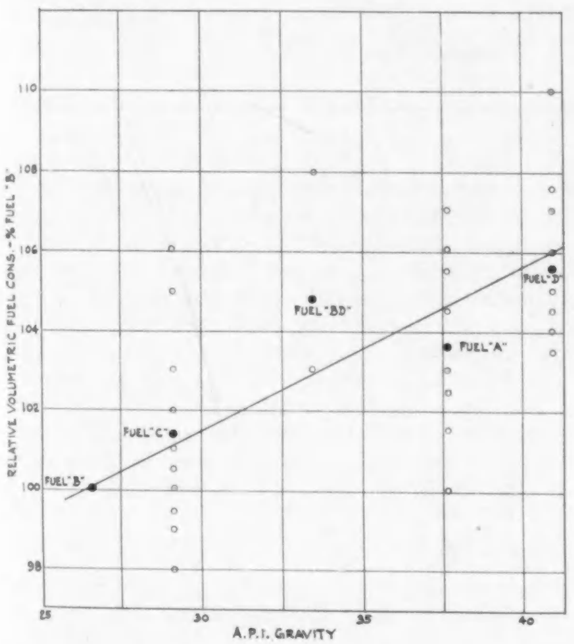
Reference Fuel	Thermal Value, %	Maximum Power	Volumetric Fuel Consumption
A	96.1	95.25	103.4
B	100.0	100.00	100.0
C	99.0	97.00	101.5
D	94.4	93.15	105.5

These figures reveal clearly that most of the adverse effects of the different fuel properties can be accounted for by the corresponding change of the thermal values.

For practical purposes, the thermal value of a fuel is



■ Fig. 22 – Odor versus cetane number



■ Fig. 21 – Relative fuel consumption versus API gravity

best expressed by its gravity which can, therefore, be used to advantage as a criterion for its relative merits with regard to power and consumption.

The curve on Fig. 21 shows the plotted results of the consumption test versus the API gravity of the reference fuels. Due to their basic relation to gravity and heat value, some other properties, such as viscosity and volatility, may reflect the foregoing results in a corresponding manner.

For average commercial fuels, an increase of the volatility, or a decrease of the viscosity which will accompany an increase of the API gravity must, therefore, likewise be taken as indicative of lowered volumetric thermal values, resulting in loss of power and economy.

Odor – Exhaust odor ratings of the four test fuels, while included in the original program, have only been made by The Texas Co. Since these tests were made on only one engine and have not been confirmed by others, the data are not considered in the major conclusions, but are included in Appendix B for further consideration.

It is interesting to note, however, that this laboratory reports that under the specific conditions used, exhaust odor objectionability is principally a function of fuel ignition quality (Fig. 22) decreasing as cetane value increases. It is also indicated that odor is independent of volatility. The recommendation is made that other laboratories continue this phase of the investigation.

■ Summary

In summarizing the data from these full-scale diesel engine tests, it is realized that many of the fuel characteristics are related, and though engine performance varies with the test values on the fuels used, it may or may not be a direct relation.

A summary of engine effects as related to the fuel properties is given in Table 8. From this table, which is based on test results obtained on the selected reference fuels, the following conclusions may be drawn:

The ignition quality of the fuel (cetane number ASTM) affects starting and engine smoothness. Increasing the cetane value of the fuel improves starting and makes the engine run smoother. Some engines show a greater sensitivity to cetane value than others but all show improvement in operation as far as carbon deposits, smoke, exhaust odor, smoothness, and starting are concerned.

The more volatile the fuel, the greater the amount of exhaust smoke and the greater the amount of combustion-chamber deposits. Gravity, viscosity, carbon residue, flash point are all related to the volatility of the fuel and, as such, show effects on exhaust smoke and combustion-

Table 8 – Summary of Engine Effects as Related to Fuel Properties

Fuel Property	Limits of Investigation	Starting	Smoothness	Smoke	Exhaust Odor	Power	Fuel Consumption	Combustion-Chamber Deposit
Ignition Quality (Cetane Number)	33.2-56.1	Direct relation. Increase in cetane number improves starting.	Direct relation. Increase in cetane number improves smoothness.	Relation apparent. Data indicates a min. cetane value for specific condition. Increase from this point shows no improvement.	Direct relation indicated. Increase in cetane number results in decreased exhaust odor objectionability.	No apparent relation.	No apparent relation.	Relation indicated. Decreased deposits with increase in cetane number.
Volatility (ASTM) 90% Point End Point	 449-672 487-729	No apparent relation.	Some relation indicated. Decrease apparently detrimental, thought not conclusive.	Direct relation. Lower volatility results in more smoke.	No apparent relation.	No specific independent relation indicated.	No specific independent relation indicated.	Relation indicated. Deposits increase with decrease in volatility.
Flash (PM)	157-238				No Independent Data			
Viscosity 100 F S. S. Universal Kinematic (Centistokes)	 31-51 1.54-7.81	No apparent relation. Data scattered.	Some relation indicated. Viscosity increase appears detrimental though not conclusive.	Relation indicated. Increase results in more smoke. Probably a reflection of volatility.	No independent data.	No specific independent relation indicated.	No specific independent relation indicated.	Relation indicated. Deposits increase with increase in viscosity. Probably interrelated with volatility.
Gravity (API) Heating Value (Btu/Gal)	26.6-40.8 135,452-143,347	No apparent relation.	No apparent relation.	Relation indicated. Increase results in more smoke. Probably a reflection of volatility.	No independent data.	Direct relation. Probably due to thermal value of fuel.	Direct relation. Probably due to thermal value of fuel.	Relation indicated. Correlation probably due to interrelated characteristics.
Carbon Residue (10% Bottoms)	0.027-0.21	No relation.	No relation.	Relation indicated. Decrease results in slight improvement.	No independent data.	No relation.	No relation.	Relation indicated. Decrease results in slight improvement.
Modified ASTM Gum Hercules Residue Test		No relation.	No relation.	Relation indicated. Decrease results in slight improvement.	No independent data.	No relation.	No relation.	Relation indicated. Decrease results in slight improvement.
Sulfur, %	0.07-0.60				No Independent Data			

chamber deposits which may or may not be independent effects.

The viscosity (Saybolt sec) affects smoothness and smoke. The importance of viscosity was recognized in the basic research program as involving the consideration of ease of circulation, atomization, penetration, injection-pump plunger leakage, lubrication quality, heat content, volatility, and overall power output. The data on power output show a decrease in power with a decrease in viscosity which may be caused partially by the effect of viscosity.

The gravity (API) affects smoke, power, and fuel consumption. The carbon residue on 10% bottoms (ASTM) affects smoke and combustion-chamber deposits.

This paper is a report of the cooperative full-scale testing carried out by this group to date. It is hoped that the results of this initial series of tests will serve as a basis for further study which will add to the knowledge of the automotive diesel-engine fuel requirement.

APPENDIX A

In view of the need for a universal diesel fuel suitable for mobile Army and certain high-speed Navy equipment, the data and advice available to the Automotive Diesel Fuels Division of the Cooperative Fuel Research Committee have been analyzed and are offered as an aid in the development of the specification for such a fuel. The D-2 ASTM fuel oil specification was reviewed, and it was generally agreed by this group that due to the recent developments in high-speed diesel engines as well as a more complete understanding of the effect of certain fuel characteristics on engine performance, some revision of the ASTM D-2 specification may be desirable.

Any fuel specification which must take into consideration the availability of base stocks from the greatest possible number of sources and which must apply to fuel suitable for a variety of types of engines, of necessity, must be a compromise if a single specification is mandatory.

The following is a résumé of a series of discussions of the more important fuel variables. The discussions on the several items by the cooperating members were based on information available from the cooperative full-scale engine tests, laboratory data, and field experience.

■ Viscosity

The importance of viscosity was recognized in the basic research program of the Division as involving the consideration of ease of circulation, atomization, penetration, injection-pump plunger leakage, lubrication quality, heat content, volatility, and finally, overall power and economy. Although viscosities lower than 35 sec Saybolt Universal at 100 F tend to decrease the power output at a given pump setting due to plunger leakage and inherently lower heat value, in order to broaden the base of fuel supply, a lower limit of 33 sec Saybolt Universal Viscosity at 100 F is considered practical for high-speed diesel engines. In any case, where worn pumps are encountered, resetting the pumps will in most cases restore the maximum power available, if the power loss is due to plunger leakage. It is further recognized that, when operations are anticipated in a given area at atmospheric temperatures below an average daily minimum of plus 10 F, the necessity of reaching correspondingly lower pour points may require a fur-

ther lowering of the viscosity, in which case a 31 sec Saybolt Universal Viscosity at 100 F is justified. It is understood that these viscosity values will be determined by the kinematic method, defined by the ASTM method D-445-39T, and converted to Saybolt Universal Viscosity according to the ASTM method D-446-39T. The following viscosity values therefore seem desirable: 33 sec Saybolt Universal Viscosity, that is, 2.11 centistokes viscosity minimum and 43 sec Saybolt Universal Viscosity, that is, 5.22 centistokes viscosity maximum at 100 F.

■ Volatility

The upper limit of the boiling range will be governed by the requirements of the high-speed automotive type engines. The wide speed, load, and temperature variations encountered in this type of service indicate the desirability of complete distillation of the fuel below 700 F. To insure minimum smoke, odor, lubrication-oil contamination, and engine deposits, it will be necessary to prevent undue concentration of heavy ends in the fuel. Therefore, an ASTM 90% distilled point at 650 F should not be exceeded. Inasmuch as the minimum flash point, to be discussed later, will tend to control the permissible amount of the undesirable low boiling fraction in the fuel, it is not thought necessary to define the lower end of the boiling range. Summarized, this would indicate a 650 F maximum 90% point and a 700 F maximum end point for a universal diesel fuel, when determined by ASTM method D86-35.

■ Flash Point

Smoothness of combustion and fire hazard are in some way related to front end volatility and can most conveniently be defined by flash point.

Available information indicates that satisfactory engine operation is possible with fuels having flash points lower than 140 F, and that such a flash point should prove to be a conservative minimum for insuring freedom from operational difficulties.

■ Carbon Residue

Definition of that fuel property which affects combustion-chamber and nozzle deposits is usually controlled by the volatility limitation. However, to eliminate unusual products which might have excessive concentrations of residue within the allowable volatility range, a carbon residue test on the 10% bottoms by the ASTM method, D-189-39, is considered necessary. Performance studies indicate that fuels having a residue not in excess of 0.15% of the 10% residuum have proved satisfactory.

■ Ignition Quality

The ease with which a diesel fuel auto-ignites, affects not only the general combustion in the engine but also its starting characteristics. Since the bulk of the commercially available fuels will have cetane values less than 60, no upper limit of ignition quality was considered. It is generally recognized that high cetane values are desirable, but available fuel stocks meeting the other requirements herein expressed, occasion the use of a 47 cetane minimum value at least for fuel stocks available from certain supply areas. Information available on a majority of the engine types used, operating under a wide range of conditions, indicates satisfactory performance on fuels of at least 47 cetane value, and this should prove a satisfactory minimum value for a universal military fuel.

■ Pour Point

The pour point of a fuel oil is of triple importance, affecting in cold weather:

1. The bulk handling of the fuel
2. The pumpability of the fuel in the engine
3. The fuel filter performance at low temperature

In view of the fact that the best fuels from an ignition angle tend to have poor cold-weather characteristics, the pour point should not be specified lower than absolutely necessary. In order to obtain adequate supplies of fuels having the other necessary properties, outlined herein, it is advisable that a universal fuel for general military use have a pour point of 0 F. Since the majority of commercially available fuels have a cloud point approximately 10 F above the pour point, and since the cloud point limits the flow through the filter, a safe operating temperature for a 0 F pour point fuel would be +10 F. In those sections where the average daily minimum temperature is below +10 F, it will be necessary to use fuels of correspondingly lower pour point. Under such conditions a corresponding adjustment of the minimum viscosity may be found necessary, as discussed under viscosity, preceding.

■ Ash

The Conradson carbon residue limit of 0.15% on 10% bottoms will, in itself, limit the ash content, since that test will indicate ash in addition to carbonaceous material. A separate limitation on ash alone would therefore be superfluous.

■ Water and Sediment

While the Committee's program has added little to existing data on the importance of keeping water and sediment within negligible amounts, all reported experience appears to indicate that the value of 0.05 maximum, as incorporated in specifications already in use, is satisfactory, and in no way limits supplies or works hardship on any refiner, the water and sediment determination to be made according to ASTM method D96-35.

■ Sulfur

While the evidence regarding the absolute maximum limit for sulfur is inconclusive, experience indicates that a maximum value of 1.0% by ASTM method D129-34, is satisfactory from the users' standpoint, and such a limit has the value of permitting the use locally, in those sections of the country where only high sulfur fuels can be manufactured, of fuels produced in that territory. A low value would exclude the use of such fuels and thereby limit the total available supply.

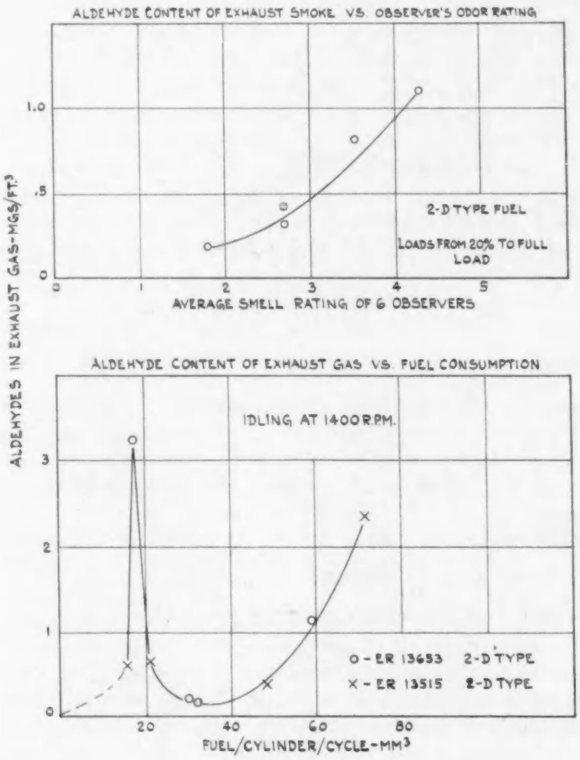
■ Corrosion

As a protection against corrosion difficulties in fuel lines of various compositions, it is considered desirable to require a 3-hr copper-strip test at 212 F. In this test, a "peacock color" is to be interpreted as an indication of "tarnish" but not corrosion; that is, a fuel giving a "peacock color" shall be considered satisfactory. A darker color is to be intepreted as unsatisfactory and not acceptable.

APPENDIX B

■ Exhaust Odor Rating of Diesel Fuels (The Texas Co., Beacon Laboratory)

The complete combustion of the hydrocarbons supplied to any internal-combustion engine should result in exhaust gases which are practically free from objectionable odors. However, many factors may prevent the complete oxidation process with the resultant discharge of partially burned or oxidized hydrocarbons with the exhaust, giving rise to observations and complaints of smoke and objectionable odors. It has been shown that at an intermediate step in the combustion reactions for normal hydrocarbon fuels, various aldehydes are formed. In their separate state, some of these aldehydes are well known to have an acrid, pungent odor, producing more or less intense irritation of the eyes and respiratory system. It was, therefore, logical that some experimenters should use the aldehyde content of engine exhaust gases as a practical measure of their objectionability.



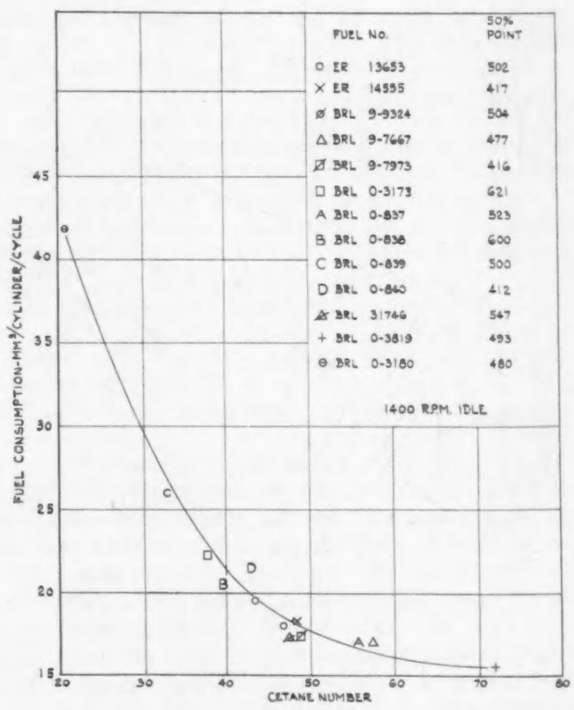
■ Fig. 23 – Typical correlation between observer odor reaction and fuel consumption with aldehydes in exhaust gas

Since the operation of diesel-engined city buses obviously presents a very critical source of exhaust odor complaints, observations were made to determine the type of operation producing the most objectionable conditions. A number of observers agreed that the later stage of deceleration prior to stopping, the idling period, and the subsequent initial acceleration period produced the greatest quantity and most objectionable fumes; and also consistently distinguished differences between fuels having not very greatly different physical characteristics. Subsequent laboratory tests, using a similar engine confirmed the field tests, and established that very good agreement existed between "sniff" tests and the total aldehyde content of the exhaust sample. Fig. 23 illustrates the typical correlation between observer reaction and total aldehydes expressed as formaldehyde in mg per cu ft of exhaust gas.

Extensive dynamometer tests indicated, on the particular engine used, that the exhaust aldehyde content was greatest at idling, diminishing at part throttle and increasing at full throttle, as illustrated on Fig. 23. Since the idling condition appeared most critical, tests were made on a wide range of fuels to determine the minimum fuel rate which would permit smooth idling. Fig. 24 indicates that the cetane value of the fuel controls the minimum quantity injected which will give uniform combustion. Obviously, decreasing the fuel rate for a given fuel below its minimum value indicated, would result in discharge of partially oxidized fuel to the exhaust with resultant objectionable odors. This curve also indicates that, under these specific conditions, a rather wide range of fuel volatility had little or no effect.

For a specific injection rate, the exhaust aldehyde content for varying cetane number is shown in Fig. 22 for an engine running at no load at a speed of 1400 rpm, and for an engine idling at 350 rpm.

Since exhaust odor objectionably is a function of the completeness of combustion of the fuel, it appears logical that the fuel's ignition quality would be its controlling variable. These data, while limited in scope, seem to confirm the generalization that fuel cetane value is its principal variable in control of odor.



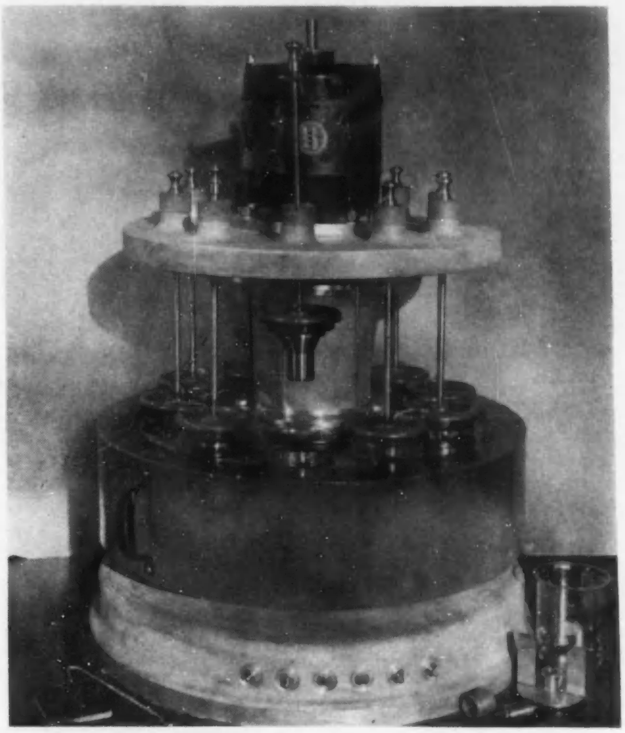
■ Fig. 24 – Fuel consumption versus cetane number

LUBRICATION of Severe-Duty Engines (DIESELS)

DISCUSSION

■

THIS paper, by J. G. McNab, W. C. Winning, B. G. Baldwin, and F. L. Miller, all of the Standard Oil Development Co., was published in full on pp. 309-325, in the August, 1941, SAE Journal, Transactions Section. Because of space requirements, the following discussion was not published with the paper:



■ Fig. A (Ryder discussion) – Pratt & Whitney version of MacCoull corrosion tester

P&W Oil Corrosion Tester

–E. A. Ryder

Pratt & Whitney Aircraft, Division of United Aircraft Corp.

THE authors have mentioned the fact that many addition agents which are desirable in some one respect, for instance detergency, cannot be used because they increase bearing corrosion. In this connection as in many others it is desirable to have a simple bench test to weed out grossly unsuitable materials and reduce the amount of engine testing necessary.

In studying oil stability Pratt & Whitney Aircraft felt the need of a better apparatus than was currently available for corrosion testing and tried to design a machine for an accelerated test consisting of heating and aerating the test oil in contact with any desired catalysts, and circulating it over a weighed specimen of the test bearing material. In addition, the apparatus should:

1. Use a small oil sample but sufficient to permit analysis after test.
2. Have all parts in contact with the oil easily cleanable.
3. Permit weighing the bearing sample as often as desired.
4. Have sufficient scrubbing action on the bearing to prevent or retard the formation of a protective film on the test bearing surface.
5. Provide for use of any solid catalyst desired (such as copper).

We were attracted by the basic principle of a corrosion rig designed by Neil MacCoull, in which the rotation of a bearing and associated parts was used to pump oil through the bearing clearance and spray it through the air in the upper part of the heated oil container. Figs. A and B show the Pratt & Whitney apparatus which uses the MacCoull scheme, but differs radically in design. It consists of a cluster of 10 beakers set in an electrically heated aluminum block, with the 10 spinners driven from above from a single motor.

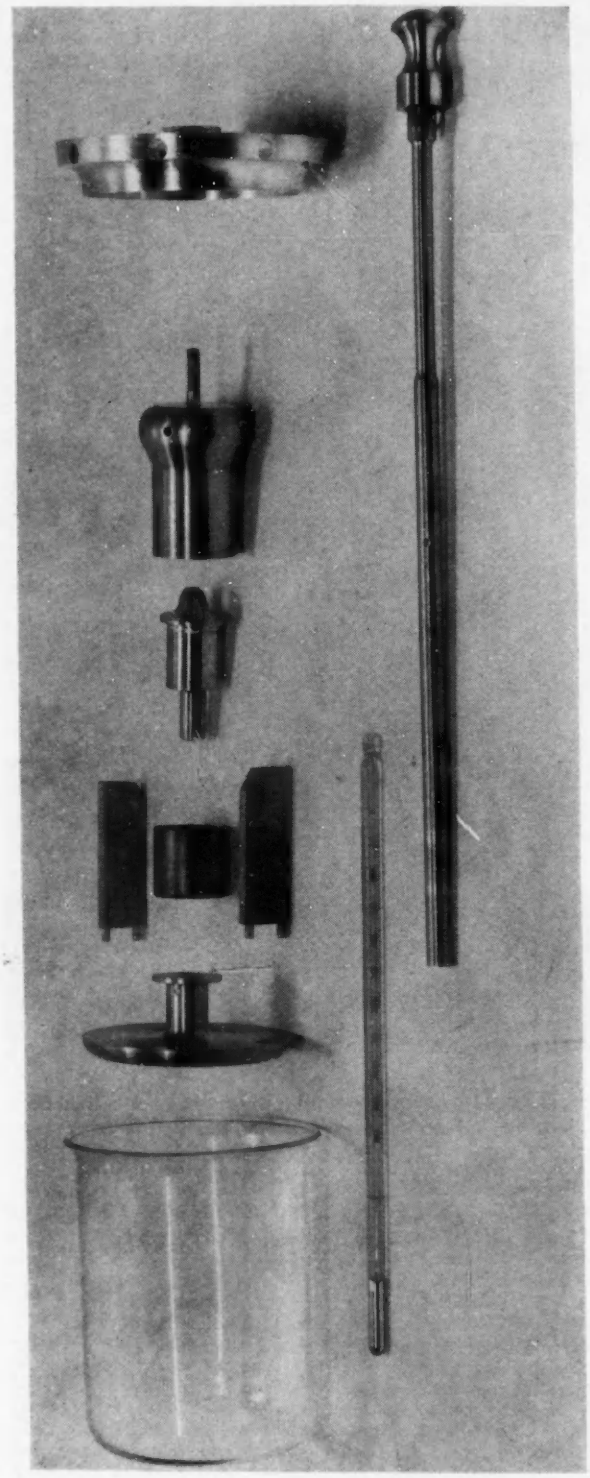
In the picture, one of the spinners is shown lifted out of its beaker, which action also raises the beaker cover, shown resting on top of the spinner. A test bearing is shown on the bench by the base of the machine. It is 1 in. in diameter by 1 in. long, and the outer surface is the test surface. The bearing is slipped over a supporting plug which is slightly tapered, and this, in turn, drops into a stainless-steel pedestal which sits in the bottom of the beaker. The two vanes standing up from the pedestal are usually made of copper, and sand-papered before each test. The stainless-steel spinner surrounds the test bushing and oil is pumped up through the bearing clearance and sprayed out from holes in the spinner top. The charge of oil (150 cc) fills the beaker to about the middle of the bushing or slightly higher. Holes in the cover give ventilation.

Each bearing may be fished out of its beaker at any time, and removed from its holder by merely tapping on the bench. It is then washed and weighed and replaced.

With temperatures in the range of 325 and 375 F, 5 hr is a long-enough test, and quite consistent results are obtained. While, of course, the oil may be analyzed after the test, weight loss of the specimen seems to be an adequate measure of the resistance of the

oil to deterioration. Correlation with aircraft engine results is good enough so that we consider this a significant test method, and others have obtained good correlation with corrosion tests made on automobile engines.

A commercial model of this corrosion apparatus will shortly be offered by an established instrument maker.



▪ Fig. B (Ryder discussion) – Exploded view of Pratt & Whitney oil corrosion tester

Disadvantages of Compounded Oils

– Chase Donaldson, *president, and*
Walter C. Bauer, *chief engineer,*
Briggs Clarifier Co.

SKELETONS and Niggers – The able presentation on the lubrication of severe-duty diesels, which you have just been reading, was undoubtedly written late at night after days of intensive research, but with the skeletons firmly locked in the closet and the nigger asleep in the wood pile.

Perhaps it will be impolite to unlock the door and upset the wood pile, but the presence of the skeleton and the nigger in this discussion may provide a few interesting reactions.

The clear and adequate presentation by the authors of the necessity of employing additive or compounded oils in the lubrication of severe-duty diesels unfortunately neglected, however, to mention the disagreement, often violent, existing on this subject, even among the oil technologists themselves, as well as the varying opinions of the designers and operators of diesel engines.

Consider the Operators' Viewpoint – Please let me suggest that we consider for a moment the viewpoint of the operator of engines as compared to that of the research technician.

Under normal operating conditions, what the operator wants to obtain is:

1. Continuous operation, with no interruptions or shutdowns for a period of time for any purpose.
2. Minimum maintenance work.
3. Long life for the engine.
4. Low-cost lubrication per hour or per mile of service.

Let us then consider the contributions to these requirements made by the additive or compounded oils and, second, the disadvantages, again from the viewpoint of the operator, of the use of such compounded oils.

At the outset, it is proper to grant in favor of the compounded or additive oils, that they may indeed be required for extreme conditions of operation where high power output for short periods of time are required or where unusual designs demand these lubricants.

Furthermore, the argument is advanced by the oil companies that the additive oils are superior to straight mineral oils in that they provide lower maintenance cost, partly through the elimination of ring-sticking and accompanying maintenance difficulties such as increased oil consumption, fuel consumption, and engine wear.

But the following disadvantages in the use of these additive oils, again from the operators' viewpoint, should be noted:

1. The cost of frequent oil changes. These, in many instances, are recommended after only 60 hr of operation. Added to the cost of frequent changes, is the inconvenience of such changes and the inaccessibility, in many cases, of supplies of the particular compounded oil which may be employed.

2. The instability of certain types of compounds or additives, in the presence of water, where there is a tendency of the additive to precipitate out of the oil.

3. In marine use, particularly, where water may be present, it is impossible to centrifuge the water out of the oil. Centrifuges can operate at from only 10% to 20% of their rated capacity on compounded oils owing to the effect of the dispersion agent on the resistance to emulsion properties of these oils. The authors, in their paper, confirmed this difficulty of centrifuging compounded oils.

4. The possible danger arising from the extended use of additive oils where it may be impossible or inconvenient to change them at the recommended intervals. It is, of course, well understood that, after the additives are exhausted or removed from the oil, the oil reverts to its original characteristics, with the added disadvantage that the solid material, often abrasive in character, is held in suspension in the oil, which, at the same time, will show a high neutralization number.

5. Certain of the older types of additive oils, some of which are still on the market, have, as is well known, a highly corrosive effect on the alloy-type bearings. Even the newer types of additive oils, in the presence of water, may prove corrosive in nature, even though theoretically non-corrosive.

6. From the viewpoint of the average operator, the appearance of the oil is no index to its condition since practically all the additive oils will be black in color after a few moments of operation, due to the dispersion of carbon throughout the oil. This question of visual cleanliness of oil being an accurate index of its condition has, of

course, been greatly exaggerated; but, for most operators, appearance is the only index which they have readily available other than actual analysis of the oil, as practiced by major fleet and diesel-engine operators.

Question – Emphasis on Additive Oils – A natural question arises at this point as to whether the emphasis, perhaps undue, which is being placed on the necessity for additive oils, cannot be relieved by:

1. Further modification in engine design.

2. Maintenance of the oil itself in its original lubricating condition.

The question of the effect of engine design on lubrication requirements has, of course, been thoroughly covered by R. J. S. Pigott, of the Gulf Research and Development Co., in his paper: "Engine Design versus Engine Lubrication,"[a] but it may very well be remarked that there is a difference of opinion among the manufacturers themselves as to the necessity for employing additive oils with their engines. One outstanding manufacturer insists that compounded or additive oils be employed in his engines at all times; another asserts that, owing to certain features of his design, there is no necessity for employing compounded oils in his engine, even under conditions of extreme overload.

Where the doctors disagree, the patient (in this case the engine operator) does not know whom to believe. His inclinations (and sometimes they are most vociferously expressed) naturally favor, however, those engines which do not require special treatment or special oils in order to obtain continuous and satisfactory operation.

On the question of the maintenance of the oil, with its *original* lubricating qualities, it is suggested that insufficient attention has been given to this phase of engine lubrication, not only in the authors' paper, but also in the research programs of many companies. The "maintenance" of oil may well be defined as the elimination of the products of oxidation and other impurities as fast as they appear in the oil in service. Such elimination of impurities is an approach to the problem from an entirely different angle than through the employment of compounded oils, which are designed first, to inhibit the formation of the products of oxidization, second, to keep any solid impurities in suspension through the use of a dispersion agent and, third, to prevent the formation of gums and hard carbon by use of a "detergent" which imparts further solvent properties to the oil. The foregoing combination of characteristics does prevent ring-sticking and other evils due to excessive engine deposits.

It is pointed out, however, that no proof was presented by the authors' paper to the effect that the same results cannot be obtained by the proper maintenance of the lubricating oil, through continuous elimination of both the soluble and insoluble impurities from the oil stream. Such maintenance can be obtained by the circulation of the oil through filtering or clarifying materials which will perform these two functions; that is, filter out solid particles and remove soluble impurities by means of adsorbent clays. There are, of course, several such devices available which have given entirely satisfactory and economical results over long periods of actual service, and have eliminated ring-sticking, reduced maintenance costs, decreased engine wear, and enabled the operator to use his lubricant over longer periods of time.

Are Laboratory Tests Conclusive? – All of the tests which have been cited today, as well as those heretofore published, tend to prove that additive or compounded oils, as compared with straight mineral oils, result in:

a. Freedom from ring-sticking under overloaded, hot, or sustained load conditions. Such tests are breakdown tests and do not necessarily parallel 90% of the conditions met in actual operation.

b. Decreased engine wear. This decreased wear, it should be considered, is possibly a function of ring-sticking, which, in the case of a straight mineral oil, or for that matter of any oil, will result in imperfect lubrication, greater blowby and hence, greater engine wear.

It is perhaps significant that all of the results of tests on additive oils in the authors' paper show extremely high contamination, both soluble and insoluble. For example, in Table 2, solid contaminations of over 10% by weight and dissolved contaminations evidenced by neutralization numbers as high as 2.6 are shown. These are respectively 20 times and 6½ times the maximum limits set up by our laboratories for safe engine lubrication. Therefore, we submit that these tests, while undoubtedly accurate, have no direct bearing on the results that can be obtained by the use of straight mineral oils where contamination is effectively controlled within safe limits.

This effect of excessive contamination is further brought out by the data in Fig. 9 which show that bearing corrosion can be controlled effectively by maintaining a low neutralization number with straight mineral oil as well as by the use of additive oils and, secondly, that the comparative rate of oil oxidation can be controlled effectively by maintaining the neutralization number below our recommended maximum of 0.4.

[a] See SAE Transactions, May, 1941, pp. 165-176.

So far as I am aware, from the literature and discussions with many oil technologists and engine designers, no conclusive laboratory tests have yet been published to disprove the assertion that straight oils, properly maintained, as defined herein, cannot do all that additive oils or compounded oils accomplish, and can furthermore, achieve these same results at a lower overall cost and at greater convenience for the operator.

Proper maintenance of the lubricating oil, as has been defined, involves the use of a device having a capacity adequate for the volume of oil in the system, for the load and horsepower rating of the engine, and for the temperature condition encountered. In other words, the oil maintenance device must be of adequate capacity to remove continuously, for long periods of time, all the products of oxidation, both soluble and insoluble, as fast as they are formed in the engine. This ability to maintain the oil is dependent upon the size of the device, the flow rate of the oil through it, and the temperature encountered, as well as the loading conditions of the engine, and is a problem of design for which the proper solution is always obtainable.

Too often in the past such laboratory or field tests as have been run have involved oil maintenance devices of entirely inadequate size, or of incorrect design for the job in question.

In general, the field results from the practice of oil maintenance tend to prove that engines can be operated continuously with infrequent maintenance requirements, that long engine life can be obtained, and that a low cost per hour or per mile of operation is possible. These characteristics are what the operator desires.

Characteristics of a Straight Mineral Oil Properly Maintained – A straight mineral oil which is properly maintained by the use of an adsorbent type of filter or clarifier has two important characteristics which can also be obtained through the use of an additive agent; namely, a detergent or solvent action, and resistance to oxidation.

A "clean" oil with a neutralization number below 0.4 has distinctly solvent or detergent properties; the addition to a dirty crankcase of such clean oil will dissolve many of the impurities from the crankcase walls as long as the detergent or solvent properties continue. An oil, however, as it builds up its neutralization number and gum content either through use or through contamination from existing impurities in the engine, will lose its solvent properties; which can, however, be restored, or better maintained, through the use of an adsorbent type of clarifier.

A "clean" oil has a much higher resistance to oxidation than a dirty oil where the impurities, whether soluble or insoluble in nature, act as catalysts and further accelerate the breakdown of the oil; therefore, maintaining the oil clean with a low neutralization number and with few or no solid impurities in the oil stream, reduces the tendency to oxidation.

Proper maintenance of the lubricating oil, therefore, prevents the formation of gum, lacquer, or varnish deposits within an engine even under severely overloaded conditions, thus insuring the same type of lubrication, including freedom from ring sticking, which is secured by the use of additive agents in the oil.

Are Additives Necessary?

This very interesting paper clearly demonstrates the action of additives and their characteristics, but there is no mention whatsoever that any thorough tests have been conducted to indicate that a straight mineral oil, properly maintained, will not perform equally as well as an additive oil in the lubrication of heavy-duty diesels. Actual field experience on all types of gas, diesel, and gasoline engines over a period of years would indicate that the desired objectives can be accomplished without the use of so-called additive or compounded oils. Further research along this line is therefore indicated in order to give to the engine designer and the operator a more complete picture.

It is therefore suggested that the engine manufacturers give further consideration to the operators' desires and, as suggested by Mr. Pigott, endeavor to provide engine designs which will function under unusual and difficult conditions without the use of additive oils; and, second, that the manufacturers explore more thoroughly than have the research laboratories, the effect of proper maintenance of the oil through the use of adsorbent types of clarifiers, in place of the additive oils, which present so many disadvantages from the point of view of the engine operators.

A Possible Answer to the Problem – It may very well be that the answer, from the operators' point of view, will consist of the use of a certain stable type of additive or oxidation inhibitor, combined with an adsorbent clay type of clarifier which will permit the use of the oil for extensive periods of time and give a clean appearance, rather than a black oil. This will give the operator some indication of his oil condition and still accomplish all the results which are justly

claimed with the present type of additive oils, without the disadvantages and expense of frequent changes.

There are certain additive oils on the market using primarily an inhibitive agent which can be successfully employed with adsorbent types of filters and, as far as can be determined, give equivalent results to those obtained by other additive oils requiring frequent changes.

Development of Diesel Lubricants

– G. L. Neely

Standard Oil Co. of Calif.

As a representative of the manufacturers and marketers of the first successful commercial compounded diesel lubricating oil, as well as the first non-corrosive compounded diesel lubricant, we are well aware of the magnitude of the problems involved in the development of these products. The authors of the paper are to be congratulated on their able presentation and their contributions in this field.

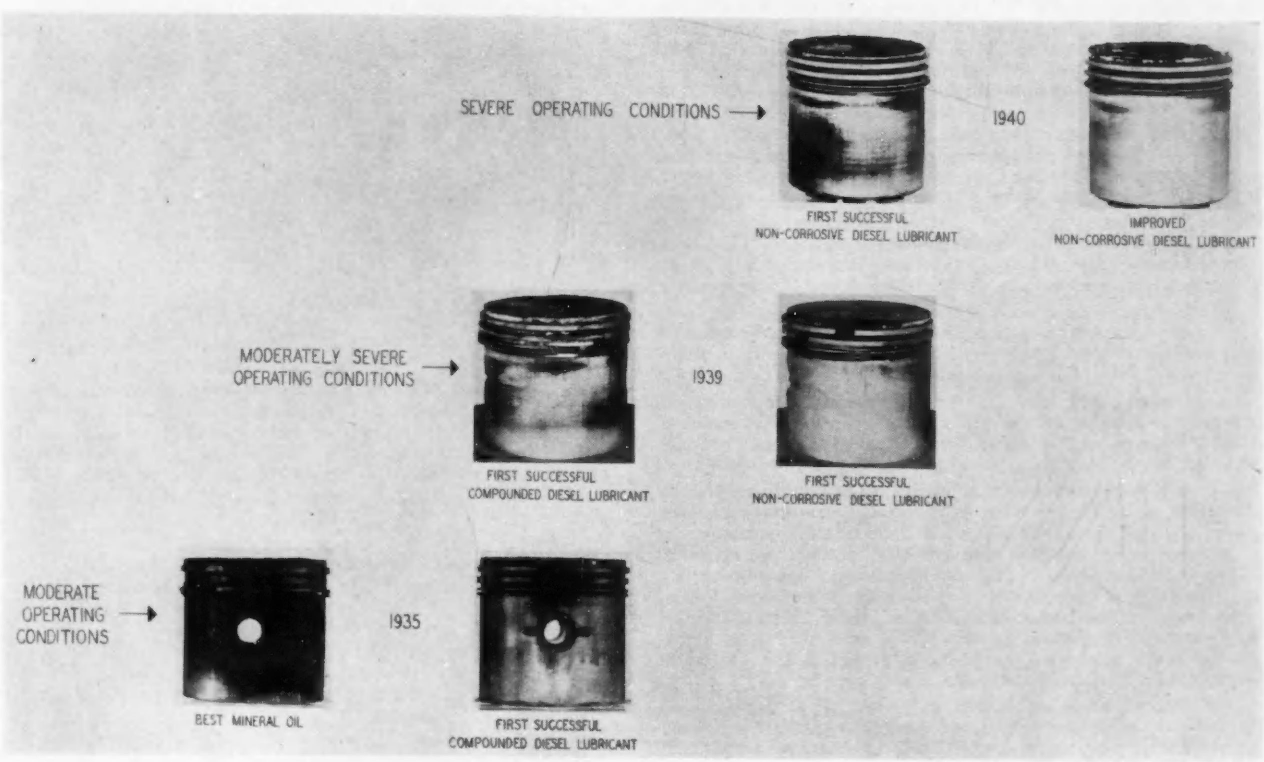
Compounded lubricating oils are now recognized throughout the world as an established necessity for the successful operation of severe-duty engines. While this condition provides a fitting testimonial of the quality of the oils now marketed, it is likely to result in the conclusion that such lubricants are simple propositions and that it is only necessary to add a compound to an oil to obtain an improved product. Our experience has been that the properties of an oil are almost always modified by the addition of any oil-soluble compound, but that 99 out of 100 will be found to have a severe weakness in one or more important service properties. When it is considered that the requirements of each engine differ from those of other engines, some idea of the difficulty of obtaining the optimum combination for general service may be realized. Some engines stress detergency, some stress stability, while others stress both, and it is for such reasons that we have continuously emphasized the idea of "asking the engine," rather than judging compounded oils by specifications or artificial laboratory bench tests.

Progress in the compounded oil field has been continuous since the introduction of the first compounded diesel lubricating oil. From a laboratory point of view this continual improvement in quality has necessitated the use of increasingly severe test conditions. The relation between lubricating oil quality and the severity of the test conditions during the past six years is shown in Fig. C. In 1935, under relatively mild conditions, the difference between an uncompounded mineral oil and the first compounded diesel lubricating oil, was approximately that of day and night. However, as research was continued, room for improvement was made by increasing the severity of the test conditions. The comparison between the first compounded diesel lubricating oil and the first improved product, which incidentally was introduced in 1939 as the first non-corrosive oil, is shown by the second set of piston photographs. Attention is called to the improvement in performance as shown by the reduction in piston deposits and oil ring groove clogging. Although the latter oil was marketed throughout the world and was specifically approved by some 35 diesel engine manufacturers as the result of tests in their own engines, further improvement was made in 1940. The comparison between this non-corrosive compounded oil and the improved non-corrosive oil is shown by the third set of photographs. This comparison was obtained under even more severe conditions than the second set.

These laboratory test conditions are severe enough to take care of almost any service conditions now encountered, but progress has not stopped, and still more severe conditions are used for our experimental work directed towards oils of the future.

Continuous Progress Shown

Our purpose in showing these comparisons from laboratory tests is to emphasize the fact that progress in compounded lubricating oils has been continuous since their inception. Our laboratories in Richmond are now engaged in a broad research program investigating new compounds and combinations of compounds that are very promising, and we feel sure that still further progress will be made during the next several years. At the same time it is our opinion that these developments will be found applicable, not only in diesel lubricants, but also in the motor oil and aircraft fields, and that the time will come when all high-grade lubricants for internal-combustion engines will be compounded.



■ Fig. C (Neely discussion) – Stages in the development of diesel lubricants

1942 CAR DESIGN TRENDS

by THOMAS A. BISSELL
Technical Editor, SAE Journal

IN designing the 1942 passenger cars, automobile engineers have served two masters – and served each of them well. They have met their obligation to the government by eliminating from their designs many thousands of tons of critical materials urgently needed in building the nation's defenses and, at the same time, they have satisfied the expectations of the buying public by packing their new cars with fully as many improvements as offered in any normal model year. Examples in the following pages describe hundreds of redesigned parts appearing in the 1942 models in which both of these twin objectives have been achieved – critical materials have been eliminated by substitution of alternate materials and the part has been improved simultaneously in performance, durability, or appearance.

For the third successive year, developments in the drive and its control stand out among the mechanical improvements made. Studebaker, Hudson, and Mercury-Lincoln announce new semi-automatic drives, and previously announced automatic and semi-automatic units have been refined and improved. Instead of radically new mechanisms, the new drives feature various combinations of tried and proved devices – fluid couplings, vacuum-operated clutches, semi-automatic transmissions, and overdrives – assembled and controlled in ingenious ways to produce novel and flexible means of driver control.

Eight engines had been shifted to iron or steel pistons at the time of their announcement, and most of those that still carry aluminum pistons are scheduled to change within the very near future. In spite of these changes, with a few exceptions, the 1942 engines have been made more powerful, and many have higher compression ratios. A Buick engine innovation is the rough-surfacing of crankpins, which is claimed to aid lubrication and increase bearing capacity and life. Introduction of this process has touched off a sharp debate on the relative merits of smooth versus rough bearing journals.

Riding comfort has been increased in a number of makes by revising spring rates, introducing spring leaf inserts of various materials, and improving spring lubrication. Buick has increased ride stability to such an extent by adoption of wide-base tire rims that one stabilizer has been eliminated and another made lighter. This company also pioneers with a foot-operated parking brake. A greater proportion of the braking effort has been shifted to the front wheels in most of the brake changes made. Chrysler introduces a white wheel trim ring which extends over tires and gives the illusion of white side walls.

A trend in body design carried out in many of the 1942 models is the extension of the front fenders so that they flow into or through the front doors, and some of the rear fenders now flow into the rear doors. A number of bodies have been lowered through changes in the frames, springs, and/or wheel and tire size, and a few have longer wheelbases. Outside running boards have all but disappeared, being available only on a few very low-priced cars and several high-priced conservatively styled models. An innovation worthy of mention is the recessed headlamps found in 1942 DeSotos which have sliding flush-type shutters to conceal and protect them in the daytime. Directional signals appear on additional lines, and are further standardized. Another novelty in car lighting is the concealed running-board illumination introduced by Hudson.

All models appear with more or less of their customary decorative chrome at the time of their announcement. As is well known, all makers are scheduled simultaneously to eliminate all decorative bright metal on some date within the next few months, when stocks of these decorative materials are exhausted, to conserve critical materials, and all manufacturers have completed their plans for this change.

Considerable redesign of heating and ventilating systems is reported. Most of it is concerned with locating the fresh air intakes at the front of the cars and making their controls more sensitive.

General

Packard's 1942 line is featured by an extension of the new Clipper bodies to all models in the line: the 6, the 8, the 160, and the 180. In addition to the four-door Clipper sedan introduced last April, a two-door Clipper club sedan (business coupe without back seat) makes its bow. Wheelbase of the 6 has been lowered 2 in. to 120 in., and this same chassis is used for the 8, the only difference, beside the engine, being in the radiator, clutch, and brakes. This 8 replaces the former 120. Clipper 160 and 180 models have a 127-in. wheelbase. The weight distribution of Clipper models is changed, with both engine and passengers moved forward with relation to the wheels.

In addition, a number of "Senior" Packard models are offered, carrying the more conservative body lines as exemplified in the 1941 models. These models include three convertible coupes; a 6 on 122-in. wheelbase; and an 8 and 160 each on a 127-in. wheelbase. Senior models in the

[This paper was presented in part before a meeting of the Chicago Section of the Society, Chicago, Ill., Oct. 14, 1941.]

160 and 180 Packards are available again this year in the 138- and 148-in. wheelbases.

All three models of Hudson's 1942 line – the 6, the Super 6, and the Commodore Series – are ¾ in. lower and are provided with concealed running boards.

Chevrolet introduces a new body, the "Aerosedan," a close-coupled two-door sedan with a rear contour that forms a straight streamlined arc from the roof to the rear bumper, closely resembling those introduced on other General Motors cars in 1941. This body is 2¼ in. lower in overall height than the Chevrolet Town sedan, and 1½ in. lower at the front and rear seats. Its rear seat is 3¾ in. farther forward. Chevrolet weights are slightly higher, about 25 lb at the front wheels and 10 lb at the rear wheels.

Pontiac and Oldsmobile have re-named their lines. The Pontiac 119-in. wheelbase line is now called the Torpedo; and the two 122-in. wheelbase models are called the Streamliner and Streamliner-Chieftain respectively. As was the case in the 1941 models, either a 6 or an 8 may be had in any car.

The 119-in. wheelbase line of Oldsmobile, now called the "Sixty," offers either a 6-cyl or 8-cyl engine, and has added a club sedan with streamlined back. The former 125-in. wheelbase "Dynamic" line is now referred to as the "Seventy-Six" or "Seventy-Eight," depending upon whether it is powered by a 6-cyl or 8-cyl engine. The former Custom series is now termed the "Ninety-Eight," and is available only with 8-cyl engine. Wheelbase of this line has been increased 2 in. to 127 in.

Wheelbase of two Cadillac models has been increased – the 60 from 126 to 133 in., and the 62 from 126 to 129 in. Both models are provided with new two-window bodies that are about 1½ in. lower in overall and front-door floor heights.

Wheelbase also has been increased on the Buick 50 and 70. The increase is 3 in. in each case, from 121 to 124 in. on the 50, and from 126 to 129 in. on the 70. New, wider bodies appear on these two models in which the front fenders are carried back along the doors to join the rear fenders. Convertible phaetons have been dropped from the 1942 Buick line.

Plymouth bodies have been lowered 1½ in. due to a change in frame construction. All Chrysler products are now furnished only with concealed running boards.

All Ford and Lincoln models have been lowered 1 in.; Mercurys are 1½ in. lower. All Fords except the low-priced Special model are available with either a 6-cyl or an 8-cyl engine; the Special is furnished only with the 6-cyl engine. All cars made by Ford have concealed running boards.

Engines

Although 8 passenger-car engines already had been shifted from aluminum-alloy to cast-iron or cast-steel pistons at the time of their announcement, the upward trend in horsepower and compression ratio continues. Often engine bore or stroke was increased, exhaust back pressure reduced by opening up the exhaust system, or manifold restrictions eliminated at the same time that changes were made to heavier cast-iron or cast-steel pistons, with the net result that power was increased. In some cases, connecting rods of heavier section or stronger material, larger or different-type bearings of greater fatigue life and/or heavier crankshafts have been required simultaneously to accommodate the heavier pistons.

Horsepower of Plymouth's engine, now equipped with cast-iron pistons and a heavier crankshaft, has been boosted to 95 at 3400 rpm from 87 at 3800 rpm by increasing the bore from 3⅛ to 3¼ in. Compression ratio has been increased from 6.7:1 to 6.8:1.

The DeSoto and Chrysler 6-cyl engines also attribute their increased horsepower to an increase in bore – from 3⅜ in. to 3 7/16 in. in both cases. The DeSoto engine is now rated at 115 hp at 3800 rpm instead of 105 hp at 3600 rpm and the Chrysler 6-cyl engine is now rated at 120 hp at 3800 rpm instead of 112 hp at 3600 rpm. Compression ratios have been reduced from 6.8 to 6.6:1 in both cases. The DeSoto engine was announced with cast-iron pistons, and the Chrysler 6 with aluminum-alloy pistons. On the other hand, the increase in horsepower of the Dodge 6-cyl engine, announced with aluminum-alloy pistons, of from 91 hp at 3800 rpm to 105 hp at 3600 rpm, is due to an increase in stroke of from 4⅜ in. to 4⅝ in., and to raising the compression ratio from 6.5:1 to 6.7:1.

With aluminum pistons and the same compression ratio of 6.8:1, horsepower of the Chrysler 8 is up from 137 at 3400 rpm to 140 at 3600 rpm. The power was boosted by eliminating manifold restrictions, redesigning the cylinder head, and enlarging exhaust pipes.

Oldsmobile's 6-cyl engine is announced with cast-iron pistons and the 8-cyl engine with cast-steel alloy pistons. Compression ratios of both have been raised to 6.5:1 in redesigning the combustion chambers. In 1941 the compression ratio of the 6 was 6.1:1 and the 8 was 6.3:1. New thin-babbitt bearings of greater fatigue life are introduced on all main and connecting-rod bearings. The crankshaft and connecting rods have been made heavier on the 8-cyl engine. The net result of these changes is that the horsepower ratings of the two engines are the same as last year – 100 for the 6 and 110 for the 8.

A 5-hp increase is recorded for each of the three Packard engines, all of which are announced with aluminum-alloy pistons. The 6 is now rated at 105 hp; the "120" 8 at 125 hp; and the large 8 at 165 hp; all ratings are at 3600 rpm. Increases in compression ratio of from 6.39:1 to 6.7:1 for the 6; and from 6.41 and 6.45:1 to 6.85:1 on the 125-hp and 165-hp 8's, respectively, are primarily responsible for the power boosts. The new thin-babbitt bearings have been adopted by Packard.

All three Studebaker engines have been shifted to cast-iron pistons; horsepower ratings and compression ratios are specified as the same as in 1941.

After the production of approximately the first 4700 engines, Willys-Overland will shift to pistons of cast iron alloyed with molybdenum. Horsepower is specified at 63 at 3800-4000 rpm, and compression ratio at 6.48:1 for either aluminum-alloy or cast-iron pistons.

Combustion chambers of the Nash Ambassador 6 and 8 have been redesigned to increase turbulence, particularly at the base of the single spark plug, and to carry the water jacketing up higher, especially around the valves. The flywheel of the Nash Ambassador 600 also has been redesigned, and weighs 10 lb less.

As announced with aluminum-alloy pistons, compression ratios of all three Nash engines have been stepped up: the 600, from 6.7:1 to 6.87:1; the 6, from 6.3:1 to 6.5:1; and the 8, from 6.3:1 to 6.6:1.

Power ratings of the Buick light 8, which has been shifted to cast-iron pistons, have been revised downward, and the compression ratio has been dropped. With single carburetor it is now rated at 110 hp at 3400 rpm instead of 115 hp at 3500 rpm; with compound carburetion the 1942 rating is 118 hp at 3600 rpm as compared with 125 hp at 3800 rpm. Compression ratio with single carburetor has been lowered from 6.5:1 to 6.00:1, and that with compound carburetion from 7.00:1 to 6.3:1. The compression ratio of the 165-hp 8, announced with aluminum-alloy pistons and employing compound carburetion only, also has been lowered – from 7.00:1 to 6.7:1.

Horsepower rating of the Ford 8 engine has been stepped up from 85 to 90 by reduction of back pressure through larger exhaust pipes and muffler, and refinements in valve operation and carburetion. The Mercury 8 horsepower also has been boosted from 95 to 100 by combustion-chamber changes, larger intake ports, and reduction of back pressure through larger exhaust pipes and muffler. Mercury compression ratio is now 6.4:1. A 1/16-in. increase in bore, now 2 15/16 in., and redesigned combustion chambers in a new cast-iron instead of aluminum-alloy cylinder head are chiefly responsible for boosting the power of the Lincoln 12-cyl engine from 120 to 130 hp. Compression ratio remains the same at 7.00:1.

Cylinders and Valves

Elimination of aluminum-alloy cylinder heads and of the nickel content in valves has necessitated a number of design changes announced under this heading. Furthermore, adoption of cast-iron pistons has required suitable changes in the combustion-chamber design and compression ratios of a number of engines.

The combustion chambers of the Nash Ambassador 6 and 8 have been redesigned to increase the turbulence, particularly around the spark plugs and to carry the water jacketing up higher, especially around the valves. Single ignition is employed instead of twin ignition used in the past. Combustion-chamber redesign also is announced on the Chrysler 8 and both Oldsmobile 6 and 8. An important change in the Mercury combustion chamber is a slot, 2⅛ in. wide, that is now milled between the cylinder wall and the valve ports so that the available area for the admission and expulsion of gases is increased. Lincoln has shifted from an aluminum-alloy to a cast-iron cylinder head with redesigned combustion chamber.

With the elimination of nickel from valves of all Ford products a deeper cantilever section was required in the valve head to produce the required hot strength, and the angle of the under side of the valve head was increased from 8 to 15 deg. Chrome-molybdenum alloy steel now replaces chrome tungsten for valve seat inserts in Fords, Mercurys, and Lincolns.

The pressure angle of the timing chain teeth on Pontiac 6-cyl engines has been increased from 17 to 22 deg to eliminate chain whip and increase chain life.

Cams on the camshaft of Chrysler engines are now tapered with the high side to the rear. The cams now hit the bottom of the tappets slightly off center, and this action causes the tappets to revolve.

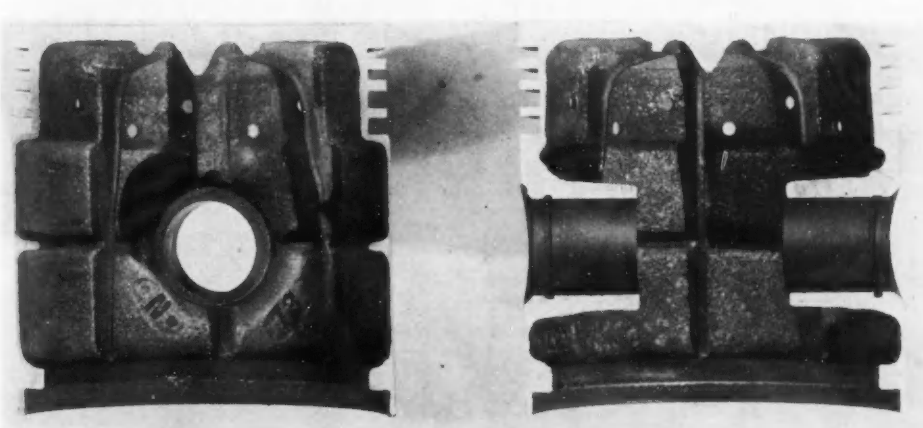
Pistons and Rings

New cast-iron or cast-steel pistons on 8 of the 1942 engines overshadow developments in this field. These pistons weigh from 42 to 84% more than their aluminum-alloy predecessors, and are protected by either a tin or insoluble phosphate coating.

Plymouth's cast-iron pistons, designed to accommodate the larger 3¼ in. bore of the 1942 engine, weigh 25.29 oz as compared with 14.5 oz for the aluminum-alloy pistons in the 1941 engine with a bore of 3⅛ in. They are cam ground and employ chilled iron ribs to reinforce the piston walls. The sidewall of these cast-iron pistons measures 0.035 to 0.050 in. at its thinnest point. New pistons on the Oldsmobile 8-cyl engine are of cast-steel alloy, and are said to weigh only 42% more than their aluminum-alloy predecessors. It is claimed that these steel-alloy pistons are lighter than comparable cast-iron pistons which weigh as much as 84% more than equivalent aluminum-alloy pistons because, owing to the greater strength of this material, it can be cast in thinner sections. The 6-cyl Oldsmobile engine has shifted to cast-iron pistons.

The cast-iron or cast-steel pistons used in all Studebaker, Plymouth, DeSoto, and the Buick small 8 engines are given a protective insoluble phosphate coating by the "Lubrizing" process; those for Oldsmobile are tin-plated as will be those used in Willys Americars.

To provide the additional strength and stiffness required



■ Cross-sections of Plymouth's new cam-ground cast-iron piston. It is 3¼ in. in diameter, weighs 25.29 oz, and has chilled iron ribs to roinforce the piston walls

for cast-iron pistons on its smaller engine, which weigh about 75% more than their aluminum-alloy predecessors, Buick now uses a stronger steel, SAE 1340 manganese, for its connecting rods, instead of SAE 1040 formerly employed. In addition, connecting rods are shot-blasted. On its 8-cyl engine, Oldsmobile uses a stiffer connecting rod that is about 5% heavier.

Piston rings on the Pontiac 6 will be tin plated this year, and the smoke holes in the skirt have been removed.

Redesign of the Mercury piston has been necessitated by the slot that has been milled between the combustion chamber and the cylinder wall. The top land has been made wider, moving the top ring down so that it is not exposed directly to the hot gases entering and leaving the cylinders.

Crankshafts and Ventilation

Adoption of cast-iron and cast-steel pistons has necessitated heavier crankshafts with larger bearings on some engines. On others, a change to the new thin-babbitt bearings has provided the required additional bearing fatigue strength. Vibration dampers have been added on several crankshafts. Several unique features appear. Buick introduces an unorthodox "rough" finish on its crankpins, claimed to increase bearing life and capacity; Lincoln has come out with a "flexible" flywheel said to cushion crankshaft deflections; and Chrysler introduces "tapered-wall" bearings.

Because of the increase in bore and shift to cast-iron pistons, the crankshaft of the Plymouth engine has been made heavier and a vibration damper added for the first time. Main bearings are now 2½ in. in diameter instead of 2¼ in.; connecting-rod bearings are 2 1/16 in. instead of 1 15/16 in. in diameter. The crankshaft of the 8-cyl Oldsmobile engine also has been made heavier.

Packard has adopted the new thin-babbitt bearings on both connecting-rod and main bearings. High lead babbitt, of thickness ranging from 0.0045 to 0.008 in., is used. To provide the additional fatigue life necessary with cast-iron and cast-steel pistons, Oldsmobile has adopted for all main and connecting-rod bearings thin-babbitt bearings of the type that employ a porous matrix to bond the babbitt to the steel backing.

In shifting the Studebaker Champion engine from aluminum-alloy to cast-iron pistons, a vibration damper similar to that used on the Studebaker Commander engine has been added to the Champion crankshaft.

Tapered-wall main and lower connecting-rod bearings are introduced by Chrysler. These bearings are thicker on the tops than on the sides and are claimed to equalize the pressure all the way around the crankshaft and make the bearings dimensionally exact.

Buick introduces a new rough finish on its crankpins in its 1942 crankshafts. Instead of the regular machining, grinding, and lapping operations that give crankpins a highly polished surface with an average depth of surface depressions of 4 to 5 micro-in., the crankpins are first rough-ground and finished to take off the tops only of the irregularities to about 40 to 70 micro-in., so that the surface contains thousands of microscopic depressions. These depressions are claimed to serve as minute oil reservoirs that help to maintain the oil film under heavy load. The rough-surfaced journals are said, therefore, to increase bearing life, and to increase the capacity of the bearings to stand heavy loads. It is reported that this new development has contributed to the ability of the light 8 Buick engine to shift to heavier cast-iron pistons without change in bearing size.

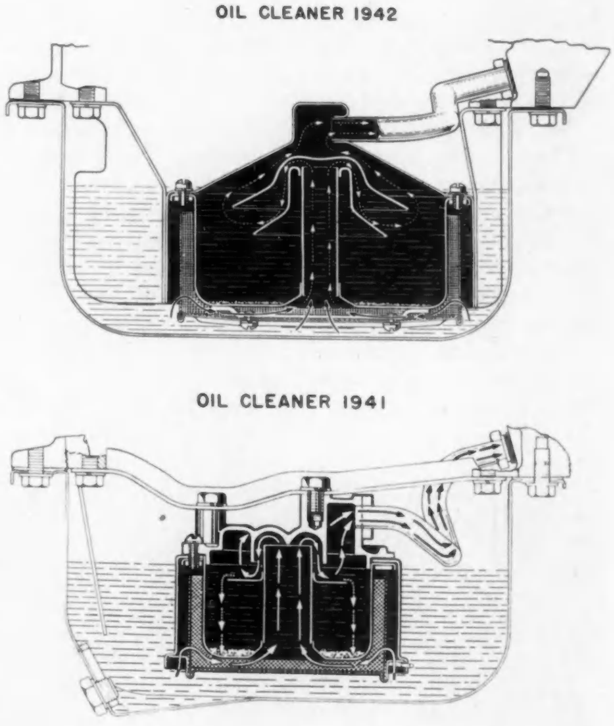
Lincoln introduces a "flexible" flywheel in which a number of "spokes" between the hub and the rim are said to provide a certain amount of flexibility in cushioning small crankshaft deflections caused by the power impulses of the engine. The Nash 600 flywheel also has been redesigned and its weight cut from 30 to 20 lb to improve acceleration and to increase top speed.

The oil-filler pipe through which air is admitted to the crankcase ventilator on all Packards has been made longer, raising the air inlet above the top of the engine.

Lubrication

Redesign of the Pontiac oil cleaner to eliminate its aluminum-alloy cover features the lubrication developments.

In their drive to eliminate critical materials from their cars, engineers often have had to redesign parts completely to accommodate alternate materials. In some cases it has been possible at the same time to improve the part. Such a case is the Pontiac oil cleaner. Redesign was undertaken



▪ Comparison of 1941 and 1942 Pontiac oil cleaners. In the 1942 design the aluminum-alloy cover has been eliminated and the efficiency improved

to eliminate its aluminum-alloy die-cast cover and replace it with a steel stamping, but other design changes were made simultaneously which made the cleaner markedly superior. The dirt-retaining basin was made four times larger; the oil outlet was made concentric with the basin so that each drop of oil would travel approximately the same distance through the cleaner and at practically identical velocity with every other oil particle. This design change is reported to do away with the eddies formerly caused by unequal velocity of oil currents just before they reached the outlet, tending to disturb dirt particles as they are settling and to permit them to re-enter the oil stream. The third design feature is that the contour and shape of the path through the cleaner has been established by test to improve the efficiency of the cleaner. The 1942 Pontiac filter has an efficiency of from two to three times the original filter for fine dirt that will just go through an 80- to 100-mesh screen.

A different type of oil filter with larger capacity has been adopted as standard equipment on all Chryslers. The interleaf spring method of lubrication used in the leaf springs of Ford and Mercury cars has been changed so that on top of each leaf one groove, three times larger in area than the two grooves formerly used, now carries the lubricant. Lincoln cars use rubber inserts between the leaves.

The oil pressure gage on the instrument panel of the 1942 Packard 6 and 8 is now electrically operated, a feature previously found only on the Packard 160 and 180.

The oil drain plug at the bottom of the engine oil pan on Willys Americars has been moved from the middle to the side of the oil pan to facilitate draining; also the oil dip stick has been lengthened on all Willys products.

Cooling Systems

Wider, lower radiator grilles and novel bumper constructions have changed the location of the air inlets to the radiator cores and arrangement of baffling in a number of cars. Other developments include revision of cylinder cooling control, and changes in fan type and drive, and in water-jacket design.

The new wider, shorter shape of the radiator grille and the deeper-sectioned front bumper of 1942 Chevrolets cause different airflow characteristics and necessitate the use of a new radiator core. This core has a frontal area 5½ in. greater; it is both lower and wider. Heat-transfer characteristics, however, remain essentially the same.

In addition, a metal plate has been added to cover the space between the top of the grille and the top of the radiator core to prevent air from passing above the core instead of through it. A covered hole several inches in diameter is provided in the plate above the grille and core to permit access to the front of the core in order to clean it. Because the 1942 Chevrolet radiator core is supported through the inner edges of the long front fenders, the brace rods which formerly extended from the top of the radiator to the dash have been eliminated in the 1942 models.

A novel feature of the front-end construction of 1942 Plymouths and Chryslers is an additional air intake or scoop located behind and under the bumper. In construction, the splash pan is pulled down to form the lower part of the additional air intake, and the upper part of the passage is formed by the gravel shield that connects the front bumper and radiator grille. The air is delivered to the scoop under and through the bumper and is guided by suitable baffles to the radiator core. On Oldsmobiles, a new double bumper construction locates the radiator grille between the two bumper bars.

Fans on the Ford and Mercury 8-cyl engines are now mounted on a separate bracket midway between the generator shaft and crankshaft and are driven from the generator by a separate belt. Formerly these fans were mounted on the crankshaft but, with this separate drive, fans are now driven at 1.11 times crankshaft speed. In addition, a 16-in. fan with 4 blades evenly spaced replaces the former fan with 6 eccentrically spaced blades. Cooling water passages between the cylinder block and cylinder head have been recalibrated on Ford and Mercury engines to provide more even cylinder cooling.

Cadillac has substituted the blocking type of thermostat for the radiator shutters used for many years in its cooling system. When starting, it shuts off the water circulation in the left cylinder block on which the heater take-off is attached. Both engine and heater warm-up time are reduced with this arrangement, the heater warming up in about half the time previously necessary. Also, there is a 10% increase in the Cadillac frontal ventilation area.

To improve cooling, water jackets have been carried up higher, especially around the valves, in the new cylinder heads employed in the 1942 Nash Ambassador 6 and 8.

An additional seal has been added on the Pontiac water pump. It consists of a slinger pressed on the shaft between the rubber water-pump seal and the ball bearing. The slinger seal was added to pick up drops of moisture which may get past the rubber seal and throw them into a circular surrounding groove. A hole drilled through from the under side of the water-pump body to the groove provides drainage.

Fuel Systems

Back pressure of Ford and Mercury 8-cyl engines has been decreased by increasing the diameter of the exhaust pipes, both muffler pipe and tail pipe, and by employing a larger muffler which is oval-shaped to give more ground clearance.

An automatic signal is provided on all Packard Clipper fuel tanks which whistles continuously while the tank is being filled until it is within 1 gal of full. When this level is reached, the signal stops, thus warning against overflow. Another feature of Clipper fuel tanks is the downward extension of the gasoline filler pipe almost to the bottom of the tank. With this arrangement, the gasoline enters the tank below the surface with a minimum of agitation.

Carburetors of Willys Americars are equipped with automatic choke for the first time this year.

The manual throttle control has been eliminated on all 1942 Pontiacs. A small change in the fast idle cam of the 1942 carburetor, which slightly increased the initial idling speed for cold starting, contributed to make the manual throttle control superfluous.

The wire link which ties the float-valve pin to the float lever on Pontiac dual carburetors used in the 8-cyl engine also is employed this year in the single carburetors of Pontiac's 6-cyl engine. This link has been added to prevent sticking of the float valve pin caused by gummy fuel deposits.

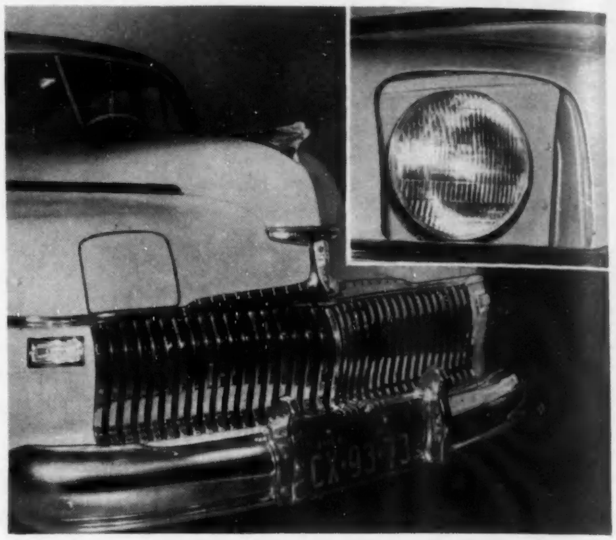
Equipping the carburetor with a non-stalling device is one of the features added with the Liquamatic drive used on Lincoln and Mercury. This device opens the throttle slightly whenever the engine vacuum drops below the normal vacuum for engine speed. It consists of a diaphragm control operated by engine vacuum which prevents the throttle from closing against the stop under such conditions.

Buick has modified its compound carburetion set-up slightly to improve hot-weather running and low-speed idling. One of these changes is the addition of a lockout on the rear or auxiliary carburetor that prevents its valve from opening until the engine is warm. The rear carburetor no longer can come in until the choke is off. The lockout is thermostatically controlled.

The Buick crankcase ventilator tube has been relocated to eliminate the possibility of oil fumes getting into the carburetor. Now the outlet of the crankcase ventilator is run to the center of the oil-bath air cleaner so that the oil fumes from the ventilator are collected in the oil-bath cleaner. An oil drain has been added to the air cleaner to handle oil collected from the ventilator outlet.

Capacity of all Buick gasoline tanks has been increased 1 gal. The 40, 50, 60, and 70 Buicks now have 19-gal tanks, and the 90 has a 22-gal tank.

Cadillac gasoline tanks have deeper ribbing and rearranged baffles.



■ DeSoto "steel eyelid" headlamp design. Headlamps are recessed into the front fenders as shown in the upper insert, and are covered during the daytime by flush-type sliding panels whose movement is controlled from the driver's compartment

Electrical Systems

Revisions in the electrical system have been made all through the cars, and a novel headlamp construction is introduced.

In order to get the distributor on the camshaft center between the cylinder block and the fan, now located midway between the generator shaft and crankshaft on Ford and Mercury 8's, necessitated a flat distributor design with its coil located on the engine instead of on top of the distributor head. Vacuum spark control is introduced in Nash Ambassador 6 and 8 distributors.

Buick has gone back to 14-mm spark plugs for all engines, replacing the 10-mm size introduced on the 1941 models. The change was made in conjunction with the lowering of compression ratios.

The starter control has been moved from the instrument board to the accelerator on all Packard models. Long, narrow batteries are located under the hoods of all Packard Clippers; the 6 and 8 use a 15-plate, 100 amp-hr battery, whereas the 160 and 180 use a 17-plate, 120 amp-hr unit.

Tops of all General Motors convertibles are now operated by electric motors, one on each side of the body, instead of the vacuum cylinders used previously. Two buttons on the instrument board control the movement by means of a two-way switch.

DeSoto introduces a novel feature in the "steel eyelids" or sliding shutters used to cover its headlamps in the day time. The headlamps are recessed in the crown of the fenders so that, when they are covered by the shutters, the shutters continue unbroken the contour of the fenders. The shutters are opened for night driving and closed for day driving by means of a handle located under the instrument panel. Pulling the lever opens the shutters, turns on the lights; pushing it closes the shutters, turns the lights off.

The Chevrolet stop-light switch has been relocated and operates differently. Instead of being mounted at the rear of the break main cylinder and operated by the fluid in the brake system, the new stop-light switch is inserted between the brake pedal and the main cylinder and is operated mechanically. The change was made to eliminate the possibility of a switch failure causing leakage in the brake fluid system.

To simplify the control of the car lighting system, Pontiac has combined the head and tail-light switch and the instrument light switch into one unit. When the switch button is pulled out to either parking or driving position, the instrument lights will come on at the same time. A rheostat connected in the instrument light circuit permits the driver to increase or decrease the intensity of illumination on the instrument panel and clock by rotating the light switch knob clockwise for brighter and counter-clockwise for dimmer. When turned as far as possible in the counter-clockwise direction, the instrument and clock lights are turned off.

Cadillac's back-up light is now controlled by a new automatic switch that is operated by the transmission lever when it is thrown into reverse position. Also, Cadillac's tail lamp and rear direction signal are separate units this year.

The concealed running boards on Hudson Super 6 and Commodore models are equipped with courtesy lights, built into the door frames at the bottom, which illuminate the running boards at night when any door is opened.

For 1942 Cadillac has adopted the three-way ignition lock with a third position when the engine is off and the accessories on, in addition to the regular on and off positions. The entire edge-lighted dial of Plymouth's speedometer now changes color with different speeds, instead of the pointer only.

Engine Mountings

The two engine radius rods formerly used in conjunction with the Ford and Mercury torque-tube drives have been eliminated to minimize the conduction of engine noise

to the frame and body, and a new rubber cradle-type rear engine mounting that takes the end thrust is now used.

Clutches

Friction of the clutch release mechanism on Mercury and Lincoln cars has been lowered by mounting the throw-out levers on needle bearings.

To remove a power throb that was being transmitted from the engine through the clutch housing, a more flexible mounting of the clutch housing to the engine is employed on the Nash 600. The housing is mounted rigidly at the bottom, the bolts running through a steel block. At the top, however, the corresponding block is made of rubber about 5/16 in. thick and will permit about 0.020 in. movement of the engine and/or clutch housing.

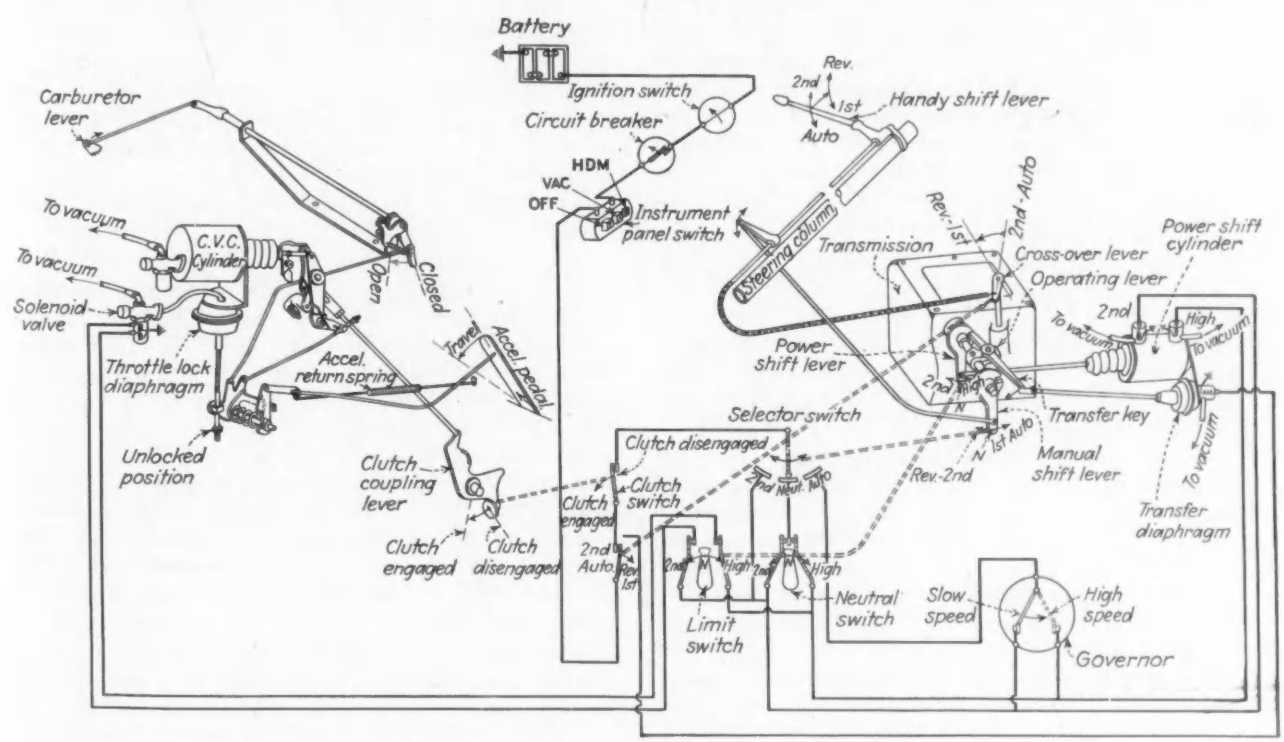
Transmissions

Three new semi-automatic drives introduced by Hudson, Studebaker, and Mercury-Lincoln contribute the outstanding mechanical innovations in the 1942 cars. Rather than introducing new mechanisms, all three of these drives employ combinations of well-known devices which have been in successful use for some time, such as fluid flywheels, vacuum-operated clutches, and/or conventional clutches in combination with vacuum-operated semi-automatic transmissions with or without overdrive. The three drives vary considerably in the types of units combined and the methods by which they are controlled under various driving conditions.

Hudson "Drive-Master"

Hudson's "Drive-Master" offers unusual flexibility of operation. This drive combines Hudson's vacuum-operated "Vacumotive" clutch, available for many years as optional equipment on Hudson cars, and a conventional clutch, with a three-speed semi-automatic transmission. The only change in the clutch is that a throttle lockout has been added, which will be explained later. This semi-automatic transmission consists essentially of Hudson's conventional three-speed transmission adapted for semi-automatic operation between second and third gears, and provided with a second-gear ratio of 1.82:1 to facilitate starting in second gear. Last year, the second-gear ratio of the Super 6 and Commodore Hudson transmissions was 1.65:1.

The fore-mentioned flexibility of operation is obtained by permitting the driver to choose any one of three driving methods by pressing one of three buttons on the instrument panel marked: "Off," "Vac," or "HDM." Depressing the "Off" button will permit conventional operation of the gearshift lever and clutch. Pushing down the "Vac" button will give semi-automatic clutch operation, the same as would be obtained in a 1941 Hudson equipped with a Vacumotive clutch. Pressing the "HDM" (Hudson Drive-Master) button cuts in the combined semi-automatic clutch and transmission system. Another feature that contributes to the flexibility of the system is that gears may be operated manually or the clutch operated conventionally any time the driver desires, regardless of which of the three driving methods is in operation, or of car speed. As in the past, a



■ Complete layout of Hudson "Drive-Master" system

vacuum cylinder controlled by solenoid-operated valves is provided to operate the "Vacumotive" clutch; in addition a similar vacuum cylinder is used to shift the gears on the semi-automatic transmission and a transfer diaphragm is employed to disengage the transfer key from the hand-shift lever and to engage it with the power shift lever. A single centrifugal-type governor switch with two sets of contacts is used to control the speeds at which the semi-automatic transmission can be shifted from second to high and vice-versa, as well as to control the operation of the Vacumotive clutch. Hudson's overdrive also may be added to the "Drive-Master" system to give additional ratios in both second and third gears. In such case, the same governor switch also regulates the operation of the overdrive, a separate terminal being provided on the switch cover for this purpose. An accompanying illustration shows schematically the complete layout of the system.

When it is desired to operate the semi-automatic drive starting from rest, the "HDM" lever is first depressed, the clutch is thrown out with the foot and the gearshift lever shifted manually from neutral into high. When the gear-shift lever is thus shifted, the circuit to the solenoid valve controlling the transfer diaphragm (shown at the far right in the illustration) is closed, admitting vacuum to the transfer diapraghm which disengages the transfer key from the hand-shift lever and engages it with the power-shift lever.

Now, as the accelerator is depressed, the clutch engages automatically and the car starts to move forward in second gear. When a speed of about 13 mph or more has been reached, the centrifugal governor switch will permit the power shift vacuum cylinder to shift the gears from second to high when the accelerator is released. If the accelerator is not released, however, the car will stay in second gear, regardless of speed. Similarly, with the car in high gear, the shift back into second can be made by releasing the accelerator whenever the car speed has dropped below about 10½ mph; the shift-back cannot be made until the foot is lifted from the throttle, regardless of speed. As in the past, the "Vacumotive" clutch disengages when the accelerator is released when the car speed is under about 20 mph, and engages when it is depressed. Thus, practically all normal driving can be done with the brake, throttle, and steering wheel.

If for any reason such as climbing long, steep grades, a conventional second speed is desired above the shift-back speed when the car is in "Drive-Master," it can be obtained by throwing out the clutch with the foot and shifting the gearshift lever manually to the second-gear position. The car will stay in second gear until the gearshift lever is again shifted manually.

Stopping is accomplished by stepping on the foot brake. During stopping the transmission is shifted automatically from high to second gear as the foot is lifted from the accelerator, and the car is ready for a new start. Shifts into and out of low gear and reverse are accomplished manually in the conventional manner. Low gear is recommended only for rough or muddy driving conditions, or for negotiating extremely steep grades.

The throttle locking device was added to the "Vacumotive" clutch to prevent clash and noise when the gears are being shifted. In operation of this device, a solenoid valve is energized, admitting vacuum to an additional diaphragm known as a "throttle lock diaphragm" (shown below the clutch vacuum cylinder at the left in the illustration). The action of this diaphragm pulls up on a diaphragm cable which acts through suitable linkage to lock the throttle so that there is no response to further depressing of the accelerator.

This drive, with or without overdrive, is offered as optional equipment on all Hudson models.

Studebaker "Turbo-Matic" Drive

Studebaker's "Turbo-Matic" drive is offered as optional equipment on Commanders and Presidents. It consists essentially of a fluid coupling, a semi-automatic clutch, and semi-automatic overdrive transmission, with suitable mechanical and electrical control. The semi-automatic overdrive transmission comprises Studebaker's conventional three-speed transmission combined with an overdrive of the type that is controlled entirely by inserting or withdrawing a pawl from the collar of the sun gear to lock or release the overdrive. An accompanying illustration gives a schematic layout of the entire Turbo-Matic drive.

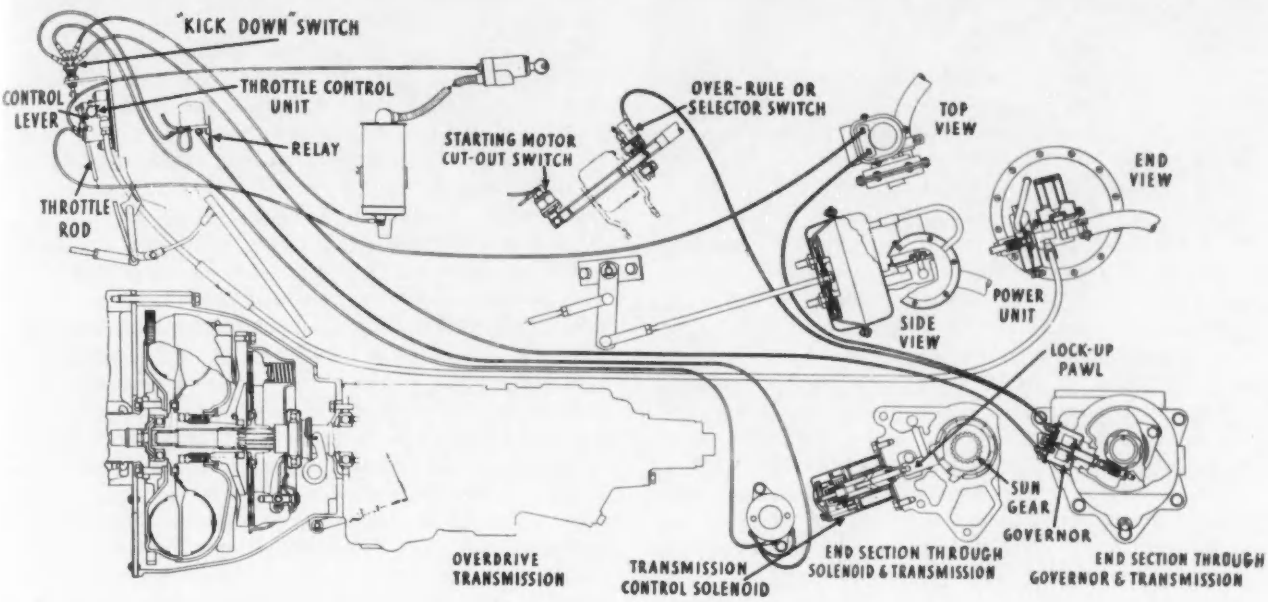
Because of the semi-automatic action of the clutch, which is controlled by the throttle and a centrifugal governor, the clutch pedal is eliminated. The transmission, in conjunction with this clutch, is semi-automatic in operation. For most city driving, the second-gear position of the gearshift lever and semi-automatic operation in the "traffic range" between second gear and second-gear overdrive are recommended. For road driving in the "cruising range," however, the high-gear position of the gearshift lever is specified with semi-automatic operation between high gear and high-gear overdrive. Thus only an occasional manual gearshift between second and third gears is required for most normal driving. Use of low gear is recommended only for starting under extremely poor road conditions.

At speeds above 14 mph, the change from conventional to overdrive is accomplished by lifting the foot momentarily from the accelerator pedal; the return to conventional gear can be made at any speed by pressing the accelerator all the way through. In addition, as the car coasts down to speeds below 6 mph, a centrifugal governor acts automatically to release the overdrive and return the car to conventional gear.

The torque of the engine is transmitted at all times through the fluid coupling.

A conventional single-plate clutch is located at the rear of the fluid coupling. Engine manifold vacuum, acting on a diaphragm connected to the clutch release mechanism, automatically operates the clutch. With the car at a stand-still and, for example, the shift lever in second gear, or traffic range, the clutch is kept disengaged by a governor mounted on the transmission which dictates through a solenoid that the clutch be kept disengaged when the car is stopped or traveling below approximately 6 mph. The solenoid operates a valve which connects engine vacuum to one side of the diaphragm, thus keeping the clutch disengaged.

Initial depression of the accelerator pedal opens the switch in the throttle control mechanism, breaking the circuit to the solenoid-operated valve in the power unit to manifold vacuum line. As this circuit is broken, the valve is closed and the vacuum on the back side of the power diaphragm is reduced to a pre-determined point (controlled by an adjustable vacuum regulator) which allows initial engagement of the clutch. Up to this point of operation, the carburetor throttle valve has not moved from the idle posi-



• Schematic layout of Studebaker "Turbo-Matic" drive showing controls for the clutch and overdrive units and wiring diagram

tion (this is accomplished by the use of a spring-loaded sliding joint in the throttle linkage) but, as the accelerator pedal is further depressed, the atmospheric bleed valve, operated by the throttle linkage, and the carburetor throttle valve begin to open simultaneously which allows further engagement of the clutch as the engine output torque increases. The time for complete clutch engagement is controlled by the amount the accelerator is depressed beyond the point at which the throttle linkage-operated bleed valve begins to open.

To shift gears, the throttle is released and the shift lever is started towards its new position. The initial movement of the lever, through a switch in the gearshift controls, completes the circuit to the power unit solenoid. This releases the clutch and the shift is completed. The clutch re-engages by the opening of the throttle as previously described.

It will be seen that three controls affect clutch operation: (1) a governor on the transmission which disengages the clutch below speeds of 6 mph; (2) a control in the throttle mechanism which can overrule (1) and engage the clutch; and (3) a control on the shift mechanism which disengages the clutch at start of shift and permits re-engagement when shift is completed. The clutch will not disengage unless throttle is closed.

Control of the overdrive is by a transmission-mounted solenoid which operates a pawl to lock or release the overdrive. At speeds below 14 mph, the governor driven off the rear of the transmission keeps the circuit to the transmission solenoid open. The pawl will then be at the unlocked or outward position. Thus, all starts are in conventional ratio.

At speeds above 14 mph, the governor completes the circuit to the transmission solenoid. Then, as soon as the foot throttle is released, the solenoid pushes the pawl to the locked or overdrive position.

When the accelerator is pushed clear to the floor, a switch in the throttle mechanism breaks the circuit to the transmission solenoid, permitting the return spring to pull out the pawl to the unlocked position. Consequently, a conventional ratio is instantly available for periods of rapid acceleration.

The transmission range selector (shift control) is on the steering column as in previous Studebakers.

To eliminate the possibility of starting the engine while in gear, a switch on the lower end of the gearshift control forestalls operation of the starting motor unless the range selector is in neutral position. A key on the gearshift shaft depresses a ball in the switch only when the shift lever is at neutral position. This action closes the switch contacts and permits operation of the starter motor.

Mercury-Lincoln "Liquamatic" Drive

The new "Liquamatic" drive, which is offered as extra equipment on Mercury and Lincoln, comprises essentially a fluid coupling, conventional clutch, and a semi-automatic three-speed transmission provided with an overrunning clutch on the transmission countershaft. In addition, when used on the Lincoln only, an overdrive is employed to provide two extra speeds in the forward semi-automatic driving range. A conventional shift lever and clutch are employed to place the drive in one of three possible positions: reverse, an emergency low recommended only on extremely steep grades and in heavy sand or mud, and the forward driving range in the high-gear position which is used under all ordinary driving conditions.

In operation as installed in the Mercury car, the gearshift lever is shifted from neutral to the high position in the conventional manner, and left there for all normal driving conditions. All ordinary driving from then on is done by operation of the accelerator pedal and clutch. In starting, the drive is through the fluid flywheel, clutch, and second-gear train in the transmission. The ratio of this second gear has been raised to 1.83:1 for quicker get-away. At any

time after the car speed has reached approximately 12 mph, the centrifugal governor of the automatic transmission will permit the vacuum cylinder to shift the transmission from second to high when the driver's foot is lifted momentarily from the accelerator. For additional acceleration at speeds below 35 mph, the transmission can be shifted back to second gear by pushing the accelerator pedal all the way down. Above 35 mph, the transmission will remain in high gear. When the speed of the car decreases to below 10 mph, the gears are shifted automatically from high to second. Thus the transmission is always returned to second gear ready to start as the car slows down during stopping. The slipping action of the fluid flywheel permits the car to be stopped at any time simply by applying the brakes.

If it is desired to use the engine as a brake, as in descending steep grades, or when the car is pushed or towed to start the engine, the gearshift lever may be placed in second gear in the conventional manner and the transmission is "locked" in second gear. The semi-automatic operation and the overrunning clutch on the transmission countershaft are locked out as the gearshift lever is shifted to the second-speed position. The clutch pedal may be used in the conventional manner at any time.

When an overdrive is added in the Lincoln installation, six forward speeds instead of three are made available to the driver as follows: first, first-overdrive, second, second-overdrive, high, and high-overdrive. Overall gear ratios of these six forward speeds are respectively as follows: 10.32:1, 7.24:1, 8.12:1, 5.66:1, 4.44:1, and 3.11:1.

As in the Mercury arrangement the high-gear position of the gearshift lever is used for all normal driving but, in the case of the Lincoln drive, four forward speeds are available in the semi-automatic range – second, second-overdrive, high, and high-overdrive. The difference in operation is as follows: After the car has accelerated to a speed of about 12 mph in second gear, the accelerator is released momentarily, causing the transmission to shift into high gear as before. But, after a speed of 23 mph has been attained, a second similar release of the accelerator pedal causes the transmission to shift into high-overdrive. However, if the car is started from a standstill and allowed to accelerate up to 23 mph without previously releasing the accelerator pedal, and the pedal is then released quickly, the transmission will shift into second overdrive. With the next momentary release of the accelerator, above 23 mph, the drive will skip the regular high-gear position and go into high-overdrive. The reason for this action is that the shift is made faster in the overdrive than in the transmission; the overdrive and semi-automatic transmission are controlled by separate governors and work independently.

When driving at any speed over 23 mph, it is always possible for the driver to shift into a lower gear to provide additional power for acceleration by depressing the accelerator pedal clear to the floor. If the speed is between 23 and 35 mph, the transmission will shift back from high-overdrive to second-overdrive, the first time that the pedal is depressed to the floor. If depressed a second time, it will shift from second-overdrive to second gear, providing still faster acceleration. If the speed of the car is above 35 mph, only one kickdown is required – from high-overdrive to high gear.

As in the Mercury, when the car slows down below 10 mph as for a stop, the transmission is returned automatically to second gear.

The low-overdrive ratio can be obtained by lifting the foot momentarily from the accelerator when a speed of 23 mph or more has been attained with the gearshift lever in the emergency low-gear position.

The shift to second is made in the same way as with the Mercury when it is desired to use the engine as a brake, except that, in addition, a button on the instrument panel must be pulled out to lock out the overdrive. The overrunning clutch of the transmission is locked when the gearshift lever is moved to the second-gear position as explained previously. An indicator light on the instrument panel tells whether or not the overdrive is in operation.

The overdrive used in the drive is the same available in 1941 Lincolns and has a ratio of 1:1.428. It is again available in connection with a conventional transmission and clutch on 1942 Lincolns.

The Mercury rear-axle ratio at 3.54:1 is the same whether or not Liquamatic drive is used. Lincoln uses a 4.44:1 ratio for the Liquamatic drive instead of the standard 4.21:1 ratio. The fluid coupling used with the drive on the Lincoln has 24 vanes on one member and 27 on the other, the different numbers being selected to avoid synchronous noise. The Mercury fluid coupling has the same diameter but has fewer vanes on each member. The overrunning clutch on the transmission countershaft drive gear was required to permit this gear to free-wheel when the transmission is locked up in high since the second-speed gears are always engaged.

Another feature of the Liquamatic drive is that the carburetor is equipped with a non-stalling device which opens the throttle slightly whenever the engine vacuum drops below the normal vacuum for engine speed. The device consists of a diaphragm control operated by engine vacuum which prevents the throttle from closing against the stop under such conditions. Two-speed axles are no longer offered on cars built by Ford.

A torsional vibration damper has been added to the "Hydra-Matic" drive employed by Cadillac. The damper is similar to the spring-loaded type used in clutch plates. It comprises 8 coil springs arranged circumferentially in the hub of the driving member of the fluid flywheel. The springs are driven from the flywheel member by pins to which friction plates are riveted. The damper was added to damp out engine torsional periods that occasionally caused a slight rumble at speeds around 30 mph. On the Oldsmobile Hydra-Matic drive, hydraulic synchronization of the reverse gear teeth has been improved for easier reverse-gear engagement.

Two other transmission changes have been made by Cadillac – the transmission mainshaft is now shot-blasted and the gearshift lever is located on a ring that is concentric with the steering column and rotates around it.

Chrysler's "underdrive" semi-automatic transmission, used in conjunction with fluid drive, is now available on Chrysler 8-cyl models; it is standard equipment with fluid drive on the Crown Imperial. Overdrive is no longer available on Chrysler 8's.

The fluid coupling used on Dodge, DeSoto, and Chrysler 6-cyl models has been redesigned to increase its efficiency and to give quieter operation throughout the driving range. The number of fins on the driving member has been increased to 48, and on the runner, to 44. The shape of the driver has been changed so that less reservoir oil space is needed.

To increase shifting leverage, longer gearshift levers are employed this year on Ford, Mercury, and Lincoln.

Rear Axles

With heavier cast-iron or cast-steel pistons and accompanying slower top engine speeds and with three new semi-automatic transmissions, there has been considerable re-alignment of rear-axle gear ratios in the 1942 cars.

With the shift to heavier cast-iron pistons, the rear-axle ratios of Studebakers with conventional transmissions have been changed from 4.56:1 to 4.1:1 on the Studebaker Champion, and from 4.55 to 4.09:1 on the Commander and President models. Optional gear ratios are 4.56:1 with overdrive for the Champion and 4.55:1 for the Commander and President. All Buick standard rear-axle ratios except the long-wheelbase 40 and the 60 have been raised. The short-wheelbase 40 is 4.1:1; the 50 has been increased from 4.1:1 to 4.4:1; the 70, from 3.9:1 to 4.1:1; and the 90, from 4.2:1 to 4.55:1. Rear-axle ratios of all Chrysler products also have been re-aligned. Plymouth's has been reduced from 4.30:1 to 3.9:1; Standard Dodge rear-axle ratio in sedans and convertible coupes has been reduced from 4.3:1 to 4.1:1; in club coupes, from 4.1:1 to 3.9:1. With fluid coupling the rear-axle ratio in Dodge sedans, club coupes, and convertibles is now 3.9:1 instead of 4.1:1. The axle ratio of the Dodge three-passenger coupe is 3.73:1 with or without fluid drive. The Standard DeSoto rear-axle ratio has been decreased from 4.1:1 to 3.9:1, those of 3.54:1 and 3.73:1 for use with underdrive transmission remaining the same. Chrysler New Yorker and Saratoga 8's ratio with underdrive transmission is now 3.36:1, and the Crown Imperial standard ratio with underdrive is 3.58:1. Lincoln uses a 4.44:1 rear-axle ratio with Liquamatic drive instead of the standard 4.21:1. Ford and Mercury rear tread has been increased 1¾ and 1¼ in. respectively, to 60 in.

Brakes

Many changes in the 1942 brakes are toward increasing the proportion of the total braking effort taken by the front wheels. In most cases the front wheels are given a greater braking capacity either by increasing the brake width or by increasing the size of the front wheel brake pistons. Several innovations are introduced in the control of parking brakes; Buick announces foot-pedal operation and Cadillac now uses a T-shaped lever.

The Packard 1942 6 employs 12 x 1¾ in. brakes in front and 11 x 1¾ in. brakes in the rear. Last year both front and rear brakes were 11 x 1¾ in.

Packard 160 and 180 Clippers have 12 x 2¼ in. brakes on front and 12 x 2 in. brakes on the rear, and the Senior 160 and 180 models of 138 and 148-in. wheelbase have 12 x 2¼ in. brakes on both front and rear.

Pontiac front brake width has been increased from 1¾ in. to 2 in., and the diameter of the front brake wheel cylinders has been stepped up from 1 to 1 1/16 in. Both front and rear brake drum diameters remain 11 in., and the rear brake width continues at 1¾ in. In addition, the "triple brake seals," employed on Pontiac rear brakes since 1935, have now been extended to the front brakes which formerly were provided with dual seals. The added seal consists of a stamping attached to the backing plate at the rear of each wheel brake.

Heavier brake drums are used in 1942 Cadillacs. To permit more uniform expansion of the brake drum and to stiffen it, and also to serve as a labyrinthian type of seal against the entrance of dirt and water, a flange has been added on the edge of the brake drum nearest the center of the car.

Cadillac also introduces a T-shaped hand-brake lever located several inches below the instrument panel on the left-hand side of all models. To apply the hand brake, the T-shaped lever is pulled out; to release the brake, the lever is twisted. In addition, a rubber boot has been added on the end of Cadillac hand-brake cables to seal out water and dirt.

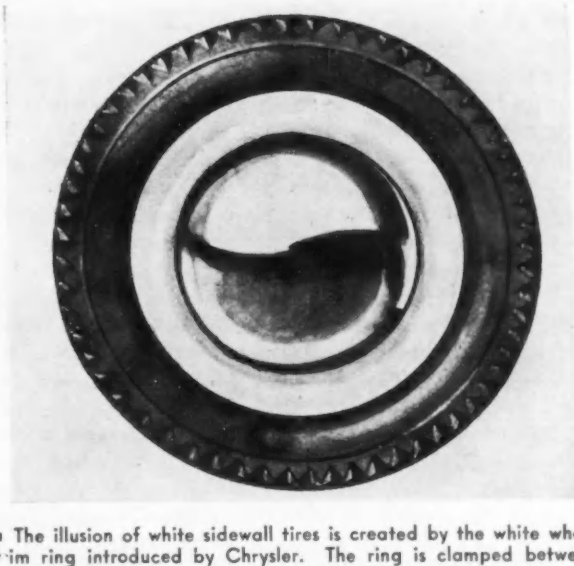
Buick introduces a new type of parking brake control called the "StepOn." Instead of the hand lever, this design employs a small foot pedal control located above the floorboard on the extreme left of the car. Depressing this pedal with the foot engages the parking brakes, and pushing a T-shaped hand control on the lower left corner of the dash sets them. Brakes are released by pulling out the same hand control. The remainder of the system is essentially the same.

Increases in brake width also are announced by Oldsmobile. On the 119-in. wheelbase 6 and 8 the increase is from 1¾ in. to 2 in., and the "Seventy-Eight" and "Ninety-Eight" lines have 2¼ in. brakes this year instead of the 2-in. width of comparable 1941 models. All brake diameters continue at 11 in.

The distribution of braking effort on Ford and Mercury cars has been changed to 60% front and 40% rear from 55% front and 45% rear in 1941 cars, by increasing the size of the forward-shoe brake piston on the front wheels from 1¼ in. to 1⅜ in. in diameter; all other brake wheel pistons remain the same size.

Wheels, Rims, and Tires

Buick has gone over to the new wide-base rims on its 1942 models. The rim width of the 40 and 50 is 6 in., 1 in. more than last year; at 6½ in., the 60, 70, and 90 rims



■ The illusion of white sidewall tires is created by the white wheel trim ring introduced by Chrysler. The ring is clamped between the hub cap and tire and extends out over the tire rim

are $1\frac{1}{2}$ in. wider than in 1941. Tire sizes on all Buicks remain the same as last year, although the recommended air pressure has been reduced to 25 psi.

To give a dressy appearance similar to white-wall tires that have been discontinued to save rubber and zinc oxide for defense, Chrysler has developed a white wheel trim ring which clamps into the space between the hub cap and extends over the tire rim to give the impression of white tire sidewalls from a distance. It is standard equipment on all Chryslers except the Royal.

Mercury and Lincoln now use 15-in. rims and 6.50-15 and 7.00-15 tires, respectively, instead of 16-in. rims and 6.50-16 and 7.00-16 tires. This reduction in rim size contributed $\frac{1}{2}$ in. of the total $1\frac{1}{2}$ in. and 1 in. that Mercury and Lincoln bodies, respectively, have been lowered.

6.50-15 four-ply tires are used on Packard 6's and 8's except the convertible coupes which use 7.00-15 four-ply as do the 160 and 180 Clippers. Long-wheelbase (138 and 148-in.) Packard 160's and 180's use 7.00-16 six-ply tires.

The 1942 Studebaker President is provided with 7.00-15 tires instead of the 7.00-16 size employed last year.

Rear Suspensions

A number of changes in rear suspension have been made to permit frames and bodies to be lowered, to revise spring rates, and to improve lubrication.

Because of the increased tire and wheel stability gained by the adoption of wide-base rims, Buick has been able to eliminate the rear stabilizer on all models. Buick also has reduced the rate of its rear coil springs by using a longer length of smaller gage wire. In conjunction with these changes the rear shock-absorber calibration has been revised. In the Buick 50 and 70 the rear springs are now placed ahead of, rather than over, the rear axle to reduce the side-rail deflection.

Frames of Fords and Mercurys have been lowered 1 in. and Lincolns $\frac{1}{2}$ in. in the rear by changing the camber of the softer rear spring. To prevent bumping and to equalize the ride with increasing load, the rear spring has been given a variable rate – it stiffens as it goes down about 10 or 15%. This variable rate has been effected by having the spring go through the flat position and slightly reverse itself with heavy load before it bumps. The rear springs have fewer leaves this year and a lower rate. The rear tread has been increased $1\frac{3}{4}$ in. for Ford, $1\frac{1}{4}$ in. for Mercury, and Ford and Mercury rear springs are now 49.2 in. long instead of 48 in. 1942 Ford and Mercury now have a single large groove in each leaf for interleaf lubrication instead of the two smaller grooves employed previously.

A new transverse strut or stabilizer is introduced on 8-cyl Chrysler models. The stabilizer is mounted behind the axle and extends from the rear-axle housing to the chassis frame. A telescopic shock absorber is mounted at the connection of the stabilizer with the frame. On 6-cyl models, the stabilizer is mounted ahead of the axle.

Wax-impregnated interleaves or liners, introduced last year on the Crown Imperial, are now furnished on the grooved rear springs of all Chrysler 8-cyl models. Metal spring covers are said to be unnecessary on springs provided with these liners.

Rear leaf springs of all Packard Clipper models are inset from the frame rails, producing a tapered mounting in which the springs are closer together at their front extremities than at the rear, thus increasing resistance to side shocks.

In order to lower the overall body height of all Hudsons by $\frac{3}{4}$ in., all frames are kicked up 1 in. more at the rear axle, and the rear springs dropped $\frac{3}{4}$ in. To accommodate this lowering of the rear springs, their rate has been increased 10 lb per in.

Shock-absorber links in the rear suspension of 1942 Chevrolets have self-contained rubber bushings at both ends instead of at one end only.

To quiet the action of its rear leaf springs, Pontiac has added oil-impregnated hardwood liners. Three liners are employed per spring – two 1/16-in. thick go between the top and second, and second and third leaves; and one 1/28-in. thick separates the third and fourth leaves. Each hardwood strip extends about $\frac{1}{2}$ in. beyond the ends of the lower leaf that it protects.

Front Suspensions

Changes in front suspensions have been made for many reasons – to lower frames and bodies, to revise spring rates, to improve lubrication, and to accommodate new wide-base rims or increased car weights.

Ford and Mercury front tread has been increased $2\frac{1}{4}$ in. for Ford, $1\frac{3}{4}$ in. for Mercury, to 58 in., and the length of both front springs has been increased from $42\frac{1}{2}$ to 44 in. Lincoln front tread is now 59 in., $2\frac{1}{2}$ in. more, and front spring length has been increased from $44\frac{1}{2}$ to $45\frac{1}{4}$ in. Ford and Mercury springs now have a single large groove instead of two smaller grooves for lubricant. Redesign of the axles permitted Ford and Mercury front springs to be set $\frac{1}{2}$ in. lower, contributing $\frac{1}{2}$ in. of the total amount that these frames have been lowered in front. In addition, Ford, Mercury, and Lincoln frames have been lowered $\frac{1}{2}$ in. by changing the camber of the front spring. This spring change necessitated redesigning the tie-rod, locating it on top of the steering spindle arm instead of beneath it.

Since the redesigned front springs have a lower rate, the rigidity of the stabilizer in the front end of Ford and Mercury cars has been increased by stepping up its diameter by 1/16 in., and a new transverse radius bar or "track" bar has been added to take care of side forces in steering and to insure alignment of axle and frame. This "track" bar extends from the steering gear to the right end of the axle, parallel to the transverse drag link.

The long torque arms employed in other Packard front suspensions are not used in Packard Clipper models because of the division of the braking load between the upper and lower levels of the suspension effected in these models.

The coil springs in Chevrolet front suspensions have been made heavier to compensate for the 25 lb increase in car weight at the front of the car.

Because of the increased stability obtained with the new wide-base rims, Buick has been able to use a lighter front stabilizer. Buick front shock absorbers now have larger orifices in the valves to soften the action of the front suspension.

Frames

New wider and lower bodies, longer wheelbases, and changes from outside to concealed running boards are responsible for the majority of changes made in the 1942 frames.

Plymouth has shifted from the X-type of frame with channel side members to the box or "double-channel" type of frame used on all other Chrysler products. This change has permitted the bodies to be mounted 1½ in. lower than in the 1941 models. Body and outrigger brackets on the frames of all cars made by Chrysler have been changed to accommodate concealed running boards, and the extreme front of the frames has been changed to give more support to the front bumper by lessening the overhang of the bumper bars.

Thickness of the side rails of Oldsmobile frames has been increased, and a cross member has been added at the front to accommodate Oldsmobile's new double bumper front-end construction.

New frames accommodate Packard's wider, lower Clipper bodies. They employ box-section rails both fore and aft of the X-member which is 9½ in. deep at the cross-over. Five cross-members are used per frame.

Lowering the overall height of all cars in the Hudson 1942 line by ¾ in. was achieved by kicking up the frame over the rear axle 1 in. more, and by dropping the rear springs ¾ in. in each model.

By increasing the thickness of the stock used on the side members and X-members, Pontiac has increased the weight of its Torpedo cabriolet frame by 70 lb.

Frames of the Cadillac 60 and 62 and Buick 50 and 70 have been lengthened to suit the increases in wheelbase of 7 in., 3 in., 3 in., and 3 in., respectively. The Cadillac 60 frame has been stiffened by increasing the side-bar thickness from 9/64 to 5/32 in. Buick 50 and 70 frames are wider in the rear to accommodate the new, wider bodies.

Steering Systems

A new transverse radius bar, called a "track" bar has been added to the front end of Fords and Mercurys to take care of side forces in steering and to insure alignment of axle and frame. This bar extends from the steering gear to the right end of the axle, parallel to the transverse drag link. Mercury and Ford steering column tubes are of heavier tubing and the wheel now turns on a self-adjusting ball bearing in the top of the column just under the steering wheel.

Buick introduces a universal type of steering-gear mounting in which a bracket raises the gear so that a depression in the frame at this point is no longer necessary. This mounting has a self-aligning feature which permits adjustment to suit the driver.

Shifting away from steels containing considerably larger percentages of critical materials, Chrysler has extended the use of Amola steel to the following important steering parts throughout the entire Chrysler line: steering shaft, steering worm, roller tooth, tie-rod bolts, and kingpins.

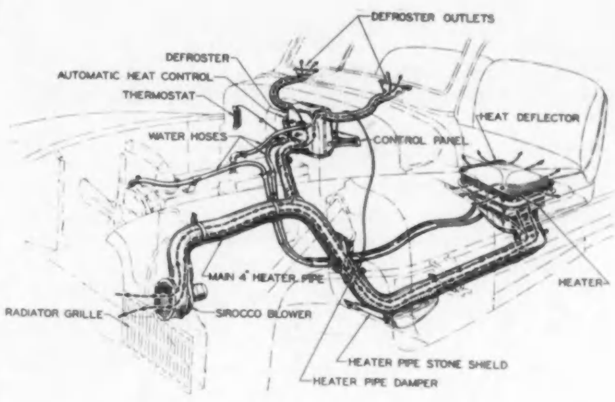
A two-spoke steering wheel with a horn button in the center will be furnished as standard equipment on Pontiac Torpedo and Streamliner models. Standard on Pontiac Streamliner-Chieftains and extra equipment on the other two lines is the 1942 "Safety-Flex" wheel in which the horn has been extended to form a complete circle making it easier to blow the horn when the steering wheel is turned. Full horn rings also appear this year on Plymouths, DeSotos, Chryslers, Dodges, Cadillacs, and Oldsmobiles.

In all Packard Clippers and convertible coupes, a needle bearing is used at the top of the vertical wheel support instead of the bushing used previously. The turning radius of the Packard Clipper is 21 ft for the 6 and 8, and 22 ft for the 160 and 180. Needle bearings are employed in the upper ends of the kingpins of all Willys-Overland cars this year.

Equipment

Extensive redesign of several heating and ventilating systems highlights the developments in this field. Virtually all systems are now provided with fresh air intakes located at the front of the car to utilize the air pressure induced by the car's forward motion. Directional signals appear on additional lines and are further standardized. Trumpet-type steel horns replace the zinc-alloy seashell type formerly used.

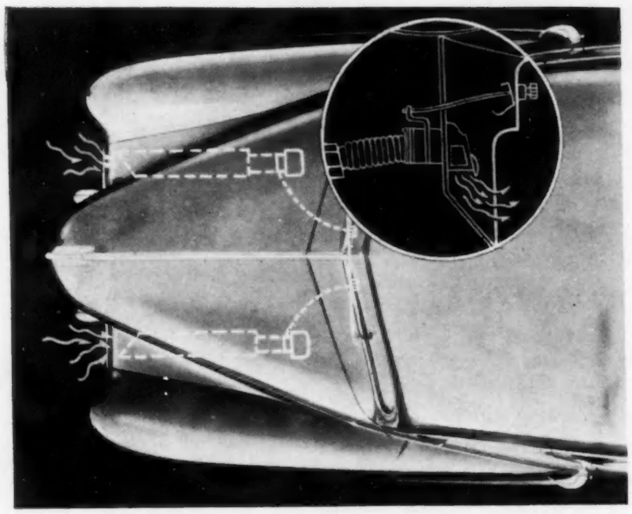
Pontiac engineers consider the redesign of the Pontiac under-seat heater as one of the most important mechanical changes in their 1942 models. This year, instead of recirculating the air inside the body, the heater draws 100% fresh air from outside. A larger-capacity electrically driven blower, now located just below the left headlamp and back of the radiator grille, forces fresh air through a pipe to the heater core as shown in an accompanying illustration. Thus, the interior of the car is placed under positive pressure, with the movement of air outward. The heating element or core, through which hot water from the engine water jackets is circulated by the engine water pump, is



■ Pontiac 1942 underseat heater

located in the floor directly under the driver. A thermostatically operated valve regulates the flow of hot water through the heating element under the seat, and the car temperature can be set at the desired value by means of a knob on the control panel. A branch from the 4-in. pipe connecting the blower and heater leads to the defroster assembly placed in the center of the dash. If it is desired to augment circulation of heater air with cooler fresh air from the blower, a knob on the control panel, which operates a damper permitting air to enter the defrosting slots unheated, can be turned. Turning this same knob to the "de-icing" position starts a blast of hot air through the defroster slots. At 35 mph the rush of air through the heater pipe is sufficient to operate the system with the blower turned off.

Cadillac's under-seat heater has a new outside air inlet, new controls in the radio grille, and a higher-output windmill-type fan on the defroster. With the elimination of cowl ventilators on all Cadillac models, the air is now taken in through two front openings, one in each fender dust shield, through screens in the baffles and is conducted into the car interior. Advantages claimed for this arrangement of air intake are that it is rain-proof and takes full advantage of the air pressure induced by the forward motion of the car.



■ Diagrammatic view of 1942 Cadillac ventilating system showing new front-located air intakes

Buick's 1942 heating and ventilating system also features two new fresh air intakes located on either side of the car just behind the radiator grille. The central heating unit is located under the front seat. Operation of the blower is not recommended at speeds above 30 mph. Automatic temperature control is provided.

The air intake of the Nash "Weather-Eye" heating and ventilating system is now permanently open as the cowl ventilator through which outside air is drawn into the system is now fixed in an open position. The advantage claimed for this arrangement is that passengers are protected against the former possibility of forgetting to open the cowl ventilator when the system is in use.

The thermostat capillary tube of the system is now located on the lower dash close to the floor boards directly in the stream of air being emitted by the system, and the control valve knob for the Weather Eye is now mounted on the body dash.

Turn signals are provided on all 1942 Packards with the exception of the Special models, and are optional equipment on Lincoln and Mercury cars. They are of the type with automatic turn-off when the steering wheel is straightened up after a turn.

Hudson's turn signals are provided with automatic turn-off for the first time. In addition, the control switch now comprises a lever mounted on the left side of the steering column instead of the three buttons formerly located directly beneath it.

A change in equipment made in practically all 1942 cars to conserve zinc is the replacement of the seashell or snail type horns made of zinc die castings with long trumpet-type horns of steel. They are claimed to be equal or superior in tone quality and warning ability to the zinc-alloy horns; the chief advantage of the latter is that they are more compact. To facilitate blowing the horn when the steering wheel is being turned, full circular horn rings appear on Cadillac, Pontiac, Oldsmobile, Dodge, Chrysler, DeSoto, and Plymouth.

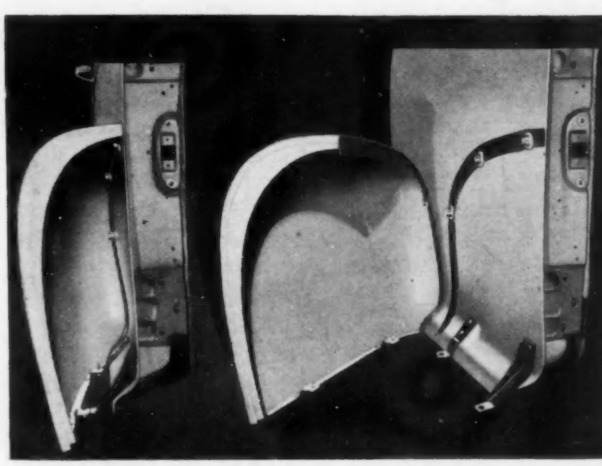
Dual horns are standard equipment for the first time this year on all Plymouths and Dodges.

The diameter of the vacuum cylinder for the Pontiac radio antenna has been increased ⅜ in. to 1¼ in. to increase its lifting power.

Cadillac announces a new bumper jack with an extra-heavy base to replace the wheel jack formerly available.

Grilles and Sheet Metal

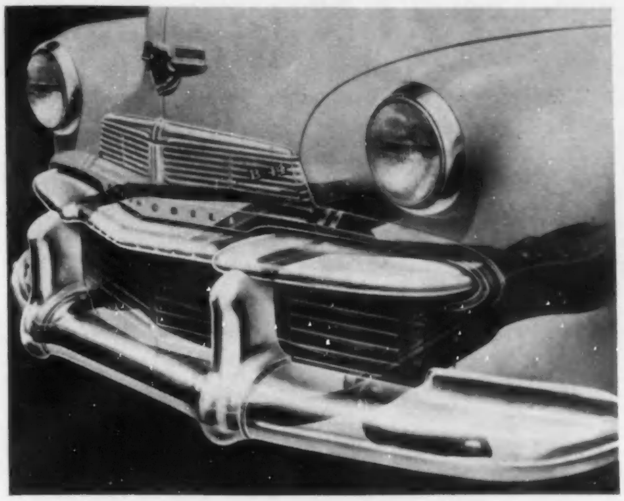
The 1942 models continue the trend toward wider, lower, heavier-appearing radiator grilles. In many models the grilles appear to extend across the full front width of the car, the effect being obtained by dummy grilles extending across the front of the fenders. In keeping with this



■ Construction of Chevrolet front fender extension or "fender cap," and method of bolting to front door

general trend, headlamps have been spread as much as 15 in. farther apart, front bumpers are wider and heavier, and parking lights have been re-positioned. Circular buttons on each side of the radiator grilles on several models can be used either to mount fog lights or to serve as decorative medallions.

Extension of the front fenders so that they sweep back into or through the front doors is one of the outstanding innovations in appearance of many of the 1942 models. In some of these bodies the rear fenders also extend forward into the rear door. Many of the new bodies are stamped originally in this form but, on a number of General Motors models, the rear end of the fender which swings with door, or the "fender cap," is a dummy because the main fender ends in a vertical steel wall where the fender cap begins. The wall is curved to provide clearance for the cap when the door is opened. The fender cap is not a part of the door but is a separate part bolted to the door in a manner shown in an accompanying illustration. Removal for repair or replacement is easy because the bolts are accessible when reached from underneath the cap. The bottom of the cap aligns with the bottom of the door, and underneath it and conforming to its contour is a shelf-like extension from the entrance step which the door conceals. This extension can be utilized as a foot scraper.



▪ Oldsmobile "double-duty" bumper construction

▪ This photograph illustrates a number of changes made in the Chevrolet front end, including the wider and lower radiator core, the new metal baffle plate extending between the tops of the radiator grille and core, and the new splash shield

In extending the fenders on some models it was found that the door-opening geometry had to be revised because the fender cap interfered with the main portion of the fender when the door was opened. In the former Chevrolet door, for example, the lower hinge was slightly ahead of the upper hinge. Inside the flange around the door was a rubber weather seal which clamped against the door frame. In the new door, to correct the interference condition, the lower hinge is moved back far enough so that, when the door is opened, its outside leading flange and the fender cap swing more rapidly in toward the car center. The revised construction and hinging also necessitated suitable modification in the sealing of the front doors.

Radiator grilles of virtually all 1942 models are now of sheet steel stampings instead of zinc die castings to conserve zinc for defense purposes. So skillfully have they been designed and fabricated that, when chrome-plated as they appeared on the 1942 models at the time of their announcement, it takes an expert to determine the difference in material and fabrication.

The new Ford parking light housings are a striking example of ingenuity in designing for alternate materials. These housings formerly contained six critical materials – they were die cast in zinc alloy containing aluminum and magnesium and plated with successive layers of copper, nickel, and chromium. The alternate design contains no critical materials, being made of molded glass silver-plated on the inside. In addition, the alternate parking light has equal utility and is considered to be better looking than its predecessor.

Some of the most extensive changes in Chevrolets have been made in the front-end sheet metal. The hood nose is wider and blunter and extends down to the headlamp centerline. The former hood side panel has been eliminated, and the hood brought back to the door. The hood hinge mechanism has been revised and now raises automatically higher than before and, with a prop, may be held in almost vertical position because of the elimination of former brace rods extending from the top of the radiator to the dash. A splash shield has been added between the front bumper and the grille and fenders on all Chevrolets; it is an extension of the baffle plate which formerly served only to close the opening between the bottom of the grille and the bottom of the radiator body.

In connection with the new front-located air intakes of the Cadillac and Buick heating and ventilating systems, cowl ventilators have been eliminated on all models. In addition, Cadillac has eliminated the hood louvers.

The front fenders on all Chrysler products are flared out more, and are flatter on top; all have alligator-type instead of side-opening hoods this year.

"Double-duty" bumpers feature the front-end appearance of the 1942 Oldsmobiles. Instead of a single-bar bumper with the conventional grille guard, the front bumper consists of two separate horizontal bars connected by vertical bumper guards. Both horizontal bumper bars are joined to the car frame by steel bars, as shown in an accompanying illustration. In addition the upper bumper bar is supported by two horizontal channel braces which are bolted to the frame. Cooling air enters through the radiator grille located between the two bumper bars.

Bodies

The trend toward wider bodies continues for 1942, although there are fewer really new designs. Concealed running boards now have replaced the exposed type on all but a few models. They are illuminated at night when the doors are opened on one line of cars. Two-door, close-coupled sedan-coupes with straight streamlined backs continue to gain in popularity. Electric motors replace vacuum cylinders on many convertible designs and free-wheeling inside handles are used more widely on the rear doors of four-door sedans.

One of 1942's widest and lowest bodies is the Packard Clipper with a door-to-door front seat width on four-door sedans of 62½ in. as compared with 55 in. for 1941 Packard bodies. The overall height unloaded is 65 9/16 in. for the 6 and 8, and 66 in. for the 160 and 180, more than 2 in. less than corresponding 1941 Packard models. As in 1941, running boards are optional on Packard Senior models. Cadillac 75's are furnished only with running boards.

All bodies of the 1942 Hudson line are provided with concealed running boards. On the Super 6 and Commodore models, a courtesy light, built into the door frame at the bottom, is switched on automatically when any door is opened to illuminate the running board at night. This year a convertible sedan has been added to the Hudson 6 line.

A pair of courtesy lights is now located on the backs of front seats of 1942 Studebaker Deluxstyle and Skyway four-door sedans. The lights are turned on when either rear door is opened and switched off when it is closed. Although running boards are listed as standard equipment on Willys-Overland cars, on both DeLuxe and Speedway models, their absence is optional. To protect Willys Americar bodies further against noise and vibration, 29 lb of sound-deadening vibration-absorbing compound is sprayed on the front floor panels, cowl tops and sides, transmission covers, toe boards, radiator air deflectors, rear wheelhouse panels, and fenders of the 1942 models. Front doors of Americars are fitted with self-holding door check straps to hold the doors open.

Rear quarter windows have been added to the Oldsmobile, Buick, Pontiac, Cadillac and Chevrolet convertibles. Each window is a quarter circle of glass which may be rotated into the body side for ventilation, whether the top is up or down. A new top operating mechanism powered by electric motors in both sides of the body replaces the former vacuum cylinders on these General Motors products. Two buttons on the instrument board control this movement. In case of failure of the electrical system, the tops can be raised or lowered by hand without harm to the system.

Pontiac has added the streamlined back sedan coupe available on its 122-in. wheelbase lines to its 119-in. wheelbase 1942 Torpedo line.

The cowl ventilator of all Nash models is now held permanently in the open position. This change was made mostly to insure a supply of fresh air for the heating and ventilating system.

New bodies are announced by Cadillac for the 60 and 62 models and by Buick for the 50 and 70 models in conjunction with their increases in wheelbase.

The Cadillac bodies are of the two-door type and set about 1½ in. lower on their frames. The new Buick bodies are wider than their 1941 predecessors, measuring 62 and 52 in. in width at the front and rear seats, giving seats that are wider by 1¾ and 4 in. respectively. Front fenders of these bodies are carried back through the doors until they meet the rear fenders.

Chevrolet's new Aerosedan is 2¼ in. lower in overall height than the Chevrolet Town Sedan and 1½ in. lower at the front and rear seats. Its rear seat is 3¾ in. farther forward. It is a close-coupled two-door sedan with a rear contour that sweeps back in a single arc from the roof to the rear bumper.

Plymouth announces new body types – a club coupe in the Deluxe line and a four-door town sedan with no rear quarter windows in the Special Deluxe line. The Special Deluxe club coupe has full-width rear seats instead of drop seats.

The luggage compartment lid lock, license light, and deck handle on all Chrysler products has been redesigned to make the lock more accessible. Locks appear on both front doors of the Plymouths, Dodges, and DeSotos for the first time this year. Concealed running boards only are available on all cars made by Chrysler, and front seats are adjustable over a 5-in. range.

The flush-type pushbuttons, which replaced the door handles on some 1941 Lincoln models, have now been extended to cover the entire line. Ford has dropped the coupe with auxiliary seats.

To provide greater safety for children riding in the back seat free-wheeling safety lock door handles are employed more extensively on rear doors. When the button on the sill is in the locked position, the back doors of a four-door sedan are locked inside even though the front doors are open.

Conclusion

Even though it is planned to "freeze" the designs of the 1942 passenger cars for the next few years in the interests of national defense, the author wishes to emphasize that this paper describes the 1942 models only as they were on the day of their announcement. While the models *will* be frozen in that no design changes will be made solely to improve or restyle the cars, there is nothing so certain as a steady redesign of parts to accommodate alternate materials in place of defense-needed critical materials. The coming elimination of all bright work is just one of the future changes that will make the late 1942 models considerably different than they were on the day of their announcement.

MECHANICAL SUPERCHARGING of DIESEL ENGINES

by H. L. KNUDSEN
Chief Engineer, Cummins Engine Co.

THE most serious problem arising from higher degrees of supercharging is the increase in exhaust temperatures and the amount of additional heat to be handled. This conclusion is expressed following a theoretical exploration into the possibilities of supercharging and the ultimate limit to which it is possible to go. Efficiencies which may be expected with increasing degrees of supercharging, with and without compressor intercooling, are predicted.

Some of the present-day superchargers are discussed, including the Roots, vane, centrifugal, and exhaust turbo-type blowers, and the advantages and disadvantages of each are given.

Before closing his paper, Mr. Knudsen emphasizes the need for more compact and efficient accessories. No attempt has been made, he says, to improve the specific capacity of these units, with the result that, "as we go down in engine size and up in horsepower, the auxiliaries become larger, heavier and bulkier than ever—so much so that, at the present time, we are almost to the point where the engine proper is completely hidden behind an assorted number of clumsy and unwieldy accessories."

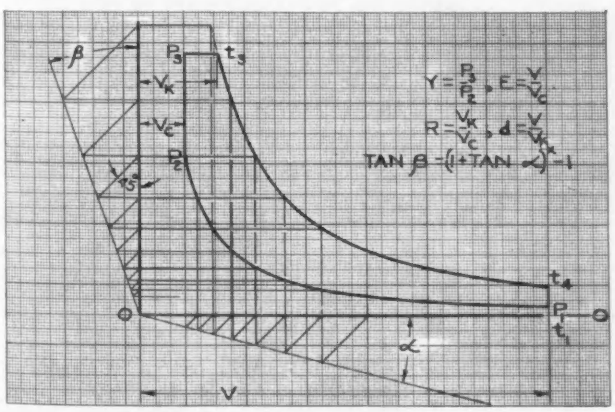
IN discussing supercharging of diesel engines, the first question one might ask is, why supercharging at all? And the answer is, our eternal quest for greater specific output from a given cylinder displacement, the object being less weight and space requirement for our powerplants.

Of course, there are other means of increasing the specific output of an engine besides supercharging. The first and most obvious is to improve the combustion cycle to the greatest possible extent; that is, with natural aspiration, produce the highest possible mean effective pressure. After we have gone as far as we can in this direction, then there still remains the possibility of increasing the speed of the engine; however, in modern diesel-engine design, we have already gone as far as it seems practical at this time in utilizing our combustion cycle as well as speed and, therefore, if we want to progress further, it becomes necessary to go to supercharging; that is, increasing the air capacity of our engine by artificial means.

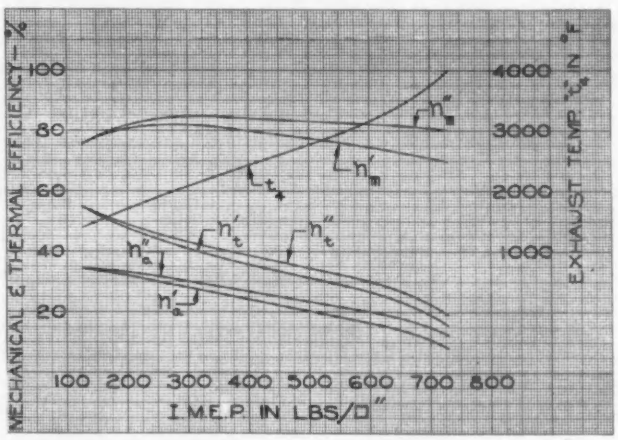
When you come to a study of the problem, it is surprising to see how many methods there are of supercharging an engine cylinder. One method, which we shall mention first because it is the most efficient, is to utilize the kinetic energy of the exhaust gases to drive a turboblower. A turbine wheel and a blower impeller are mounted on the same shaft, the exhaust gas furnishing the power to the turbine and the blower in turn delivering the air to the engine manifold. This method of supercharging, however, so far has been applied mainly to larger engines and is today being used by a number of manufacturers in this country and abroad.

We first began to hear about supercharging during the

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 5, 1941.]



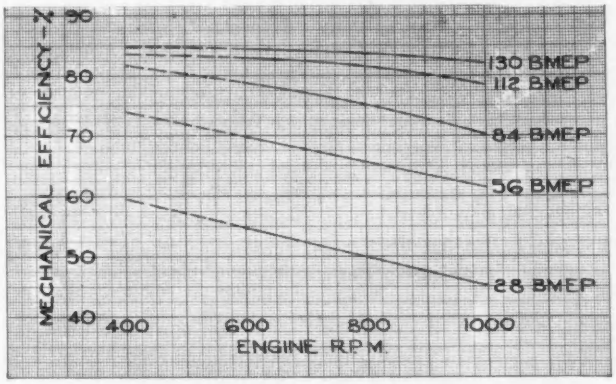
■ *Fig. 1 – Theoretical indicator diagram



■ Fig. 2 – Efficiency and temperature curves

first World War, and the French professor, Rateau, I believe, was one of the first to suggest the turboblower system, it being further developed later on by the Swiss engineer, Alfred J. Buchi, whose system is now widely used.

Another method of supercharging is to use a mechanically driven blower, either directly hooked to the engine or driven by some independent power source. These mechanically driven superchargers are again of different designs; the centrifugal type of blower or compressor is being used, and the Roots type of blower is quite a favorite at this time. In England, the so-called Centric blower or vane type has been used. This type of blower is similar to the Bendix vacuum pump, now used quite extensively for braking purposes on trucks; that is, the rotor is mounted eccentrically in a housing and is fitted with sliding blades, which, in the Bendix type, are driven against the rotor housing by centrifugal force. However, in the Centric type, the motion of the blades is mechanically controlled, and, therefore, it would be better suited to higher pressures and speeds. The vane type, with automatically controlled blades, I would not consider very satisfactory for higher pressures – at least, our own experience with a blower of that type was anything but satisfactory.



■ Fig. 3 – Mechanical efficiency of Cummins supercharged diesel engine – 2309 cu in. displacement

There is another method of supercharging, which, however, is applicable only to very large engines using crossheads. In this type of engine, the under side of the piston (it should be remarked that this applies only to four-cycle engines) is used to furnish the surplus air required for supercharging. It will be understood that, in this type of engine, there are two air strokes of the piston to one power stroke, and, therefore, about 90% increase in air over a naturally aspirated engine could be furnished; that is, proper allowance has to be made for the area of the piston rod.

Another method of supercharging is to use the kinetic energy of the air column to produce a sort of ram effect in the cylinder. This latter method, however, I would consider limited to stationary powerplants where the speed is constant and long pipes can easily be arranged for.

Some work has also been done in using the exhaust column to produce a vacuum in the cylinder during the exhaust cycle and thereby causing a greater column of air to enter the cylinder. This system is mainly applicable to two-cycle engines, however, and some engines indeed have been run on this system without any other means of scavenging. This is the Kadenacy system.

Another system being used, which is applicable to both two-cycle and four-cycle engines, is the so-called "after-charging" system. The Busch-Sulzer Co., of St. Louis, Mo., has used this system in its two-cycle engine, increasing the power output of the engine about 20%. The system requires an additional compressor, for the two-cycle engine, of higher pressure than the scavenging pump. After the scavenging ports and the exhaust ports are closed, another set of ports opens to allow the entrance into the cylinder of high pressure air; that is, air of moderate pressures but, of course, higher than that prevailing in the cylinder. This "after charging" system has been used in four-cycle engines also by the Burmeister and Wain Co., of Copenhagen but, according to data at my disposal, the increase in horsepower output attempted is not over 20%.

Another method that has been tried but which is applicable only to engines using air injection is to increase the oxygen content in a cylinder by the use of an excessive amount of high-pressure injection air. Of course, we do not need to be concerned about this latter system, since the air-injection engine is rapidly becoming obsolete.

We, the Cummins Engine Co., have elected to use the Roots-type blower in our engines for supercharging, and we have been able to increase our output as much as 60% in that manner. This sounds very interesting, and, naturally, one cannot help becoming curious about how far it is possible and practical to go with supercharging. Normally, in our naturally aspirated engines, the mean effective pressures in the smaller engines run around 82 psi. With supercharging, we have gone up to 131 psi, which is a 60% increase. As a matter of fact, on the test block we have produced mean effective pressures of 160 psi, and so one wonders how far we dare go, because it is obviously only a matter of increasing the air supply and, by appropriate changes in the compression ratio and rate of fuel injection, to control the pressures.

In order to explore the possibilities, let us do a little theoretical investigation of this problem. Let us assume an engine operating on the mixed Otto and diesel cycle, which is the most common mode of operation today, and let

n_t = Theoretical Thermal Efficiency

n_m = Mechanical Efficiency

n_a = Actual, or Brake Thermal Efficiency

L_t = Available Heat Energy

L = Heat Energy Supplied

E = Compression Ratio $= \dfrac{V}{V_c}$

R = Cut-off Ratio $= \dfrac{V_k}{V_c}$

Y = Pressure Ratio $= \dfrac{P_3}{P_2}$

x = The Specific Heat Ratio $= \dfrac{C_p}{C_v}$

Now, the thermal efficiency of a heat engine generally is:

$$n_t = \frac{L_t}{L}$$

and, by appropriate substitution for the mixed cycle, this reduces to:

$$n_t = 1 - \frac{1}{E^{x-1}}\left[\frac{YR^x - 1}{(Y-1) + Yx(R-1)}\right]^1$$

[1] For the derivation of this formula, see "The Internal Combustion Engine," Vol. I, 2nd Edition, by D. R. Pye, Oxford University Press.

If we now construct a series of theoretical diagrams as per Fig. 1, all based on the same cylinder displacement, and choose the values x, P_3 and P_2, we can determine the values for V, V_c and V_k for a given mean effective pressure.

By looking at the diagram in Fig. 1, it is quite apparent that we reach the ultimate in supercharging when $V_k = V$; that is, when our expansion ratio $d = \frac{V}{V_k} = 1$ or, in other words, when we have no expansion of the gases but proceed with combustion to the end of the stroke. It is also obvious by a mere examination of the diagram that the smallest practical P_1 and P_2 and the highest practical P_3 gives us the greatest possible degree of supercharging.

P_1 is determined by the degree of supercharging desired; the compression pressure P_2 and the temperature t_1 corresponding to P_1 and, it is, of course, also dependent upon the polytropic exponent x of the compression stroke.

If, for the naturally aspirated engine, we choose an initial pressure $P_1 = 12.5$ psi; $P_2 = 512.5$ psi; $P_3 = 862.5$ psi; and a polytropic exponent $x = 1.31$ and, on that basis, determine V_c and V_k for an indicated mean effective pressure of 125 psi and substitute in our formula for n_t we get a theoretical thermal efficiency of about 55%.

Now, if we increase our mean effective pressure to 250 psi, we have to double our air supply, and so our compression space V_c is doubled, since we want to keep our compression pressure P_2 constant. Again, P_1 must be increased so that the weight of the air of the volume V of the supercharged engine is twice that of the weight of the volume V of the naturally aspirated engine. This again calls for a suitable change of the compression ratio E, and these values being determined, P_3 remaining constant, we can again establish a new V_k and calculate our theoretical thermal efficiency n_t. Proceeding at suitable steps in this manner until our expansion ratio $d = 1$, we obtain curves n'_t and n''_t in Fig. 2.

The curve n'_t is based on the assumption of adiabatic compression of the compressor, and the curve n''_t is based on isothermic compression of the compressor, which at once shows the benefit of an intercooler between compressor and engine, and particularly as the degree of supercharging increases. It may be pointed out here that the increase in the theoretical thermal efficiency by using an intercooler is due to the higher compression ratio E that results with the use of cool air when P_2 remains constant.

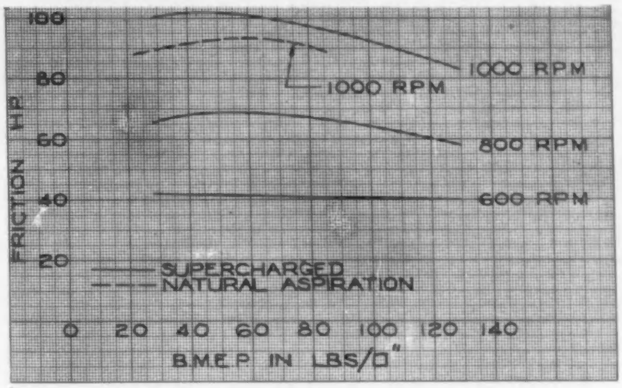
Studying the expression for n_t (the theoretical thermal efficiency) namely, we find that, if the bracketed term is eliminated, the remainder is the thermal efficiency for the Otto cycle and that the value of this expression for a constant exponent x is determined entirely by the compression ratio E.

While a direct examination of the bracketed term, which is a modifying factor for the mixed cycle, does not permit of an easy evaluation, a series of calculations shows that it is greater than 1 and increases with increasing cut-off ratio R. However, within the scope of our investigation, the compression ratio E is the dominating factor in determining our theoretical efficiency n_t.

The curves n'_a and n''_a represent the actual or brake thermal efficiency of our hypothetical engine, and is obtained by multiplying n_t by a card factor and the mechanical efficiency.

Now we are obviously confronted with a difficulty here because there are no available data of mechanical efficiencies for such a degree of supercharging as we are discussing. To determine, if possible, from existing engines the relationship of frictional losses to increasing mean effective pressures, a test was made to determine the mechanical efficiencies over the entire speed and load range of an engine of 2309 cu in. displacement. The results of these tests are plotted in Fig. 3.

The frictional horsepower can, of course, be found from these curves and, plotted as shown in Fig. 4, reveals that the maximum frictional horsepower occurs at a brake mean effective pressure of about 60 psi and that at the maximum speed, 1000 rpm, which equals a piston speed of 1667 fpm,



■ Fig. 4 – Friction horsepower for Cummins engine – supercharged and natural aspiration – 2309 cu in. displacement

there is a rather decided drop in the frictional horsepower of the supercharged engine as the mean effective pressure increases. At 800 rpm, this decrease of frictional horsepower with increase in mean effective pressure is still much in evidence while, at 600 rpm, the frictional horsepower is practically constant over the entire load range.

The naturally aspirated engine which was tested only at 1000 rpm indicates the same trend, although not in the same degree. It should be pointed out also that this engine was rather green and that, therefore, the frictional horsepower appears a little high as compared with the supercharged engine.

To determine if engines of other sizes exhibited the same tendency, a test was made of an engine of 672 cu in. displacement at a piston speed of 1800 fpm, and the curve showed exactly the same trend for both the supercharged and naturally aspirated engine.

The meaning of this somewhat unexpected behavior is not quite clear, but the author believes that the explanation may be found in the temperature changes of pistons and cylinder walls, affecting the viscosity of the lubricating oil on these surfaces.

The degree of supercharging of both test engines is equal to about 175 psi indicated mean effective pressure and, while the curves would indicate a continued drop in frictional horsepower with increase in mean effective pressure, it would seem entirely too optimistic to assume this to be true. One would be inclined, rather, to assume an increase due to the later cut-off with increased mean effective pressure. For our investigation, however, and since the compressor horsepower becomes more and more dominant as the degree of supercharging increases, we will assume:

1. That the frictional horsepower of the engine proper

remains constant and equal to that of the unsupercharged engine at full load and full speed.

2. That the compressor has an efficiency of 80% (too optimistic for available superchargers at present, but necessary if we are to be successful with higher degrees of supercharging).

3. That the compressor returns to the engine cylinder power equal to a mean effective pressure of P_1 less the atmospheric pressure.

Based on the foregoing assumptions, we arrive at the mechanical efficiency of n'_m for a compressor with adiabatic compression and n''_m for isothermal compression as shown in Fig. 2.

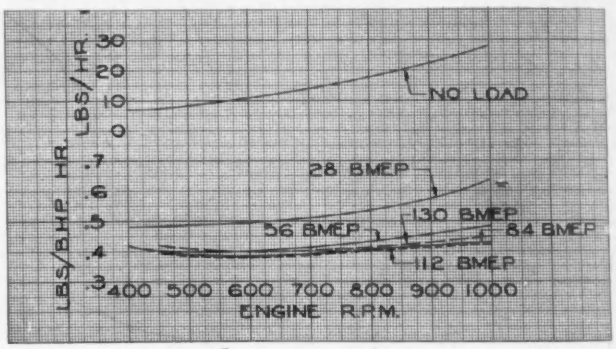
In arriving at the values for n'_a the brake thermal efficiency for adiabatic compressor compression and n''_a for isothermal compression, a card factor was chosen which gives a value of n_a for the unsupercharged engine equal to those actually obtained in laboratory tests. (In our case, the card factor is about 82%.)

Examining now the efficiency curves n_a in Fig. 2 which have been plotted against the theoretical indicated mean effective pressure, and in which is included also the returned mean effective pressure from the compressor, we note that, for the values of pressures chosen, the maximum possible theoretical indicated mean effective pressure is about 730 psi and that, as far as efficiency is concerned, where high specific output is of first importance, we can well afford to increase the degree of supercharging now in use to a considerable extent.

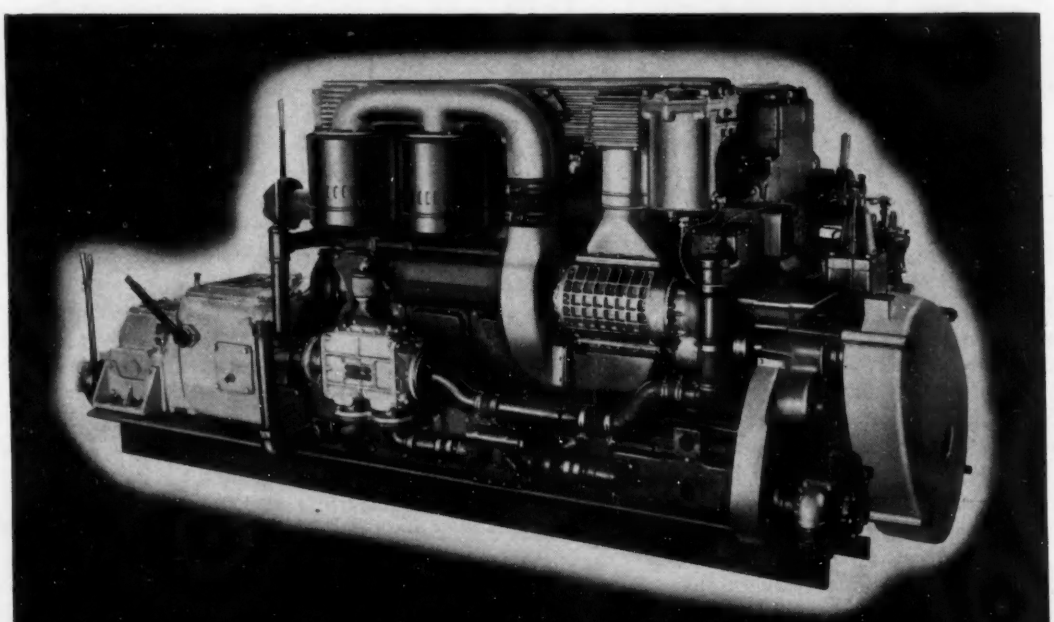
If, however, we take a look at the temperature t_4 (Footnote 2) in Fig. 2, the proposition does not look so good. The temperature t_4 was arrived at by assuming a combustion temperature t_3 of 4000 F and, using a polytropic exponent for the expansion curve of 1.4, which gives values approximating temperatures actually existing in a naturally aspirated engine. Since the valves in a high-speed supercharged engine with an indicated mean effective pressure of 175 psi are already loaded as high thermally as it seems advisable to go, and the same thing may be said of pistons and piston rings, a further increase would certainly call for special cooling of both pistons and valves. If we increase our supercharging up to, say 500 psi theoretical mean effective pressure, Fig. 2 indicates that we would reach a temperature equal to the melting temperature of steel. It is hardly necessary to point out that such temperatures will present a real problem in cooling as well as in lubrication. It also is to be kept in mind that it is not only higher tempeatures we have to handle but a considerable increase in heat units as well.

[2] The curve for adiabatic compressor compression only has been drawn.

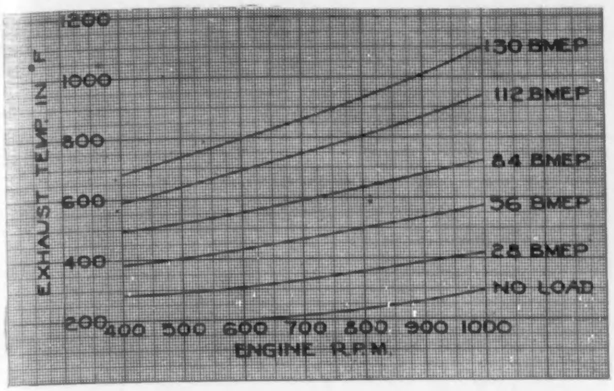
As regards the increased bearing pressures, I believe this would be the least serious of our problems although it would require a high-grade bearing, but it must be remembered that bearing pressures are not dependent upon the combustion stroke alone. For every combustion stroke, there are four inertia strokes responsible for a large share of our bearing load. We are, therefore, not increasing our bearing loads in the same ratio as our power is increased. As a matter of fact, for normal supercharging such as used



■ Fig. 5 – Fuel consumption of Cummins supercharged diesel engine – 2309 cu in. displacement



■ Fig. 6 – Cummins supercharged diesel engine equipped with a belt-driven Roots supercharger – 2309 cu in. displacement



■ **Fig. 7 – Exhaust temperatures of Cummins supercharged diesel engine – 2309 cu in. displacement**

today, the increases in bearing pressures are on the order of 5 to 15%, with a power increase of about 60%.[3]

The predicted brake thermal efficiency n_a from Fig. 2 is well in agreement with actual performances of engines operating with 50 to 60% supercharging.

It must be borne in mind, however, that the mechanical efficiency beyond a theoretical mean effective pressure of 200 psi is somewhat questionable. So, while for the higher mean effective pressures no high degree of accuracy can be claimed, the curves are nevertheless, it seems to me, interesting and of value in showing, in some degree, what we may expect.

In Fig. 5 are shown fuel-consumption curves over the entire speed and load range of a Cummins supercharged diesel engine equipped with a belt-driven Roots supercharger. The curves are self-explanatory and show that it is possible to equal and, in some respects, even better the fuel consumption of a naturally aspirated engine with a superchaged engine using a mechanically driven blower. A view of this engine is shown in Fig. 6.

Exhaust temperatures of the engine are shown in Fig. 7. These temperatures are no higher than found in a naturally aspirated engine when based on the percentage of load. Now this is contrary to our findings in Fig. 2, but easily explained, since in a supercharged engine a certain amount of air is allowed for scavenging, which cools the exhaust gases as well as the valves. But during the escape of the exhaust gases, the valves are actually subjected to a higher temperature than in a naturally aspirated engine.

That we eventually will go higher in our degree of supercharging, I do not doubt, but it goes without saying that an increase in the degree of supercharging will have to be preceded by a great deal of laboratory work.

In the further development of the supercharged engine, the blower itself should come in for a great deal of study. The present-day commercial blower of the Roots type is not very efficient, that is, from the standpoint of power consumed to drive it. Of course, in a four-cycle engine, we have a distinct advantage in that the manifold pressure produced by the supercharger is not a total loss. A considerable amount of the power expended in compressing the air is returned in the form of expansive power on the piston during the intake stroke. Nevertheless, a compressor which has only an efficiency of 59% at 8 psi pressure needs to be improved upon, and unfortunately, as the pressure goes up (which would be necessary if a higher degree of supercharging is desired), the efficiency of the blower reduces further – at 10 psi, for instance, it is only 56½%.

The reason for the poor efficiency of the blower is partly in the rather excessive clearances required in order to make the blower perform mechanically. These clearances could be improved upon very materially if we were willing to increase our weight and substitute cast iron or, better still, an Invar type of material, for instance, for both the housing and the rotor instead of aluminum which is now generally used. Perhaps this statement might be questioned, reasoning that, since both rotor and housing are subject to the same temperature, there ought to be no difference in clearances, no matter what the coefficient of expansion of the materials used. That isn't quite true, however. The temperature of the blower is by no means uniform. As a matter of fact, the temperature is much higher on the discharge side of the blower than on the intake side, and this difference in temperature tends to distort the blower quite considerably and, for this reason, it has been necessary to use clearances that result in a rather high slip speed with consequent heating and volumetric inefficiency.

Aside from the bad slip effect of the Roots blower, it might be stated also that the cycle on which it operates is the least efficient that we know. In contrast to piston compressors, vane compressors, and even centrifugal compressors, which operate with adiabatic, or at least polytropic compression, the Roots blower, in consequence of its peculiar compressive action, operates on a constant-pressure cycle. Fortunately, at the pressures we operate, the difference between constant pressure and adiabatic compression is not great but, at higher pressures, it becomes a very serious factor and one to be reckoned with.

As may be understood from the fact that the Roots blower is rather widely used for supercharging purposes, it naturally must have a number of features on the credit side too. First and foremost in this category must be mentioned its simple mechanical construction and, when it is properly designed and manufactured, its great reliability in operation. Secondly, no lubrication of the wings or impellers is required, removing a problem of oil control and operating expense that is quite important. Thirdly, its delivery characteristics are such as properly to proportion the air need of the engine to varying speeds when directly connected and, while it is affected by back pressure or resistance, is not overly sensitive to it.

Again, when all things are considered, its overall efficiency is not inferior to other practical forms of compressors when properly made and operating at low pressures such as required for supercharging purposes at this time.

The vane type of compressor or blower has some desirable features about it. It operates with polytropic compression but, in common with the Roots blower, requires very close rotor and blade clearances to keep the slip speed, and hence the efficiency within bounds. However, the author considers the necessity of lubricating the vanes enough of a detriment to prefer the Roots type at this time.

The centrifugal type of compressor, due to the extreme speeds required, does not seem to offer quite the degree of reliability one would want to have in a piece of machinery that must be maintained by operators who are often anything but skilled in the maintenance and operation of machinery of that nature. It also is more sensitive to changes in back pressures and, therefore, more apt to cause

[3] For a detailed study of this problem, see SAE Transactions, May, 1938, pp. 215-224: "Diesel Supercharging – Its Effect on Design and Performance," by Russell Pyles.

difficulties due to variations in installations operating with different kinds and types of mufflers, lengths and sizes of exhaust pipes, and so on. Generally, this type of blower has about the same efficiency as a Roots blower, but may have a very bad efficiency if not operated at the correct speed and delivery.

The good efficiency of the exhaust turboblower is not a case of superior efficiency of the blower and turbine mechanism as such, but rather because the power to drive it is derived from a source that is otherwise generally a complete loss. This does not mean that the power for driving the exhaust turboblower is obtained at no expense to the engine. The exhaust back pressure must be raised to operate the turbine, which, of course, constitutes a power loss and particularly a temperature rise. Some rough estimates indicate that the power loss sustained in driving the exhaust turboblower is about one-half that required for driving the Roots blower. The net power loss, however, is small due to power returned by the blower. This is also borne out by the fact that the fuel consumption with that type of blower generally shows an improvement in the efficiency compared with a naturally aspirated engine, while the Roots-equipped engine usually shows the same or a slightly higher fuel consumption. When, in spite of this, the Roots blower has been used so extensively, particularly in small engines, it is rather because the mechanics of the exhaust turboblower, at least up to this time, have not been all one could ask for. The bearings of the high-speed turbine and blower shaft have been a prolific source of grief, although the latest information I have on the subject is that the problem has been pretty well taken care of by the use of plain bearings instead of ball bearings.

It is said, also, that the exhaust turboblower cannot be operated with an exhaust temperature over 1100 F which, in itself, makes it unsuitable for a high-speed, high-output small diesel.

In discussing this subject of high-speed, high-output engines, there is a problem that so far has been given very little attention, but which must be taken into consideration if we are to be altogether successful in this high-speed, high-output development.

The problem is that of auxiliaries generally. No attempt has been made to improve the specific capacity of these units, with the result that, as we go down in engine size and up in horsepower, the auxiliaries become larger, heavier, and bulkier than ever. So much so that, at the present time, we are almost to the point where the engine proper is completely hidden behind an assorted number of clumsy and unwieldy accessories. When we stop to think about it, it does seem ridiculous that a small compressor, for instance, used for braking purposes in trucks, having a bore of about 2 in. and a stroke of 1½ in. and a moderate pressure of about 90 psi, cannot be operated even at the same speed as the engine, in which we are operating with compression pressures of 500 psi, combustion pressures of around 850 psi, and temperatures in the neighborhood of 4000 F. Likewise, generators for lighting and starting purposes are not recommended at speeds over 2000 rpm or, at the most, 2500 rpm, and a great many operators prefer to run them even slower because of difficulties encountered at the higher speeds. Air cleaners are positively monstrous in size and operate with entirely too much depression, robbing us of precious oxygen. Lubricating oil filters are also too bulky or, if smaller sizes are attempted, they do not do a proper job of filtering or, at least, only for an impractical short time. Manufacturers of gear water pumps also have looked askance at higher speeds, with the result that these units become oversize.

So, if we are to continue to make progress in weight and space reduction of our powerplant, it is high time that some earnest attention be given to this problem by accessory manufacturers, and one of the first problems to be considered should be that of improving the pressure range and efficiency of the supercharger.

DISCUSSION

Centrifugal Supercharger Chosen for Aircraft Engines

– Kenneth Campbell
Wright Aeronautical Corp.

THE author's choice of the positive-displacement blower over the centrifugal type from considerations of reliability, sensitivity, and efficiency appears at variance with our experience in the aircraft-engine manufacturing field. The positive displacement or Roots-type blower was the first type used for aircraft-engine supercharging in this country, being applied to one of the older Whirlwind models years ago. Difficulty was experienced from wear due to uneven expansion or from the wide clearances which this expansion necessitated, and the centrifugal compressor was chosen in the search for a more reliable type. These centrifugal units operate up to tip speeds of 1250 fps for high-altitude work, a typical design incorporating an 11-in. impeller rotating at 26,000 rpm maximum. This gives a pressure ratio of over 2.5:1 with a clearance between the stationary shroud and the open side of the impeller from 0.035 to 0.045 in. Practically no mechanical difficulties are ever encountered with these blowers.

With regard to sensitivity, when a centrifugal blower is operated separately from the engine that it feeds, as in the laboratory when the blower is driven by a variable-speed electric motor, then, indeed, the operating point may shift widely along the pressure boost and efficiency characteristic curves, and varying degrees of boost at the same speed are encountered. However, when the impeller is geared to the crankshaft, neither speed nor throttling (at the inlet) nor density change with altitude, shift the operating point on the Q/N characteristic curves except in small degree. Exhaust back-pressure variation, by slightly varying the volume and resulting power can cause a small shift along the characteristic curve; but here again, only if the supercharger is proportioned wrongly for the engine and the operating point is located away from the non-critical or relatively flat region of the efficiency curve should this shift cause any appreciable effect on manifold pressure developed. It is true that, when a variable supercharger gear ratio is provided with a given design and engine speed, this shift becomes greater and must be considered in the design of the supercharger.

Published adiabatic efficiencies of centrifugal-type superchargers for aircraft engines are approximately 70%.

Higher Efficiencies Expected for Roots-Type Blower

– John L. Ryde
Chief Engineer, McCulloch Engineering Corp.

MR. KNUDSEN'S paper on the application of superchargers to the diesel engine is of particular interest to us as manufacturers of supercharging equipment, and we take this opportunity to commend his analysis of the problem.

The author's choice of the Roots-type blower parallels our own analysis very closely. We have manufactured in production the centrifugal type and have made a detailed study of a number of the various types of positive-displacement blowers now available commercially or on paper. Representative models were studied in the laboratory to determine the characteristics of each from not only the performance standpoint but with regard to production problems,

original cost, and the question of servicing the units after they are in the field.

As a result of this analysis, the Roots-type blower was selected for a number of reasons. As pointed out by Mr. Knudsen, it is the simplest unit mechanically even when comparing it with the centrifugal blower. The centrifugal blower is perhaps simpler basically, but the necessary gearing for the high rotative speeds offsets this advantage.

Likewise, from a manufacturing standpoint and consequently the original cost, the Roots type has the advantage in that the precision required is well within the limits of up-to-date high-production machine-tool equipment. This is of particular importance in the ultimate use of superchargers on commercial engines where the cost factor looms large. Generally speaking, the cost of superchargers in the past has been such that it was almost as economical to add two more cylinders to the engine and a supercharger was only prescribed when weight and space were at a premium.

Thirdly, this type of compressor presents a very fertile field for development. For the past two years we have been manufacturing a high-speed Roots type supercharger in various sizes and have had the opportunity to study the application on a large number of different engines covering both the addition of the supercharger to present engines as well as installations on units particularly designed for supercharging. As a result of this experience, our present research activities are directed towards higher rotative speeds and of particular importance, higher efficiencies. The efficiency of the supercharger has a very marked effect on the performance of the engine and this stands out very clearly on some of the present marine engines where the maximum output is of greatest importance. On this type of engine a small variation in the efficiency of the supercharger results in a markedly greater change in the engine output.

To show this point in detail, it should be noted that the prime purpose of the blower is to deliver air of increased density to the cylinder. The supercharging compressor raises the temperature as well as the pressure and, even with intercooling, there results a definite ceiling or maximum boost pressure beyond which it is uneconomical to supercharge unless the intercooling is increased or the blower efficiency is improved. This point can be checked by carrying through Mr. Knudsen's analysis with various supercharger efficiencies and discharge temperature versus pressure characteristics.

When operating at or near the optimum supercharge or boost pressure level, an improvement in blower efficiency results in a marked change in the engine performance as to maximum output and thermal efficiency. In one particular case an increase in blower efficiency of 7% resulted in an equal increase in engine output.

As the author shows, one of the largest discredits to the Roots blower is the loss through slip or air leaking back through the clearances, resulting in low volumetric efficiency. The volumetric efficiency is a linear factor in the overall efficiency of the unit. In order to reduce the slip, we have developed units with sealed end clearances showing a considerable improvement in efficiency. Similar to the gear pump, the end or axial clearances of the rotors have a great influence on the slip and by closing these to substantially zero, the efficiency is increased materially.

Likewise a further gain was made by coating the rotors with a special plastic compound to reduce the rotor to rotor clearance without incurring trouble due to scoring or interference. This permitted the retention of satisfactory manufacturing tolerances necessary for production without a sacrifice of performance.

Need for Improved Lower-Cost Blowers

– C. G. A. Rosen

Assistant Chief Engineer, Caterpillar Tractor Co.

THE author's paper represents a vast amount of work on supercharging as applied to mobile-type diesel equipment.

It would be of interest to know what method was used in obtaining mechanical efficiency as shown in Fig. 3. The data shown in Fig. 4 require further amplification.

The use of "card factor" is not theoretically accurate and can therefore not be considered entirely reliable.

Mr. Knudsen's comments on the size and ratings of accessories are very good. Active interest should be manifested in coordinating these elements to the engine itself more effectively and efficiently.

Mr. Knudsen has shown the advantages to the diesel in burning more air – there is still, however, much reason to expect improvements in the blower.

The thermal degeneration of an engine is always a vital practical factor – it is of marked influence in maintenance costs. The temperature of air charged to the cylinders from the blower bears an important relation to power output, peak pressures, and exhaust temperature – all of which determine engine life.

An undesirable factor in blower design for variable-load mobile equipment is the necessity for designing the blower to satisfy conditions at the peak of the torque curve rather than at full-load full-speed.

To realize the theoretical advantages of supercharging, the blower must be better and cheaper.

How Tractor and Implement Engineers Can Cooperate

WHAT the tractor engineer and the implement engineer can do to help each other in a constructive way can be summarized by the following suggestions:

THE TRACTOR ENGINEER CAN:

1. Acquaint the implement engineer with the fundamentals of tractor design, explaining what parts can be compromised to help give clearances for implements and what parts cannot.
2. Go into detail on the tractor attaching points for implements, indicating where it is safe and desirable to hitch and where not.
3. Give the implement designer a course in automotive practice so that an implement may be evolved of lighter weight and smaller component parts but with adequate strength and stiffness to fill the bill.
4. Go into the distribution of weight of integral equipment on the tractor to be sure of the stability and performance are not out of balance; also give the permissible weight on both front and rear tires.
5. Acquaint the implement engineer with the limitations of the power take-off and power-lift.

THE IMPLEMENT ENGINEER CAN:

1. Teach the tractor engineer the fundamentals of agriculture as applied to the use and function of implements in producing and harvesting crops.
2. Give the tractor engineer the information about the various crops such as width of rows, size of plants, clear space between rows, and so on ad infinitum.
3. Take the tractor engineer out into the field and demonstrate the various implements to him.
4. Show the engineer just what the functions of the implements are and what results are to be expected.
5. Teach the tractor engineer how to operate the implements himself, and make sure that he does a good job.
6. Explain fully why clearance, visibility, and accurate steer are necessary in row crop cultivation.
7. Show the tractor engineer the great advantage of holding as closely as possible to the various standards for power take-off shafts, belt pulleys and drawbars, and urge him to decrease, rather than increase, tolerances for these standards.

Excerpts from the paper: "The Relation Between Tractor and Implement Engineering," by Theo Brown, Deere & Co., presented at the National Tractor Meeting of the Society, Milwaukee, Wis., Sept. 25, 1941.

The ALLISON AIRCRAFT-ENGINE D

AS far back as 1931 when aircraft engines in general were developing 0.25 to 0.35 hp per cu in. of displacement at their normal rating, liquid-cooled engines were used in the Schneider cup races developing over 1 bhp per cu in. In speed flights the same year as much as 1.13 hp per cu in., or nearly four times the normal rating, was obtained in short bursts. While this performance was obtained with very special fuels, it does demonstrate that the major problem in rating an aircraft engine is not that of *obtaining* a given horsepower but of *retaining* it, in other words, providing satisfactory durability. It is, however, interesting to note that in 1939, several years later, when the normal ratings of engines were approximately 0.50 to 0.60 hp per cu in., that the published flash performance data on similar special fuels had not increased in maximum specific output at all. It is also worth noting that none of the engines used in the cup races has been made available in any production quantities at the reduced ratings necessary for satisfactory durability. This may be because the balance in design necessary for an all-out life of 1 hr is not that required for long-time operation. Of more interest is the steady increase in the normal ratings of engines during the foregoing period and the continuance and acceleration of this improvement until current engines are now providing take-off and military ratings of 2/3 to 3/4 hp per cu in. displacement with cylinder bores of 5 in. or more, and higher values with small cylinders. It is also safe to say that this improvement will continue and probably will be accelerated during this emergency.

Considerable information is available on the development of the larger air-cooled radial engines both in this country and abroad, and of the foreign liquid-cooled engines, but very little has been made available on the liquid-cooled engine development in this country in the last 8 or 10 years, primarily because it did not reach the production stage until 1939. Since the Allison V-1710 engine is the only example of this type in this country currently in production, a discussion of the development of liquid-cooled engines over this period is of necessity confined largely to this engine.

THIS paper recounts, step by step and model by model, the development of the Allison V-1710 aircraft engine – the first liquid-cooled military aircraft engine in production in this country.

Since 1930, when the design of this V-type 12-cyl engine was initiated, the powerplant has been stepped up from 650 hp without supercharging to 1000-1500 hp in its supercharged state, and the weight has increased from about 1000 lb to 1320 lb, the author reveals.

The following three underlying reasons are given for building the first V-1710 engine:

1. That the V-12 type of construction with its small frontal area could be installed with less drag than any other type of engine.
2. That a liquid-cooled engine could be operated at a higher power output per cubic inch due to the type and uniformity of cooling.
3. That the liquid-cooled engine would be more reliable because it is less sensitive to temporary overloads on account of the heat capacity and limiting temperatures (boiling) of the coolant.

Additional advantages discussed are flexibility of radiator installation and of engine location.

V-1710-C Engine Development

The V-1710 liquid-cooled aircraft engine design was initiated late in 1930 after preliminary studies of various arrangements, sizes, and types, when an experimental order for one engine was received from the Navy Department. Prior to this time, the Allison Engineering Co. had for many years divided its small organization about equally between the development and manufacture of its well-known lead-bronze steel-backed engine bearings and miscellaneous contracting for experimental design and fabrication of items requiring quality engineering on a job-shop basis. These items involved such work as the air-cooled Liberty engine conversions, several contracts for overhauling Liberty engines, various reduction-gear developments for Liberty and other engines including an engine gear box coupling of unusual design, the extension shafts and movable bevel-gear drives for the Akron and Macon dirigibles, marine engines, a diesel engine, and a large air-cooled engine. A majority of these developments were for the military services. In 1929 the company was purchased by General Motors Corp. and design studies initiated for a product suitable for development and fabrication by the type of organization available, in addition to the bronze backed steel bearing already well established.

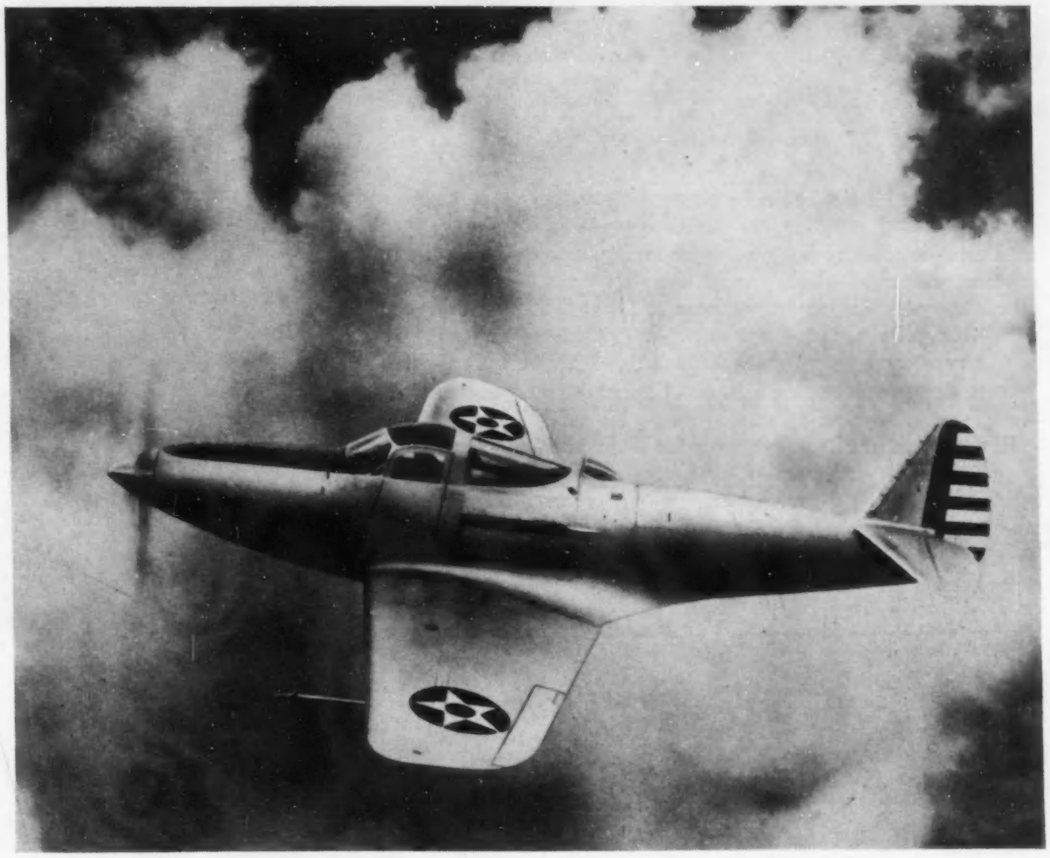
The first V-1710 model was assembled in August 1931. It is shown in side view in Fig. 1. The rating was 650 bhp at 2400 rpm at sea level on 80-octane fuel, using a compression ratio of 5.8:1, a blower ratio of 7.3:1, an impeller diameter of 8.25 in., and a reduction-gear ratio of 1.5:1 at a weight of 1020 lb. Unusual features of the design at that time were the internal gear type reduction gear, the use of a long flexible accessory driveshaft driven from the front of the engine for supercharger drive smoothness, but

[This paper was presented at a meeting of the Detroit Section of the Society, Detroit, Mich., April 28, 1941.]

E

DEVELOPMENT

by R. M. HAZEN
Vice President, Chief Engineer, Allison Division, General Motors Corp.



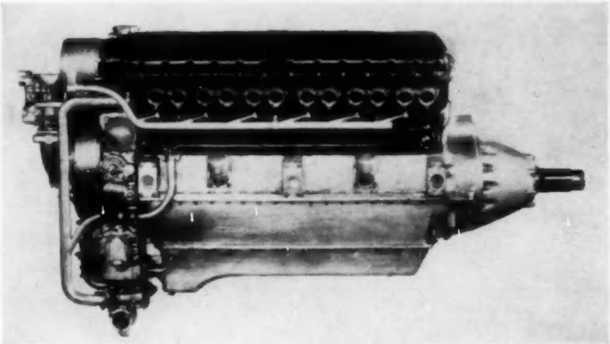
Bell "Airacobra" with V-1710-E engine

which also drove the camshafts and magneto, the cylinder-block construction, the crankcase construction, and the compact supercharger and accessories mounting arrangement. It will be noted that the internal driven gear of the reduction-gear assembly is supported by a large 10-in. diameter bronze-lined steel backed bearing, mounted in the front of the crankcase assembly.

The cylinder assembly was probably the first which was designed definitely for ethylene-glycol cooling at high temperatures. Considerable was known of the difficulties with glycol leakage and the design was planned around a unit which could be tested completely as to both pressure and temperature prior to assembly on an engine. The assembly consists of a cast-aluminum cylinder head for one bank of six cylinders. As shown in Fig. 2, six wet cylinder sleeves are shrunk in this head and the assembly closed by a coolant jacket which bolts to the head and is pulled up against a flange on each barrel by large threaded rings. The cylinder assembly is mounted on the crankcase by 14 long studs to the upper deck of the cylinder head, which take the explosion loads. The coolant distributing tube to the various cylinders is cast integral with the jacket so that there is one inlet to and one outlet from each 6-cyl assembly, making it a simple matter to test the complete block. This basic design with detail modifications has already survived a large increase in output and, from the results of development testing, appears to be capable of going some distance into the future.

The first engine was started fairly early on a 50-hr test at the 650-bhp rating. About half of this test was completed when it was found necessary to modify several parts. Calibration and development were continued until these parts were available. The engine blower ratio was then increased to 8.0:1 and, in 1932, a very satisfactory 50-hr development test at a rating of 750 hp at 2400 rpm was completed. This test was run on a rigid test stand with a four-blade wood test club.

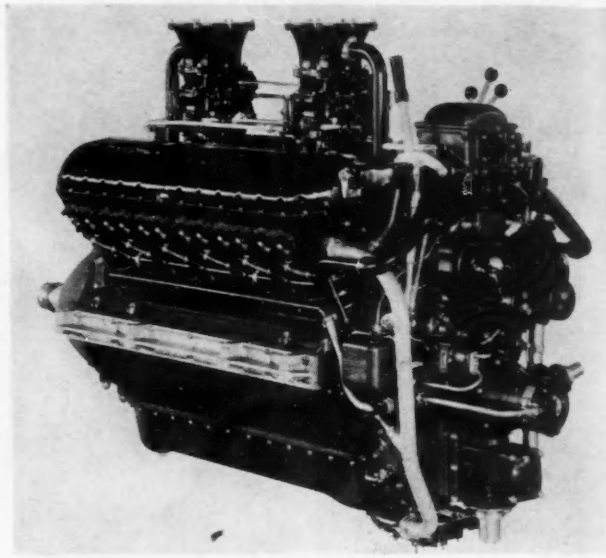
This engine had been developed primarily for use as an airplane engine. At this time it was considered desirable to have a reversing dirigible engine for proper airship control in connection with the swinging propeller mounting then in vogue. The V-1710 engine was therefore redesigned as a reversible unsupercharged or carburetted model, shown in Fig. 3. This model was built in 1933



■ Fig. 1 – Right side view of GV-1710-A engine

and completed model tests in 1935; the reversing feature, which involved an ingenious shifting of camshafts, magneto and distributor fingers, having required considerable development time. This engine would reverse from full power in one direction to the same condition in the opposite direction in a matter of 8 sec. Two additional engines of this type were ready to ship at the time the Akron and Macon were unfortunately lost, resulting in the termination of this particular development.

In the meantime the Army Air Corps became interested in the V-1710 development and obtained a redesigned model, Fig. 4, of the original type which involved the use of a 2.0:1 reduction gear, a long nose for aerodynamic reasons, modified mounting provisions including a major stiffening of the crankcase, a larger-diameter (9½ in.) impeller, and various other improvements looking towards the development of higher power outputs. This engine, known as the V-1710-C type, was delivered in 1933 as a 750-bhp engine at 2400 rpm at a weight of 1122 lb, a 100-lb increase over the original engine. It is shown in right side view in Fig. 4. A 50-hr development test on a dynamometer was completed at a rating of 800 bhp at 2400 rpm in 1934, and an improved version completed a similar run at a rating of 1000 bhp at 2650 rpm in the

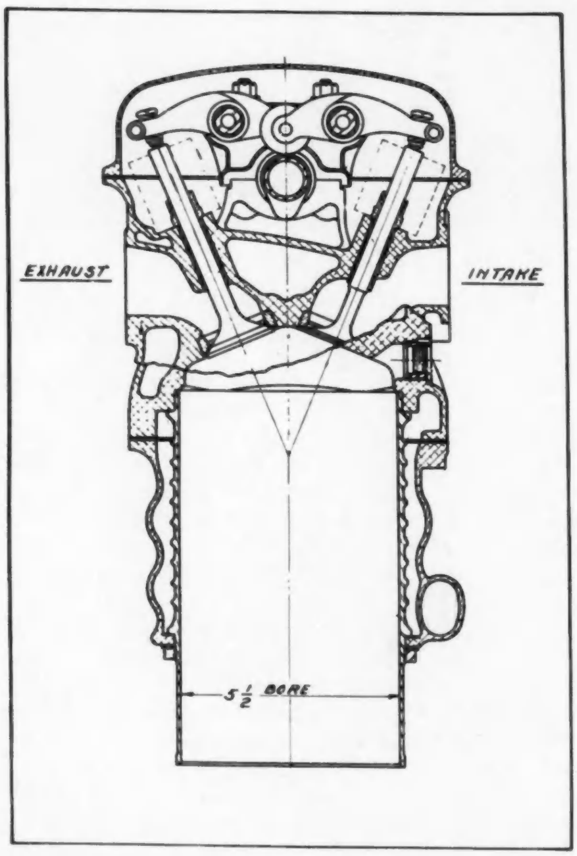


■ Fig. 3 – V-1710-B reversible engine – three-quarter rear view

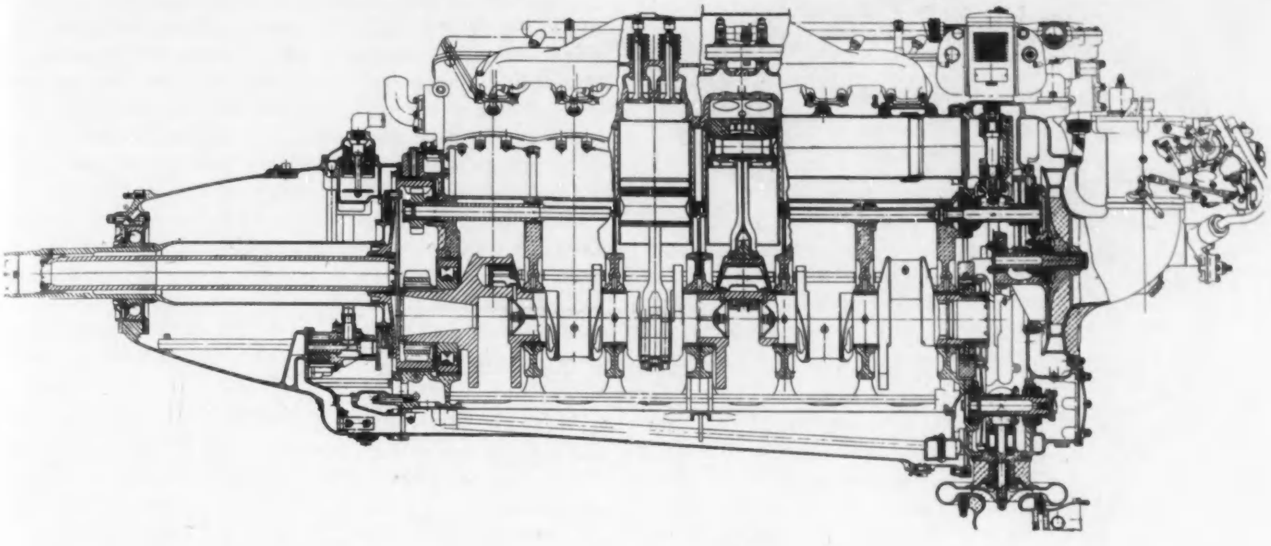
spring of the following year. At the end of this test a few items required correction, but the general appearance of the engine seemed to warrant a type approval test at the 1000-bhp rating.

Improvements were incorporated in the same engine as used on the latter development test, and a 150-hr type test started on a torque stand with a flight metal propeller of the adjustable-pitch type. The engine was mounted using the side pads at approximately the center of mass of the engine-propeller combination, simulating a proposed aircraft installation but much more rigid. This engine, which had done rather well on a diet of dynamometer testing at 1000 bhp maximum, developed acute indigestion almost immediately when run at the same power on the torque stand with flight propeller and an undeveloped mounting. A considerable development period, during which manifold fuel injection was introduced as an additional variable, then followed before it was concluded that a rather complete redesign was necessary to retain the 250-hp increase. This redesign took place early in 1936. Some of the more prominent modifications made during this development can be followed by reference to Fig. 5 which shows a longitudinal section of the C model V-1710 engine.

The long nose in combination with the 2:1 reduction gear resulted in a crankshaft torsional frequency with metal flight propeller below that previously encountered on 12-cyl engines. It was not found possible to keep the 1½-order harmonic above the operating range. This harmonic is

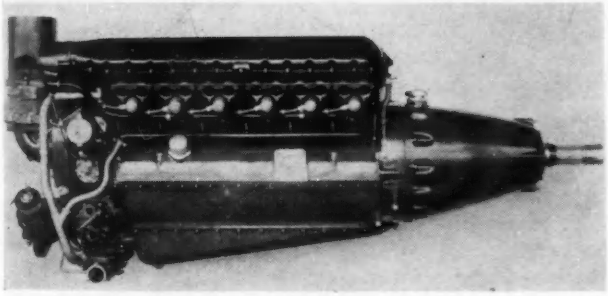


■ Fig. 2 – Cross-section of cylinder assembly



■ Fig. 5 – Longitudinal section of V-1710-C type engine

usually the most severe single node vibration in a 12-cyl 60-deg V-type engine. From the original single propeller shaft, it was found necessary to go to a flexible inner shaft to drop the $1\frac{1}{2}$ order below the operating range. This shaft inadequately supported the propeller in bending and was supplemented by an outer shaft to carry this load. It will be noted that, if torsional vibration of any type occurred, the flexible inner shaft will deflect angularly at the rear in relation to the outer shaft. A friction damper was inserted between these two members which proved very effective on single-node vibration and somewhat effective on two-node as well. This construction proved satisfactory and was carried through all the C and D model engines. This change, in combination with a considerable strengthening of the front crankcheek and crankpin to take the increased torque and overhung crankshaft pinion loading at the higher power output, was found necessary in the very early testing with metal flight propeller. These modifications then permitted sufficient endurance to develop many other troubles incident to the higher horsepower.

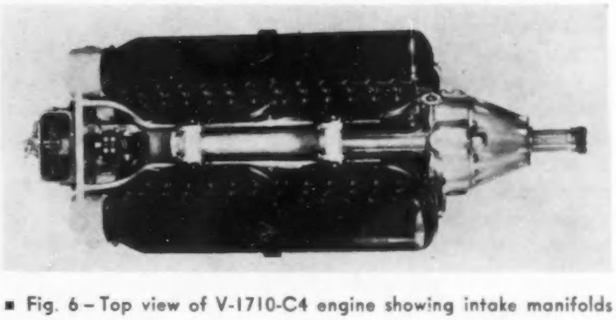


■ Fig. 4 – V-1710-C4 engine – right side view – XV-1710-3

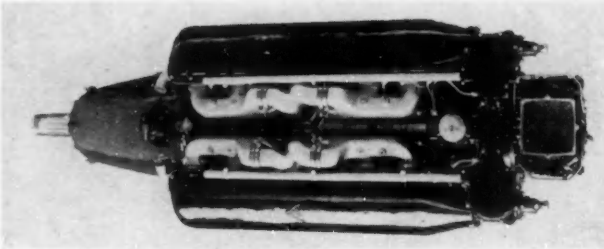
In the major redesign following this running particular attention was paid to the obtaining of uniformity of fuel and air distribution and to detailed improvement of both coolant flow on localized high heat flow points and to coolant distribution. In addition combustion and induction efficiencies were improved by combustion-chamber shape changes to reduce flame travel length and give some piston masking, which resulted in increasing the compression ratio from 5.8 to 6.0:1, by a new valve timing for better volumetric efficiency, and by modified pistons for better heat flow. Narrower piston rings and a deeper top land permitted the clearances to be reduced by more than one-third. These changes resulted in a sufficient increase in power to drop the blower ratio from 8.77:1 to 8.0:1 and to obtain the rating of 1000 bhp at 2600 instead of 2650 rpm.

■ Intake Manifolding

The intake manifold changes are of interest because of their major effect on engine operation. Fig. 6 shows the top view of the V-1710-C4 engine with the original manifolds. From an appearance standpoint it is a neat design and somewhat similar manifolds have been used on several other V-type engines. It will be noted that the main header



■ Fig. 6 – Top view of V-1710-C4 engine showing intake manifolds



■ Fig. 7 – Top view of V-1710-C6 engine – new intake manifolds

from the supercharger branches near the rear to a tee feeding the two rear sets of three cylinders each. A smaller tube runs forward and similarly handles the front six cylinders, three on a side. The modified manifolds are illustrated in Fig. 7 showing the top view of the V-1710-C6 engine. In this case the main header from the supercharger is carried to the center of the engine where it divides laterally while turning a 90-deg bend. At the junction of the main header and the two tees, backfire screens are used not only for the protection obtained but as a material aid to raw fuel distribution. Just above these screens the division to the front and rear sets of three cylinders of each bank takes place, one passage going forward and one to the rear. The 3-cyl manifolds are carefully worked out to oppose centrifugal force to gravity for good raw fuel distribution and for approximately equal ramming lengths to individual cylinders for good air distribution. The importance and effectiveness of the 3-cyl header are better realized by the following comparisons: If the compression pressures in the cylinder are taken over a speed range, with no manifolds on the ports, a certain pressure is obtained at each speed, generally increasing with speed. If a 12-in. straight-in pipe with the same area as the port is attached to the port the compression pressure is increased, particularly at the higher speeds, so that at rated speed the compression pressure is approximately 12 psi higher than without the attached pipe, due to ram. With the manifold shown which has the same axis length but which goes around three 90-deg bends, it has been found possible to obtain exactly the same increase in compression pressures as with the straight pipe. Of importance from a development standpoint on single-cylinder engines, it has been also possible to duplicate not only compression pressures but also indicated specific fuel consumption and combustion turbulence with an excellent correlation to multicylinder work.

The improvement in uniformity of air distribution is illustrated by the compression pressure versus speed curves shown in Fig. 8. The upper curves show the older manifold with the wide spread in compression pressures on various cylinders. Of interest is the effect of ram as speed is increased and the marked effect of the irregular charging interval of 60 deg and 180 deg obtained when the six front cylinders are taken from one header, as can be checked by comparing the 4L, 5L, 6L group with the 4R, 5R and 6R group. These effects are practically eliminated in the improved manifolds as shown in the lower chart. The individual cylinders on this chart lie so close together that it is not possible to show each cylinder clearly so only the maximum, minimum, and average values are given. Converted to horsepower values the upper curves indicate a power variation in different cylinders of $\pm 10\%$ which is reduced to less than $\pm 2\%$ in the lower curves. A similar improvement in maximum power and best economy specific fuel consumptions also was obtained due to improved raw fuel distribution. The correlation between indicated specific fuel consumption on the single-cylinder and the 12-cyl engine is within 0.005 lb per ihp-hr and within 0.01 lb per ihp-hr on the 24-cyl engine which is a measure of the fuel distributing efficiency of this type of manifolding. Specific fuel consumptions on a brake-horsepower basis of less than 0.40 lb per bhp-hr are consistently obtained on multicylinder engines with sea-level blowers at low-speed, high-bmep cruising ratings.

■ Cooling System

A similar development to that of the intake manifolds was carried out on the coolant circulation in the cylinder block. The purpose was to regulate the flow as uniformly as possible in all cylinders, direct a major part of the flow at higher velocities by the hotter parts of the head such as the spark plugs and exhaust valves, and arrange for operation of the system as either a tractor or pusher installation. This purpose was accomplished by the use of sleeve inserts around each cylinder and fitting into the jacket and by metering orifices at each cylinder between the header pipe and the cylinder. The orifices permit balancing the flow at each barrel and, by a relatively simple change in size, give similar balanced flow for a pusher engine with the coolant coming out the accessory end of the blocks. The sleeves accelerate the flow and improve "scrubbing" and heat transfer around the cylinder barrel and, by slots in the top end, distribute the coolant in best proportions to the hotter portions of the cylinder head.

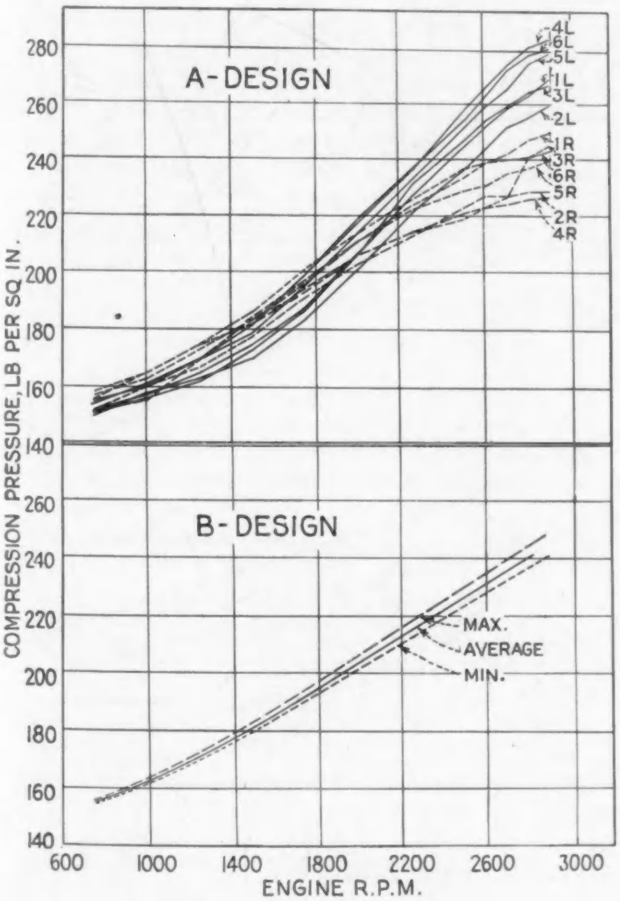
■ Type Test Approval

The redesigned engine incorporating the foregoing developments as well as detail development and strengthening throughout, which involved modifying practically all pattern equipment, was completed, acceptance tested, and delivered for type test in three months and three days. The weight had increased 70 lb to a total of 1230 lb. This engine ran 141 hr of the proposed 150-hr test before trouble of consequence developed. At this point a crack showed up in the top deck of the cylinder head at the center of the block. A new block of the same design was installed and the engine operated on a penalty run to the 245-hr point when the other bank failed in a similar fashion. While distinctly discouraging from the standpoint of losing type approval by such a small margin, the test plus the penalty run did indicate that, with correction of this defect, the engine was a sound design. Until this time the only airplane installation being made was in a workhorse airplane. The important result of this test was the release for fabrication of several engines for new airplane installations and further development.

The cylinder-head failure was mentioned particularly because it brings out the interrelation of all parts of a light-weight engine. Previous strengthening of the top deck of the cylinder head had been based on indications that this deck primarily functioned as part of the structure carrying explosion loads to the long cylinder hold-down studs, rather than as being effective in carrying the bending

of the whole power section due to centrifugal and inertia forces. Since the crack was central between No. 3 and No. 4 cylinders and cross-wise with the head, it could come from bending of the whole power section. Tests were made applying the bending forces at rated engine speed to the crankcase main bearing diaphragms without cylinder blocks in place and with blocks attached. It was found that, with blocks in place, the deflections were only one-fourth of those obtained on the crankcase alone. After stiffening the structure between No. 3 and No. 4 cylinders where the span was greater due to the center main bearing extra width, it was found that a few ounces of aluminum reduced the assembly deflection to one-fifth of that obtained on the crankcase alone. While it had naturally been assumed that the cylinder blocks were of real value in stiffening the whole structure, few of us realized previously what an important factor they were. This is our reason for believing that the block construction will be appreciably lighter per horsepower than an individual cylinder construction.

Another point showing the interdependence was brought out by the cylinder-head change. During the manifold development considerable friction running (no gas loads) was done to measure compression pressures. At speeds around 3000 rpm cylinder hold-down studs would break at the nut in a matter of 20 or 30 min. The failures showed fatigue failures starting on opposite sides of the stud, progressing to the center. It appeared that these studs must be vibrating like violin strings at certain engine speeds, the amplitude and frequency causing very rapid failures. "Muting" of the studs was accomplished by adding a boss about 40% of the stud length from the nut end which had close clearance with the hole in the head. This immediately stopped the failures. However, before the muted studs were available on all engines, the stiffer cylinder head was used with studs not muted. Stud failures occurred at 2600 rpm with the stiffer heads, or at 400 rpm lower speed than previously, the small amount of head stiffening having had a major effect on the tuning of the studs. The mutes again cured the trouble completely.



■ Fig. 8 – Compression pressure versus speed on two types of intake manifolds

A new engine with these improvements, known as the V-1710-C8 model, completed the 150-hr type approval test early in 1937. In spite of the long development period just described, it was the first engine in this country to establish itself at a normal rating of 1000 bhp or more on a military approval test.

■ Further "C" Engine Development

The C engine development then became a matter of detail refinement throughout the engine to permit increased ratings and particularly increased efficiency of the supercharger and induction system. As will be noted on Fig. 9 which shows the power versus altitude development of the C engines, the next step was an increase in the take-off rating to 1150 bhp at 2950 rpm, the normal horsepower remaining at 1000 bhp at 2600 rpm. Detailed refinement throughout the induction system allowed the blower ratio to be reduced to 6.23:1 in place of 8.0:1 used on the type test engine. At the same time it became possible to obtain the take-off and military rating of 1150 bhp at 2950 rpm by an improvement in the bmep characteristics at high speed. Part of this gain was obtained by the use of 6.65:1 compression ratio which was permissible on the same 87-octane fuel, due to the reduced blower ratio. Alterations in the lubrication system permitted diving the engine as high as 3500 rpm. Naturally considerable additional detailed refinement was required to absorb the higher horsepower and the weight gradually increased to 1290 lb. Included in this increase was the change from the float-type carburetor to the Bendix-Stromberg injection-type carburetor with its valuable non-icing characteristics.

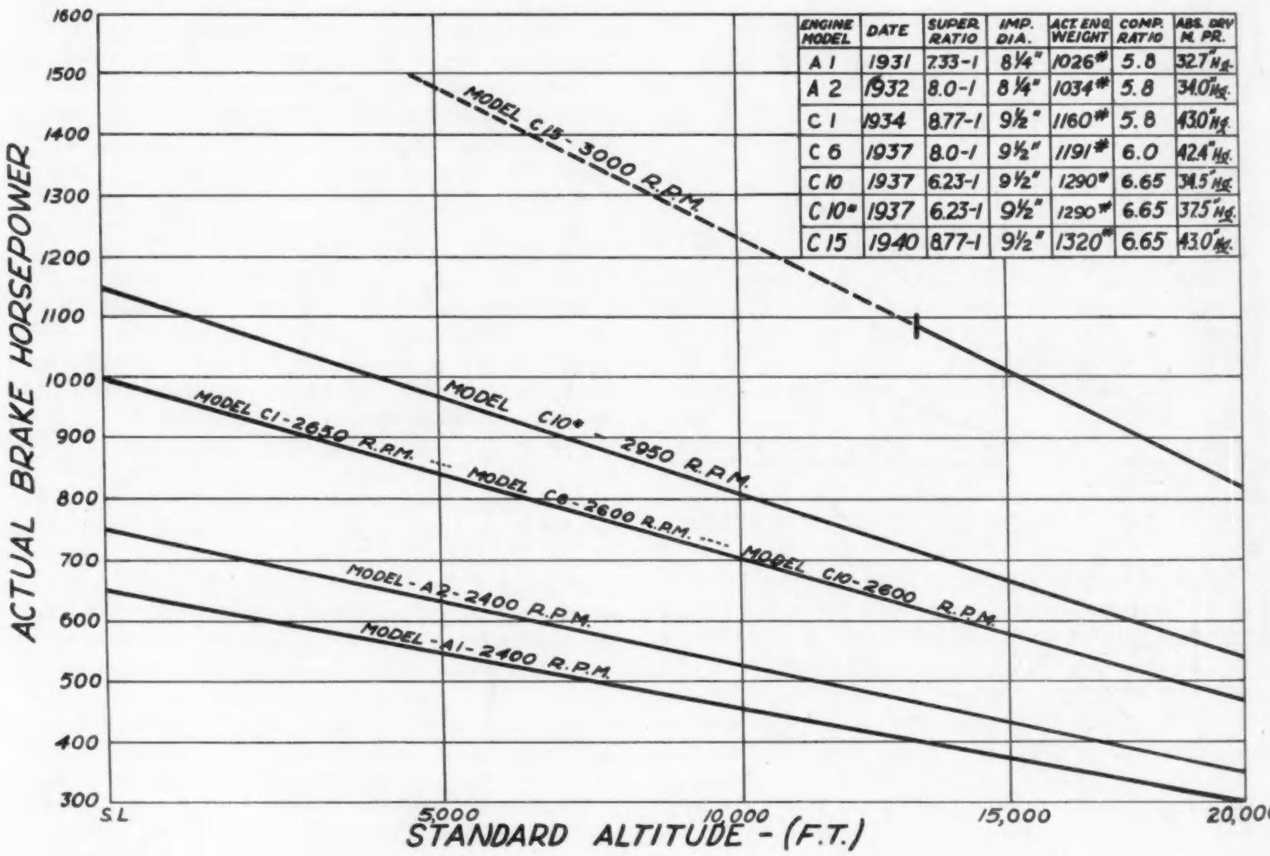
During this same period a left-hand-rotation version known as the V-1710-C9 model was built for use in one position of a two-engine airplane. As can be seen on Fig. 9, all of the C model engines to date had been rated for use with an external turbo supercharger. In 1938 it was considered desirable to widen the application of the engine by the development of an altitude-rated model. A very hurried conversion of the sea-level model to an altitude rated model, known as the V-1710-C13, was installed in the Curtiss XP-40 airplane. With additional supercharger improvements during the flight testing of the airplane a combination was found which won the pursuit competition in the spring of 1939. It is interesting to compare the altitude horsepower curve of the V-1710-C15 engine which was the production model, with that of the V-1710-C1 engine several years earlier which had the same blower ratio of 8.77:1. It also should be noted that, in order to get in step competitively with companies well established in the field, that the Allison Division was forced to build 14 different models in the first 20 engines produced of the

V-1710 size. Of these various models, two were produced for service test of 20 or more, neither one of which went into production. This situation was not due to engine shortcomings but rather to the rapid evolution of pursuit aircraft as the result of the availability of a new and adaptable powerplant. Only one of the 14 models developed over the eight-year period actually went into quantity production which, due to the present emergency, was of sufficient volume to justify the long development period.

■ "D" Model Engine Development

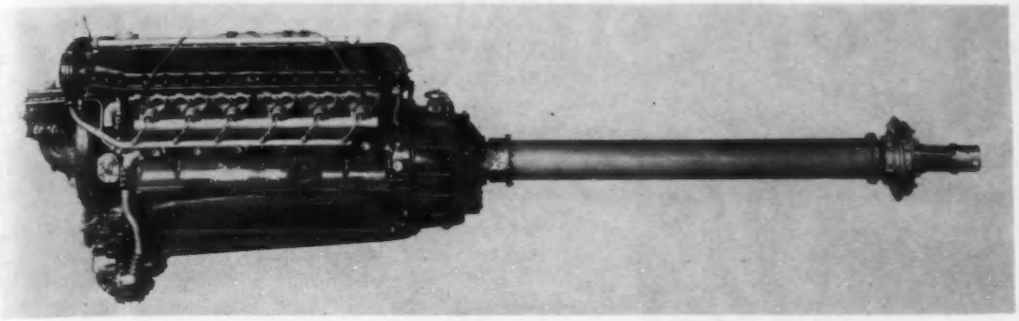
During 1936 and 1937 the first extension-shaft engine developed around the V-1710 basic engine came as an offshoot of the C model. The newly formed Bell Aircraft Co. required an in-line V-type engine with approximately a 5-ft propeller speed extension shaft with the engine installed as a pusher. Since the C model was the only V-engine available at the time, it was decided to make a conversion in which the reduction gear assembly of the C model was removed and a special nose with extension shaft applied to this model engine. The developed engine is shown in Fig. 10. It will be noted that the extension shaft is supported at the engine end by a splined joint and at the propeller end by a rubber-mounted thrust bearing which, in turn, is supported by the airplane nacelle.

The proposed use of the extension shaft was very carefully studied from the torsional-vibration standpoint and it is worth noting that the crankshaft frequency on this combination is almost identical with that used on the long-nose C engine, in spite of the fact that the propeller shaft rotates at propeller speed rather than engine speed and therefore is inclined to be quite flexible torsionally. The torsional studies also indicated that the damping means employed in the C engine, which has previously been described, would probably be inadequate for the extension-shaft arrangement. Torsional vibration tests verified this conclusion. The propeller functions as the flywheel of any aircraft engine. Moving this flywheel some distance away from the friction damper, which had previously stabilized the crankshaft, resulted in the loss of considerable effectiveness of this type damper. While it appeared adequate on the basis of single-node vibration, it became entirely inadequate on the two-node type of vibration. This arrangement therefore required the use of two-node dampers which were applied at the rear of the crankshaft. In order to retain interchangeability of the basic engine, this modification was also incorporated in the C-type engines. Aside from the torsional problems which were anticipated, the extension shaft and drive were practically trouble-free from a development standpoint. One problem arose in the installation of the engine and shaft in the airplane when it was found that the shaft-nacelle-propeller frequency tuned in with some other part of the airplane. The modification of



ENGINE MODEL	DATE	SUPER RATIO	IMP. DIA.	ACT. ENG. WEIGHT	COMP. RATIO	ABS. DRY M. PR.
A 1	1931	7.33-1	8¼"	1026#	5.8	32.7" Hg.
A 2	1932	8.0-1	8¼"	1034#	5.8	34.0" Hg.
C 1	1934	8.77-1	9½"	1160#	5.8	43.0" Hg.
C 6	1937	8.0-1	9½"	1191#	6.0	42.4" Hg.
C 10	1937	6.23-1	9½"	1290#	6.65	34.5" Hg.
C 10*	1937	6.23-1	9½"	1290#	6.65	37.5" Hg.
C 15	1940	8.77-1	9½"	1320#	6.65	43.0" Hg.

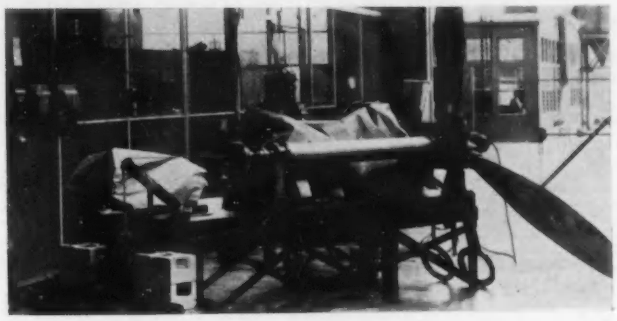
■ Fig. 9 – Power versus altitude development of V-1710-C engines



■ Fig. 10 – Right side view of V-1710-D model engine with propeller speed extension shaft

the outboard bearing which was carried on on a test set-up, shown in Fig. 11, resulted in a modification of the stiffness of the support which cleared the trouble up immediately.

On the test shown in Fig. 11, a ⅛ oz-in. out-of-balance applied to the centerline of the propeller at the critical frequency would raise the 1-ton table on which the assembly is mounted off of the floor. Since the airplane installation was a conventional landing gear type, the high end of the engine on the ground and during climb was the accessory end. As previously pointed out when discussing cooling, the engine already had been provided with metering orifices which permitted its satisfactory operation as a pusher with the coolant outlet at the rear of the block instead of at the reduction gear end. No particular difficulty occurred as a result of the pusher installation, as far as the mechanics of the engine were concerned.



■ Fig. 11 – Extension-shaft bearing support test – D engine extension shaft and propeller

■ Early Flight Installations

Long-time endurance and development on the dynamometer and torque stand are essential on high-output engines. At least as important is a similar flight program. On a new type of engine this test program is frequently not a simple matter. It is particularly difficult to get established in the single-engine pursuit field with a new type of engine for the reason perhaps best expressed as "nothing fits everything."

The first flight installation was made in an obsolete Consolidated two-seater formerly using a Conqueror engine. Due to the differences in dimensions, weight and horsepower as well as propeller thrust locations and other factors, the Allison installation was not a particularly happy installation from the flight-control standpoint. It was therefore used to obtain 300 hr flying time as soon as possible and future flying discontinued. The engine gave a good account of itself in this installation. The first of the more modern installations was made in the Curtiss XP-37 airplane early in 1937. Due to the fact that this airplane was, in reality, a modified design of a radial air-cooled installation, the full benefits of the liquid-cooled V-type engine could not be shown to maximum advantage. As can be seen from the view of this airplane in Fig. 12, an unusual pilot location was necessitated by the application of the in-line type engine to a basic airplane designed for a short, large-diameter radial engine and also to the application for the first time to high-powered engines of a turbo supercharger. The installation of the D-type engines in the Bell *Airacuda* and the first flight test of this unusual airplane-engine combination, shown in Fig. 13, took place in September of the same year. Air Corps orders for service test quantities of both of these articles followed a few months after the initial flights and accounted for such engine production as was reached by Allison during 1937, 1938, and most of 1939. It appeared during the flight development period that, while there were important advantages to the in-line engine turbo combination, an intermediate step prior to the full use of this combination would be advantageous. This step was the development of a model engine with its own supercharging which was made and installed in the Curtiss XP-40 airplane in the fall of 1938. See Fig. 14.

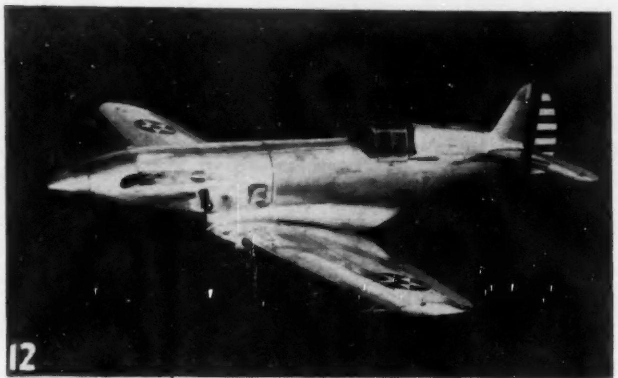
The simplicity of the altitude-rated engine installation and the fact that it reached altitudes at which the pilot could function satisfactorily without a pressure cabin made it an attractive combination from the production standpoint. As is well known, this combination was put into production in 1939 and a large number of these planes are in service both here and abroad. See Fig. 15.

A new airplane-engine manufacturer entering any field must not only present a reliable product but is forced to extend himself to show definite advantages over well-established products already in the field. This, in general, results in rather high pressure to make specialized models in order to obtain a sufficient variety of installations to warrant his being in the business. As can be seen from the foregoing, this pressure already has resulted in the D model and, late in 1937, also resulted in the construction of a left-hand-rotation C engine for use in the Lockheed XP-38 airplane.

■ Current Model Engines

There were three underlying reasons for building the first V-1710 engine, namely:

1. That the V-12 type of construction with its small frontal area could be installed with less drag than any other type of engine.



- Fig. 12 – Curtiss XP-37 airplane with V-1710-C10 engine
- Fig. 13 – Bell "Airacuda" with two pusher model D engines with extension shaft
- Fig. 14 – Curtiss XP-40 airplane with V-1710-C13 engine
- Fig. 15 – Production P-40 airplane with V-1710-C15 engine

2. That a liquid-cooled engine could be operated at a higher power output per cubic inch due to the type and uniformity of cooling.

3. That the liquid-cooled engine would be more reliable because it is less sensitive to temporary overloads on account of the heat capacity and limiting temperatures (boiling) of the coolant.

These advantages are fundamental and inherent in this type engine, particularly as compared to the radial air-cooled engine. The percentage advantage is hard to evaluate, especially with the rapid improvement in cylinder-cooling and engine-cowling technique by the air-cooled engine and airplane companies but, assuming equal engineering and development facilities, the basic advantages still hold.

Additional Advantages

As the development of the V-1710 engine was carried on, it was found that other important advantages in the V-type liquid-cooled engine were available if properly utilized. Two of these, particularly applicable to high-speed airplanes, are:

A. Flexibility of Radiator Installation – The drag of a heat transferring element is often thought of in terms of the mass air flow and the pressure drop. More often low duct entrance loss, efficient velocity converting ducts to and from the cooling element, and shockless exit to the airstream are more important factors on the total airplane drag. The liquid-cooled engine with the radiator located at the most advantageous point in the airplane for low drag has an inherent major advantage. The flexibility in radiator dimensions, and therefore low-loss ducting to suit any particular application, as compared with the inflexibility of air-cooled cylinder location and dimensions, is also important from a drag-reducing standpoint.

B. Flexibility of Engine Location – The fact that the cooling element for a liquid-cooled engine is independent of the engine permits the engine to be moved around to any location best suited to the desires of the airplane designer without particular regard to the cooling blast from the propeller. This opens up a new field for the airplane designer in locating the powerplant where it gives the optimum in airplane characteristics, in any particular combination desired for a given purpose.

While the C was passing its tests at 1000 bhp, the D was under development and the first flight installations were being made early in 1937, the paths to Indianapolis for airplane designers anxious to install the new powerplant were noticeably green from untrodden grass. A check-up was made of the companies usually interested in new powerplants. Among other things, their comments were that, while the V-1710 was an interesting engine:

Airplane Designers' Comments

1. The reduction-gear nose which had been made long for aerodynamic reasons should be shorter or longer.

2. The thrust line should be higher or lower.

3. The reduction-gear ratio (2.0:1) should be 2.25:1, 1.75:1, or dual rotation.

4. The downdraft carburetor should be updraft or side entrance.

5. The engine should be inverted, a horizontal flat engine, or a vertical opposed engine.

6. The D engine with a 5-ft extension shaft should have a 3-ft shaft, a 10-ft shaft; the shaft should be supported from the engine nose or it should run to an external gear box.

7. The sea-level models should be integrally supercharged, have built-in turbos, should be two-speed, and/or two-stage.

8. The engine should be able to run both right-hand and left-hand.

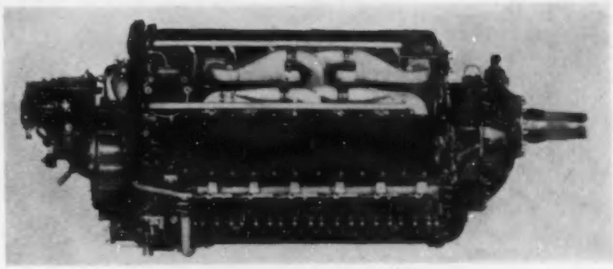
9. The engine should have up to three additional accessory drives or a separate accessory box.

While these comments are amusing when all put together, it should be remembered that the type of applications varied widely. In general, there were good sound reasons for every one of the particular items desired. We were impressed by the variety of requirements and by the necessity for as high a degree of flexibility of application as could be made available. It is also worth noting that all except one or two items do not affect the power section of the engine, but that any number of variations are desired in accessory arrangement, gear ratio, and shaft drives.

When forward studies of newer models were made, the fundamental precept followed was that the power section would be treated as a separate unit, as simple and rugged as possible with no built-in features which would detract from its value for any application. With the same thought in mind, it was decided to incorporate both single- and two-node crankshaft torsional damping suitable for any type of extension shafting which might be required, engine mounting for rubber bushings or through-rail type, crankshaft rotatable in either direction by using symmetrical ends, and lubricaton provision to suit.

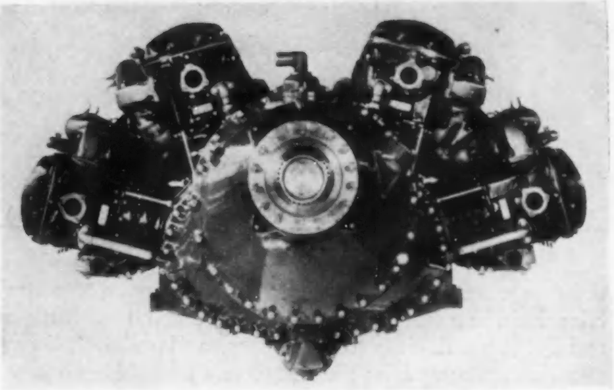
■ Model V-3420

Our first opportunity to put these plans into effect was on the V-3420 model engine (see Figs. 16 and 17), a four-bank version of the V-1710 for which an Air Corps experimental contract was received in 1937. The crankcase consists of three sections, the center section separating at the centerlines of the two crankshafts on a 15-deg angle with the vertical, with the top side narrow. The side cases are made from a single pattern. The bolting on flanges at



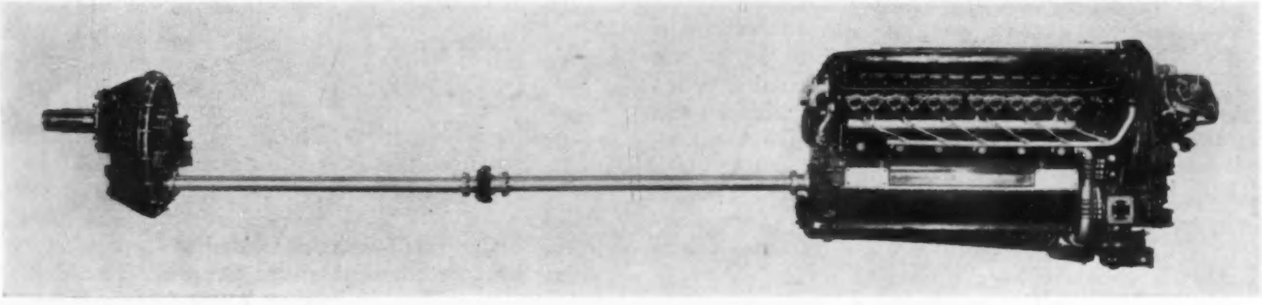
■ Fig. 16 – V-3420 engine in side view

each end are identical and inside of the cylinder pad faces so that through milling can be employed on all faces. The outer bank cylinder decks are parallel to the crankcase parting line which aids machining and simplifies inspection. Crankshafts have symmetrical flanged ends drilled and reamed to a jig so that driving plate assemblies bolted on determine the direction of rotation. The crankshaft

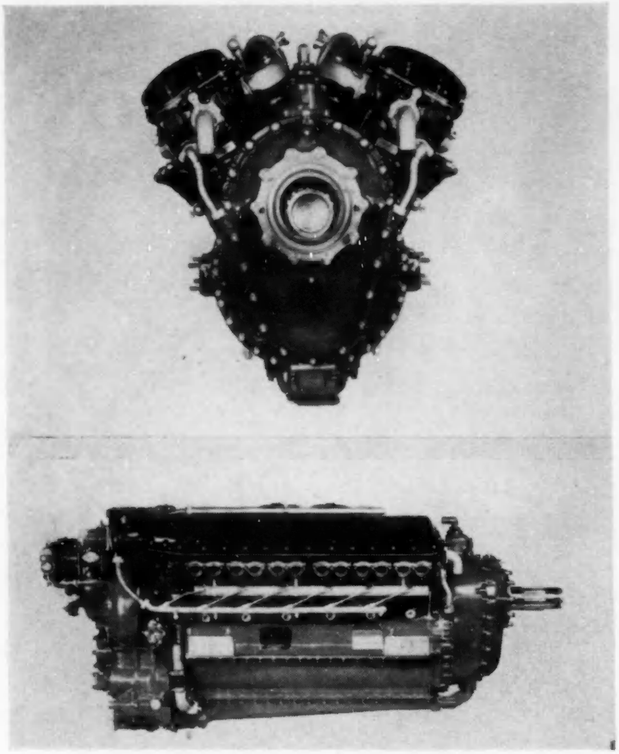


■ Fig. 17 – Front view of V-3420 engine

when turned end for end and rotated in the opposite direction thus gives the same firing order per bank so that camshafts and intake manifolds do not have to be changed for either direction of rotation. The newer V-1710 was kept in mind when designing this power section. As regards interchangeability on the new E and F V-1710 models, all parts of the power section including crank-



■ Fig. 18 – Side view of V-1710-E engine, shafting and gear box



■ Fig. 19 – V-1710-F engine – front and right side views

shafts, connecting rods, pistons, complete cylinder assemblies including valve gear and hold-down studs, intake manifolds, spark-plug cooling tubes, and radio shielding and ignition assemblies are completely interchangeable on all three models and for either crankshaft rotation. This leaves only the crankcase assembly and main bearings (which differ only in final machining) non-interchangeable in the complete power section. This simplifies not only the tooling for a high-horsepower engine but has a major effect in reducing time and development expense on the V-3420 as higher outputs are obtained on the V-1710 series.

The reduction gear bolts on the front of the crankcase and is doweled to same. Several parts of this gear are interchangeable with the reduction gear on the V-1710-F model. It is to be noted that full engine torque is transmitted only through the propeller shaft since pinions at each crankshaft center each carry half of the load.

The accessory housing is designed for building up for either crankshaft rotation simply by the addition of an opposite-hand starter dog. Each crankshaft has its own damping provisions, but the dampers are geared in such a way as to damp between shafts as well. With this arrangement practically any type of extension shaft and reduction-gearing combination can be applied to the engine. It is also interesting to note that the crankshafts can be rotated in opposite directions with special machining of the accessory housing which gives a practically zero torque powerplant with advantages for single engine installations.

The engine showed up well in its development tests but, due to the emergency and great enlargement in plant and personnel, had to take second priority in favor of engines well tried out in available airplanes. The rapid development of V-1710-E and F models and available production tooling for so many interchangeable parts brings this engine back into the picture.

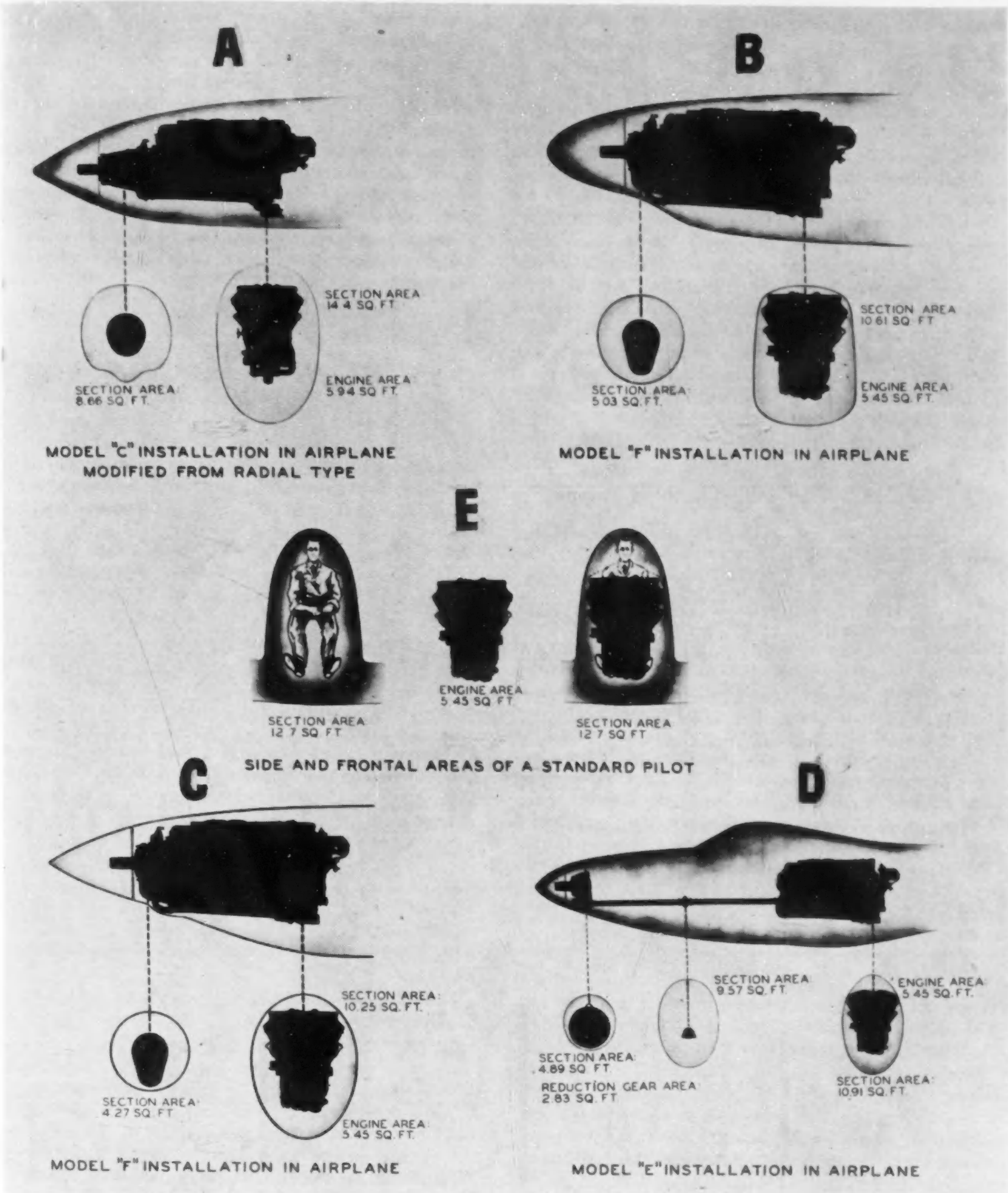
■ V-1710-E Model

This design, Fig. 18, followed closely the V-3420 and, of course, was accelerated by the interchangeability already mentioned. Left-hand crankshaft rotation was necessitated by the requirement of the outboard gear box offset for cannon provision and a right-hand tractor propeller. We approached this design, as requested by Bell Aircraft and the Air Corps, with a great deal of trepidation, knowing that other people such as Koolhaven and others in Europe had tried and dropped the idea. The problem can be appreciated better if someone asked you to put the flywheel on your automobiles out back of the garage and still make the engine run smoothly. With the propeller nearly 10 ft from the engine we wondered too. Very careful torsional analysis indicated that we would have a lower-frequency system by far than we had ever tried before but that we would get the best torsional characteristics we had ever had on an aircraft engine, if our assumptions were right. The torsional characteristics checked by a torsiometer were excellent throughout the range normally checked. The torsiometer used had in general been unsatisfactory in the warm-up range on any engine so that, when the torsiograph failed at low speed, we did not think much of it, particularly since the carburetion had not been worked out. We ran 40 hr or more of severe testing without any trouble and were about convinced the millennium had come. We then ran the engine one day for 20 or 30 min at low speed when it suddenly stopped. A flexible shaft in the rear of the engine had failed. We tried it again and it happened sooner. There followed a hectic month or two in which everything in the way of damping and quill-shaft flexibility and stiffness was tried with no success. We finally discovered that what was occurring was a "beat" condition which gradually built up amplitudes in phase and then suddenly went 180 deg out of phase with extreme amplitudes resulting. There was at no time any trouble with crankshaft, front-end extension shaft, or reduction-gear box as witnessed by the fact that the original shafting and gear box are still operating after hundreds of hours of hard testing and after wearing out a couple of engine power units. The special damper application cured completely this trouble and this engine now has the best torsional characteristics that we have ever seen on any engine. The well-known Bell *Airacobra* (*see illustration on page 489*), using this engine, has one of the most interesting combinations of features in the pursuit field.

■ V-1710-F Model

This model, shown in Fig. 19, is now completely interchangeable with the E type having the same supercharger ratio. It is furnished as both right and left hand in the sea-level models and as right-hand tractor in altitude-rated engines.

This engine, while designed for and well proved at considerably higher outputs than the C model, is some 30 lb lighter, has 10% less frontal area, a propeller thrust line at the center of area of the engine which permits shorter landing gears and smaller cowling, and is 10 in. shorter than the C engine. This combination adds up to a considerable all-around performance gain possible.



■ Fig. 20 – Shadowgraphs of various airplane forms around V-1710 engines

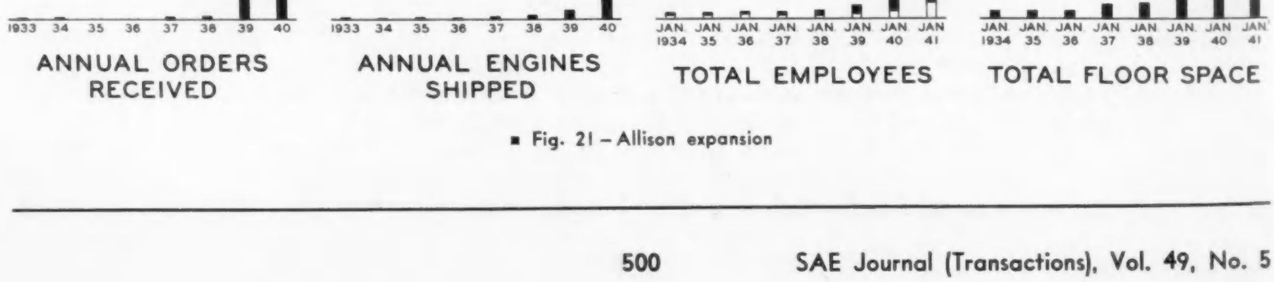
The shadowgraphs of V-1710 engine installations with various model engines are shown in Fig. 20 with a "standard pilot" size for comparison. Form A is a C-type installation in which the large fuselage in relation to engine size is partly due to the engine thrust location in relation to area of engine and partly due to the fact that this fuselage was a modification from a radial engine installation. The smaller fuselage dimensions in Forms B and C show the advantages of the F-model engine in relation to the C as well as the improvements possible with airplanes designed for a V-type engine only. Form D with the E engine shows the wide latitude allowable for armament and other installations without appreciably affecting airplane size permissible with this engine. The profile of the standard pilot shown in Form E shows the rather close correlation between engine profile and man profile. It has always been our feeling that there would be an application for the smallest airplane with high performance which would enclose a pilot. This is, and probably for some time will continue to be, the airplane most economic to produce, easiest to tool and capable of the most production which can be made available. It also appears that the power and altitude characteristics of these engines can be increased rapidly enough to keep this type of airplane well out in the fore-front as a military type.

■ Air-Cooled Versus Liquid-Cooled Engines

A paper on liquid-cooled engines would not be complete without mention of the so-called "controversy raging between proponents of air- versus liquid-cooled engines" which we have all read about recently. Due to military restrictions this discussion cannot be aired as completely as desirable at this time. The following facts are known and submitted for your consideration. Until the liquid-cooled engine entered the pursuit field the average yearly increase in speed of pursuit aircraft was 10 to 15 mph. The first year that a liquid-cooled engine was in competition the increase in speed was approximately three times the maximum previous annual increase. It has been reported on good authority that, on similar installations with engines of approximately the same horsepower, the liquid-cooled engine cruised about 60 mph faster (and therefore farther in one hour) on the same number of gallons of fuel per hour. This point is of interest not only in pursuit airplanes but in commercial service and bomber types as well. It should also be kept in mind particularly with reference to the recent comparisons between 2000-bhp air-cooled engines versus 1150-bhp liquid-cooled engines. On this point it is worth noting that, while 1500- and 1800-bhp air-cooled engines have been available for some time, the one-sided controversy on top speed was not brought into the "raging" stage until 2000-hp air-cooled engines had become available. Comparison of current production models with experimental airplanes which have not gone through their "shake-down" tests naturally favors the latter. It also should be apparent that, since large-horsepower airplanes must weigh more, cost more, and use more fuel, fewer of them can therefore take the air to do a job. This precept conflicts with the basic war-time idea of "getting there fustest with the mostest men."

■ Expansion

While this paper deals primarily with the development of the Allison engines, the expansion of the company may be of some interest. This expansion is summarized graphically in the four charts shown in Fig. 21. Comparable data as of January first of each year is given in A for engines on order, B engines delivered, C total employees, and D square feet of floor space. For obvious reasons no actual figures can be given. It is, however, only fair to state, in view of adverse publicity on engine deliveries, that more engines were delivered in each of the years 1939 and 1940 than were on order January 1st of the respective years. As a matter of fact, more engines had been delivered by the end of February, 1941, than were still on order 9 months prior to this time. Both the relative personnel and manufacturing space expansion appreciably preceded engine orders. This expansion, only possible through the resources in both personnel and capital of General Motors Corp., is another story which some day should be told. The tolerance, foresight, and technical and financial help of the U. S. Army Air Corps during the long development period were primarily responsible for the availability of the Allison engine for this emergency.



■ Fig. 21 – Allison expansion

Design of AIRSCOOPS for AIRCRAFT CARBURETORS

by M. J. KITTLER
Chief Engineer, Aircraft Division, Holley Carburetor Co.

WITH the increasing use of fully automatic carburetors, it was found to be more and more important to make sure that all elements of the installation were carefully worked out so as to insure accurate metering of the carburetor under all operating conditions. Experimental flight investigations demonstrated that this was particularly true of the carburetor airscoop, which admits the air to the carburetor and can therefore cause flow disturbances which may seriously affect carburetor metering.

Test laboratory investigations were made of a considerable number of airscoops in an effort to determine what factors contributed to the proper or improper functioning of the scoop. It was found that the shape and size of the scoop passages and also the location of the hot air valves were important factors, and a number of general design criteria were formulated, and these are discussed in detail. A discussion of the ram characteristics under various conditions of flight and for various sizes of scoop is also included.

An actual production model of airscoop was designed in which these various features were included. The aerodynamic and structural features of this airscoop are discussed and explained in detail.

Five simple rules are given to assist the designer in laying out an airscoop which will incorporate the design features which were found to be desirable.

IN the past few years, there has been a general trend to the use of aircraft carburetors of the fully automatic type in which the carburetor is expected to provide the correct fuel-air ratios for all conditions of engine power output, altitude, and temperature. To meet this requirement meant, in effect, that the carburetor had to be an extremely accurate, wide-range air meter which would regulate the flow of fuel in correct proportion to the mass flow of air, regardless of the temperature, pressure, or rate of flow of the air which the engine consumed.

When calibrations and tests were run in the carburetor laboratory on fully automatic carburetors, it often was found that very excellent results could be obtained for the various conditions to be encountered in flight operation. However, when the same carburetor was actually installed on an airplane, a flight test run, and the carburetor performance measured by means of flowmeters, torque meters, and fuel-air ratio analyzers, it was found that some of the high hopes built up during laboratory tests could not be realized in actual service. If no serious fault in the installation could be found, it was usually necessary to proceed along more or less empirical lines and work out some practical solution to the problem on the basis of flight-test results. This method would, in some cases, give fair results over a limited range of conditions. In other cases, it was found that no reasonable change which could be made in the carburetor calibration would provide even passably satisfactory operation under actual flight conditions.

As is the usual thing in experimental and development work of this type, every component of the installation which had to do with the matter of carburetion was suspected and investigated in turn, with the hope of finding one of them which might give a clue to the trouble. It was in one of these programs that a considerable amount of laboratory work was done on the carburetor airscoop and a surprising number of interesting facts brought to light.

In attempting to glean more information on this subject, it was learned that a good deal of high-grade engineering work had been done on investigations of condenser scoops for boats, elbows and ducts for ventilating systems, elbows for power plant condenser installations, and even on locomotive scoops for picking up water on the run. Surprisingly enough, there was a decided lack of information on airscoops for aircraft carburetors, although the subject certainly seems to merit such work.

In the past year, a considerable number of different designs of airscoops have been tested by the Holley Carburetor Co., and each one has contributed something to

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 13, 1941.]

the store of information on this subject. Out of the mass of data which was accumulated, it has been possible to pick certain design characteristics which would contribute to good performance in flight. The general problem of the flow of air through ducts and elbows is not a new one, and much of the theoretical side of this subject is treated in a general way in standard texts on fluid mechanics and kindred subjects, so that no attempt will be made in this paper to include any profound theoretical discussions.

All scientific reports on air measurement and air flow through ducts stress the importance of having long, smooth, straight passages properly supplied with screens and honeycomb sections for assuring streamline flow. It is manifestly impossible to meet these fundamental requirements in a design of an airplane airscoop because of space limitations, although the specifications for accuracy are not relaxed in the least because of the necessarily imperfect air-handling system.

Since practical considerations make it impossible to provide for an airscoop of perfect design, the main purpose of this paper will be to present such information and design criteria as can be used in the working out of a commercially satisfactory scoop design for a present-day engine installation with all of its limitations.

Requirements of a Practical Airscoop

Some people concerned with aircraft carburetion problems feel that aircraft carburetors should be designed to work correctly regardless of any installation considerations, and that the design of the airscoop should have no effect on the functioning of the carburetor. This would certainly be an ideal condition, but unfortunately, it cannot be completely realized.

The carburetor metering characteristics for variations in rate of airflow, for changes in air density, and for changes in air temperature, are determined by the response of the carburetor mechanism to variations in air velocity, pressure, and temperature. Anyone who has worked on problems involving the measurement of air flow in ducts has realized the difficulty of obtaining true average values of air velocity, pressure, and temperature, especially if the duct happened to be fairly large in size and the velocities fairly high. The great range of velocities, pressures, and temperatures encountered in an aircraft carburetor adds further to these difficulties. In this connection, it is interesting to note that, in so far as the author is aware, there is no air meter available on the market which will handle the quantities of air handled by a large-size aircraft carburetor, over the wide range of conditions under which a modern carburetor operates, and read automatically to the limits of accuracy expected of a modern carburetor. It would therefore seem reasonable that the carburetor should be supplied with air that is flowing at fairly uniform conditions of velocity, pressure, and temperature across the area of the carburetor entrance. This, then, is the principal job that the airscoop has to do.

Besides this, modern commercial installations call for the use of an airscoop which is equipped with a heater valve that will provide approximately 100 F of temperature rise for the carburetor air. The airscoop must therefore be so designed that with the heater valve in the full-cold position, the full-hot position, or any intermediate position, the velocity, pressure, and temperature of the air entering the carburetor are reasonably uniform. In addition to providing the proper uniform flow of air to the carburetor, the scoop is expected to provide the maximum possible amount of ram, and should also be free from potential icing difficulties.

In a discussion of some of these items, various sketches of scoops will be shown. The design of these scoops is purely for purposes of illustration, and any resemblance to any actual airscoops past or present is coincidental.

General Design Considerations

In Fig. 1 is shown what would be a practically ideal airscoop from a carburetion standpoint. This would consist in general of a long, straight duct attached to the carburetor entrance flange and equipped with screens and a honeycomb to provide smooth, straight, and well-distributed airflow. Most authorities agree that at least 10 pipe diameters of straight pipe should be installed ahead of any point where measurements are to be taken, and this proportion has been used in the design of this scoop. The elbow is supplied with turning vanes and the mouth of the scoop has a well-rounded entrance. The heater valve is located ahead of the turn, to cause a minimum of flow disturbance and to provide a maximum length of duct for mixing of the cold and hot air streams. The design shown is obviously impractical and is only given to illustrate the magnitude of the compromises which it is necessary to make in order to reach a practical design for an actual installation.

There is one factor which we may borrow from this "ideal" design of scoop, and that is the straight section immediately ahead of the carburetor. While we certainly cannot use 10 diameters of straight pipe, we can and should go the farthest possible distance out from the carburetor with a straight pipe free from any heater valves, sudden changes in shape, or other disturbing elements.

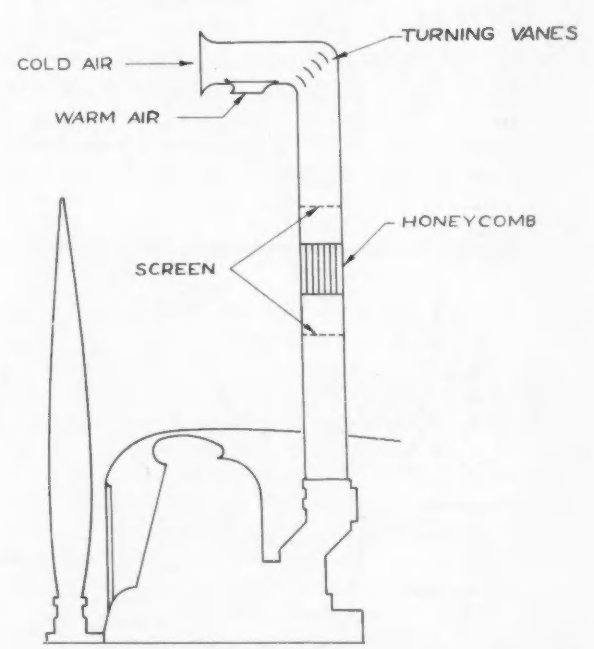


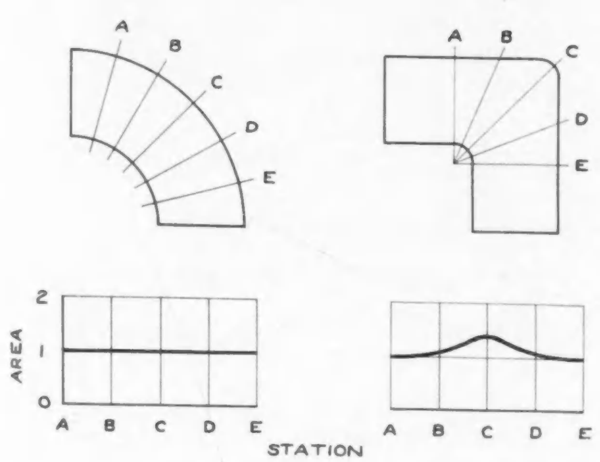
Fig. 1 – Scoop proportioned to give good air-flow characteristics

This is design criterion number one, and it is recommended that, in the design of a practical airscoop, the first lines laid in should be straight lines from the top of the carburetor flange and at right angles to it.

Rule I: Provide the maximum length of straight section of scoop immediately ahead of the carburetor.

Elbows and Turns

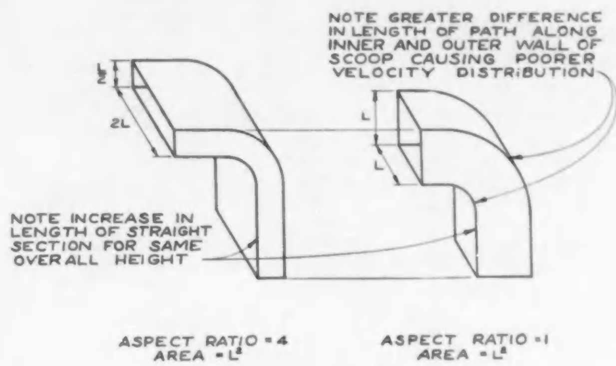
The elbow or turn in the scoop is usually the one feature which causes the maximum amount of difficulty and, paradoxical as it may sound, most of the trouble is caused by the efforts which are put into the matter of providing a smooth turn. Quite a few designs of scoop have hardly a linear inch of straight section in them, the turn or elbow portion often beginning at the carburetor flange and continuing to the mouth of the scoop. In such cases, particularly where the turn is of a rather short radius due to space limitations, the air entering the scoop simply cannot proceed through the scoop at high velocity without piling up along the outer radius. Scoops shaped as shown in the left-hand view in Fig. 2 invariably have a considerable pressure build-up from the inner to the outer radius, and have therefore correspondingly poor air distribution at the carburetor entrance.



■ Fig. 2 – Comparison of smooth-radius and sharp-turn elbows

Following Rule I as just given will prevent the design of this type of scoop and will leave the apparently incorrect solution of a substantially right-angle turn in the scoop. The abrupt turn is definitely a desirable feature, provided that the cross-sectional area through the turn is not restricted by the use of a large radius for the outside of the turn.

Fig. 2 shows some reasons why a sharp turn is better than a smooth radius. It should be noted particularly that the actual area through the sharp turn is approximately 40% greater than the area in the straight portions of the elbow. This expanded region provides for a place for the air to slow down while it makes the turn, and then speed up and proceed in the new direction. In a smooth-radius elbow, the air is continually thrown against the outer radius by virtue of its centrifugal force, with the result that there is no opportunity for the flow to become equalized across the area of the passage.



■ Fig. 3 – Advantages of high aspect ratio

A number of turning vanes installed at the elbow are of considerable help aerodynamically, although they are definitely objectionable from the standpoint of icing, since they provide additional projected area on which impact ice will form. In general, it is believed that turning vanes should be eliminated unless their use is shown to be essential, and unless other means have been provided to prevent them from becoming icing hazards.

In most types of scoops which project above the cowl line, turning vanes would be considered an icing hazard. In under-cowl types of scoops, the passage leading up to the elbow usually has a number of bends in it which would be vulnerable to impact icing. Turning vanes could therefore be used with scoops of this type without adding any hazards.

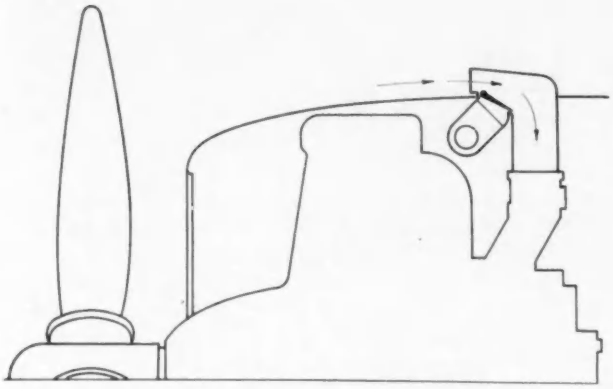
Rule II: Use a sharp right-angle elbow – not a smooth turn.

Shape of Passages

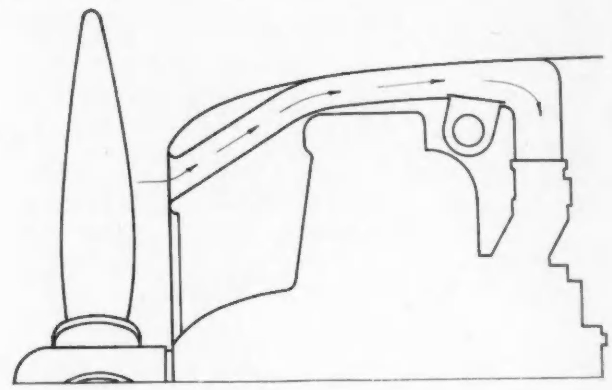
In the case of most under-cowl types of scoop, the natural space limitations call for a design of passage which is rather low in the vertical dimension and rather broad in the horizontal dimension; that is, it has a high aspect ratio. Scoops which project above the surface of the cowling sometimes have this shape, and sometimes also have a shape more nearly approaching the circular. A duct with a high aspect ratio is definitely recommended, and it should be continued in this form as close to the actual elbow as possible. A brief study of the two ducts and elbows shown in Fig. 3 will bring out the advantage to be gained by adhering to a high aspect ratio for both the straight portions of the scoop and the elbow itself, wherever possible. In the case of the high-aspect-ratio elbow, the velocity along the inner and outer walls of the scoop differs by a materially smaller amount than is the case in the scoop elbow of low aspect ratio. Another example of this general principle is the use of turning vanes in an elbow. The turning

valve of high aspect ratio is used. These few simple requirements automatically solve most of the problems pertaining to the heater valve. By putting the valve around the corner and ahead of the elbow, the turbulence produced in the elbow is used to assist in thorough mixing of the hot and cold air streams. Since the valve is at a maximum distance from the carburetor entrance, the pressure disturbance caused by variations in the valve position becomes a minimum.

It is quite important also to provide hot air passages both in the scoop and in the air heater of adequate size to permit operation in the full-hot position without undue loss of power due to restrictions. This requirement can be met more easily if the mouth of the hot-air passage in the scoop is rounded to provide a better entrance coefficient.



■ Fig. 4 – Representative design of projecting scoop



■ Fig. 5 – Representative design of under-cowl scoop

vanes may be considered as changing a single turn of low aspect ratio to a multiple number of turns of high aspect ratio, thus reducing the change in velocity between the inner and outer radius of each turn.

In most scoop designs there is a definite limitation in overall height, and here again the high aspect ratio design is advantageous in that it permits a greater straight length of duct at the carburetor for a given area and given total height.

A third advantage of high aspect ratio is that the loads on the heater valve are reduced, as described in greater detail in the paragraph on heater valves.

Rule III: Maintain a high aspect ratio in the scoop passages and elbow, in so far as possible.

■ Heater Valve Location and Design

The function of the heater valve is to provide for the admission of hot air to the scoop in quantities ranging from zero to 100% of the total. This air must be admitted to the scoop in a manner such that it will be well mixed with the main air supply where only a partial amount of hot air is used, and such that the pressure distribution at the entrance to the carburetor will not be seriously upset. Since ordinarily the air flowing through the main duct of the scoop is subject to some ram, whereas the heated air is usually under more or less static conditions, it is especially important that good pressure distribution at the carburetor entrance be maintained. In many designs of scoop where the heater valve is immediately above the carburetor entrance, it is a practical impossibility to obtain good temperature and pressure distribution at the carburetor, due to the extremely short distance allowed for the air to become mixed and its pressure equalized.

To correct these troubles, it naturally follows that the heater valve should be located in a place where it is farthest removed from the carburetor entrance. It should be, certainly, around the corner from the straight passage at the carburetor entrance and should also be in a section where the aspect ratio is rather high. This latter requirement will provide for a minimum of aerodynamic loads on the heater valve since, for a given area, the center of pressure is at a minimum distance from the fulcrum point when a

In some designs of scoop that have been tested, it has been found that the major portion of the power loss experienced when going from full-cold to full-hot was due to constricted hot-air passages in the scoop, and that the major changes in fuel-air ratio in going from full-cold to full-hot were due to pressure disturbances caused by the heater valve itself and had little or nothing to do with the change in temperature of the carburetor air.

In both cast and sheet-metal scoop construction, it is usually possible with the expenditure of but little effort to provide a rounded entrance for the mouth of the heater valve. If this is done and the minimum area of the heater valve and passages kept to at least 75% of the carburetor entrance area, the power loss due to restriction in the heater passages will be kept to a minimum.

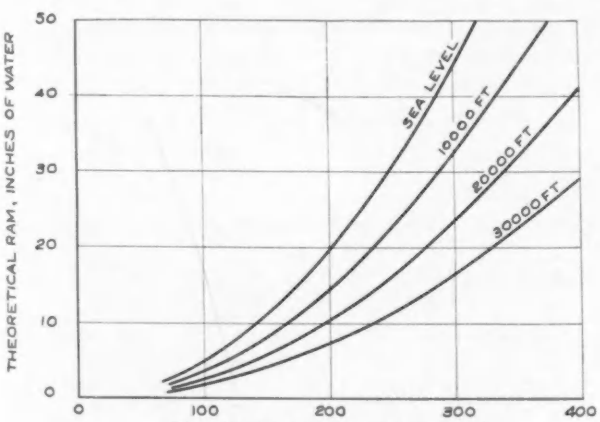
Rule IV: Keep the heater valve as far away from the carburetor as possible and locate it on the far side of the scoop elbow.

■ General Proportions of Passages

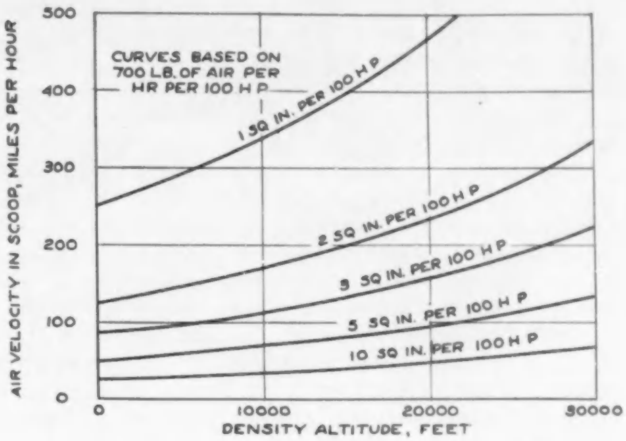
In the case of scoops which project above the cowl line, the shape of the scoop is usually pretty well established after the foregoing design considerations are followed. The airplane designer is often concerned about the drag set up by the projecting portion of the scoop and is anxious to

make this part as small as possible. A certain amount of area is required to provide proper breathing capacity for the engine and, unless some very special factors should indicate otherwise, the area of the mouth of the scoop should normally be kept within 75 to 100% of the area of the carburetor entrance, and the edges of the scoop should be rounded as much as other design considerations will permit.

In the case of under-cowl types of scoops, the situation is somewhat more complicated due to the increased length of scoop and the necessity for turning several corners in going over the cylinders and coming out under the lip of the cowl. Here again it is felt that the minimum cross-sectional area of the horizontal passage of the scoop should be from 75 to 100% of the area of the carburetor entrance. All of the changes in area should take place in the horizontal section of the scoop, the vertical section leading to the carburetor entrance being best left of a constant cross-section and shape. In most types of under-cowl scoops a fair amount of area is available at the mouth of the scoop to provide a good, smooth approach. Full advantage should be taken of all space at this point, since a good, rounded approach to the scoop will provide an appreciably higher entrance coefficient than is possible with a fairly sharp approach. Scoops which need to be tortuous in shape should be kept up to the maximum possible cross-sectional area at all points. Where changes in section are required, they should be made as gradually as space limitations will permit. This is particularly true where diverging sections are encountered in the direction of flow. If the area is increased too rapidly, the space will not fill anyway and will merely cause eddy currents and flow disturbances, with no corresponding benefits. The final vertical portion of the scoop leading to the carburetor entrance should be watched especially to prevent any rapidly diverging areas at this point. In general, a rapidly diverging area in the direction of flow causes loss of control of the flow lines, and eddy currents with resulting poor pressure distribution. Gradually converging areas in the direction of flow cause an acceleration of the flow through the passage with a corresponding straightening out of flow lines and an improvement in the pressure distribution.



■ Fig. 6 – Variation of theoretical ram with altitude



■ Fig. 7 – Variation of air velocity in scoop for varying scoop areas and varying altitudes

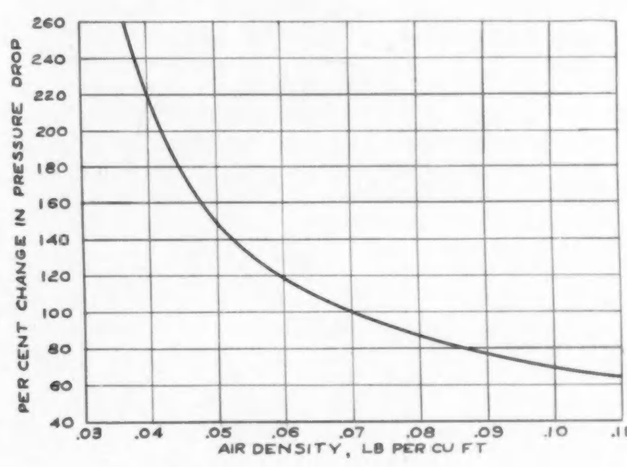
Rule V: Maintain at least 75% of the area of the carburetor entrance throughout the scoop and also at the heater valve. Avoid rapidly diverging sections, particularly just ahead of the carburetor entrance.

Fig. 4 shows a representative design of a projecting type scoop employing the design principles just given, and Fig. 5 shows a similar example of an under-cowl scoop.

■ Ram

There have been many pro-and-con discussions as to the advisability of providing carburetor ram. However, it is a fact that the use of a forward-facing airscoop, the mouth of which is in a region of fairly high pressure, will provide a substantial increase in the critical power rating of the engine by taking full advantage of ram.

The term "ram" is sometimes rather loosely used, and some definitions are in order. The term "theoretical ram" is used to denote the impact pressure of the air on coming



■ Fig. 8 – Variation of pressure drop with constant mass air flow through passage of constant area, for variations in air density

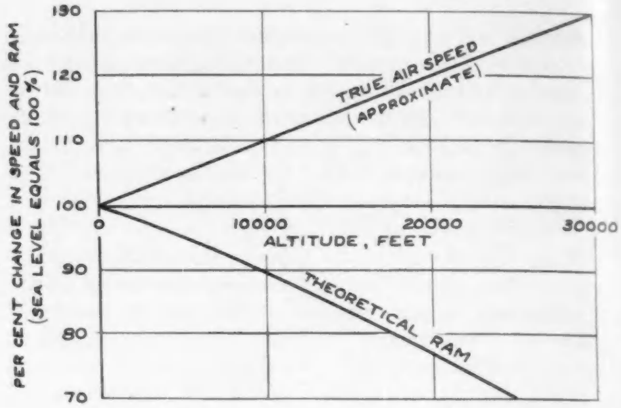
to rest. This would correspond to a pitot-static reading of the airplane air speed. This amount of ram is not actually available at the carburetor due to the air flow through the scoop and carburetor.

The term "ram" or "actual ram" may therefore be defined as the static head available at the carburetor entrance. This is equal to the theoretical ram minus the head corresponding to the velocity of air in the scoop proper. From this definition, it will be seen immediately that the ram is dependent upon a number of variables, among them being the true air speed of the airplane, the density of the air in which the airplane is flying (therefore the altitude of the airplane), the horsepower being used, the location of the mouth of the scoop with respect to surrounding pressures, and the position of the scoop heater valve.

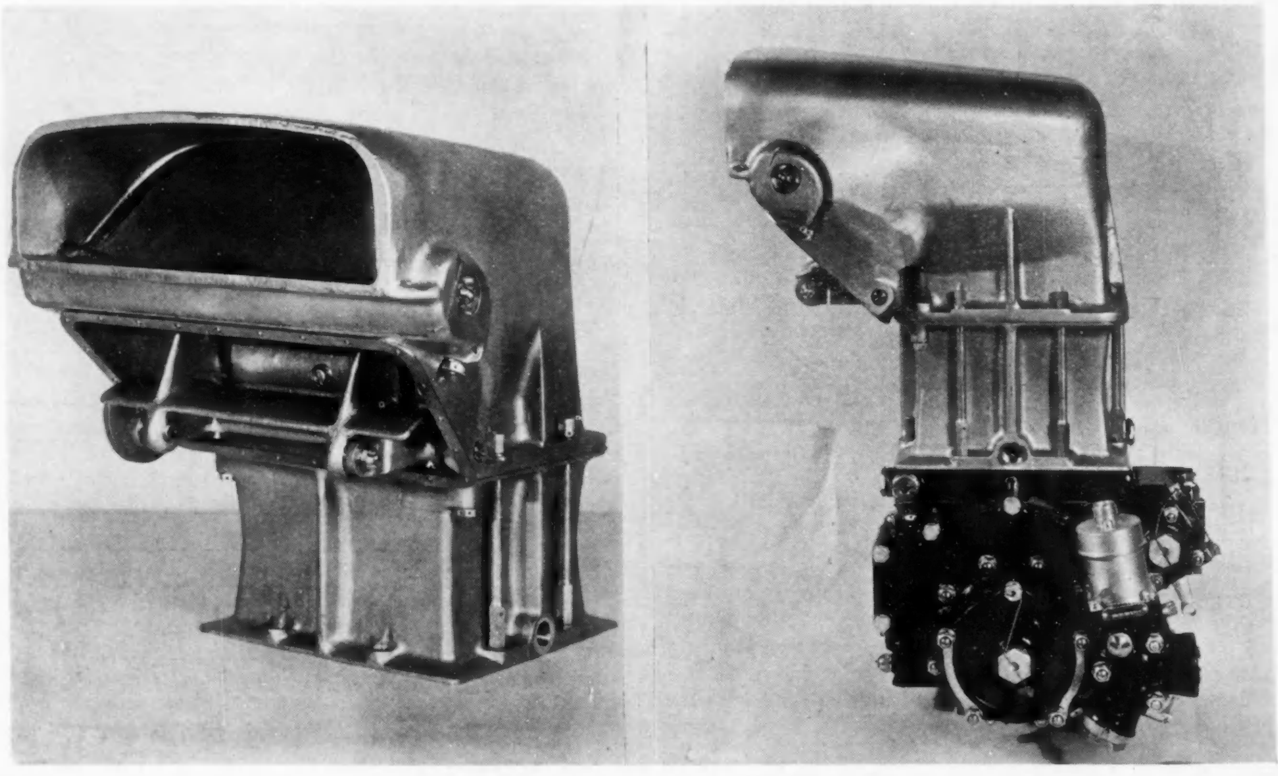
A study of some of these variables as plotted in Fig. 6 will show that the maximum theoretical ram occurs at sea-level conditions. Airplane-engine combinations which are designed to maintain a certain rated power value up to quite high altitudes will suffer particularly from loss of ram at these higher altitudes.

An engine will ordinarily require a constant number of pounds of air per horsepower, except that, in the case of engines operating with two-speed superchargers, the specific air consumption will be slightly higher when the high supercharger gear ratio is used. If we assume that a constant power is used at all altitudes under discussion, the air velocity through the airscoop will increase as the air density decreases, as shown in Fig. 7. This will cause an increase in the pressure loss through the scoop approximately as shown in Fig. 8. For a constant power, the true air speed of an airplane may be assumed to increase at the rate of 1% per 1000-ft change in altitude. This is not an absolutely accurate relationship, but strikes a fair mean, and it is believed may be used with reasonable accuracy for the purpose of this discussion. This increase in air speed would provide an increase in ram, for conditions of constant altitude, but this increase is more than offset by the loss in air density with altitude, with the result that the theoretical ram decreases quite rapidly although the true airspeed increases. This is shown in Fig. 9.

The chief ways in which the maximum amount of ram



■ Fig. 9 – Approximate variation of ram and true air speed with altitude for constant power



■ Fig. 10 – Three-quarter and side views of Holley airscoop

can be made available under high-altitude and high-power conditions are to be sure that the scoop provides the maximum possible amount of cross-sectional area and is located with respect to the nacelle, so as to be in the region of maximum impact pressure.

■ Design of a Practical Airscoop

The work on airscoop investigation which has been summarized in the foregoing remarks was begun during a siege of carburetor trouble when it became imperative that a satisfactory carburetor calibration be worked out, after the usual methods of attacking the problem had been tried and had failed. Preliminary tests showed that the airscoop was responsible for some of the difficulty and, in order to check off this point, an attempt was made to design an airscoop which would correct the carburetor metering trouble and incidentally provide as many as possible aerodynamic and structural advantages. Design limitations were laid down and everything possible was done to:

(1) Design the scoop in such a way that the carburetor would work properly for all conditions of operation.

(2) Provide for the maximum of mechanical reliability and ruggedness.

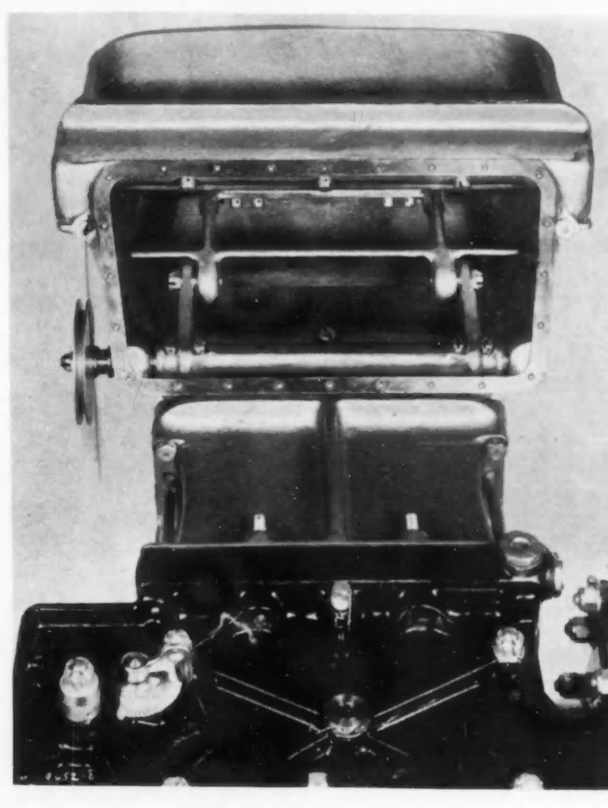
(3) Provide for easy removal of the scoop and easy maintenance.

The following description and discussion cover the type of airscoop which satisfactorily accomplished these objectives:

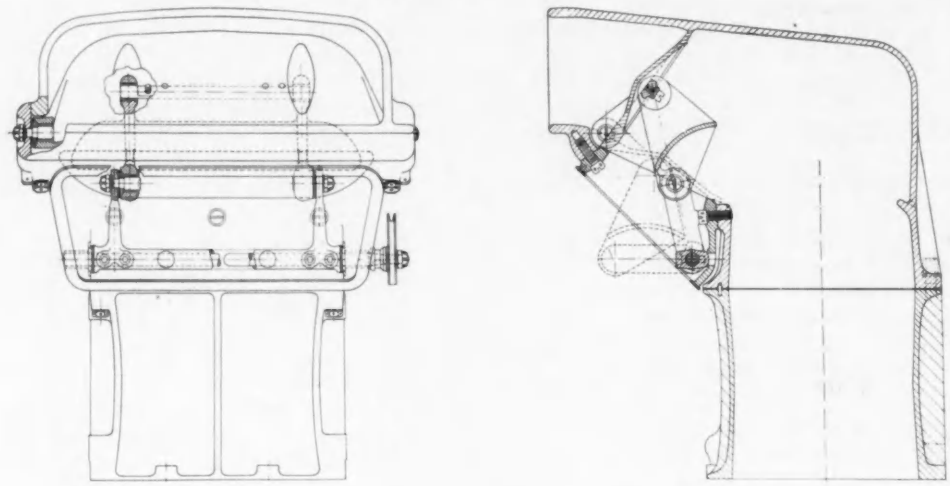
■ Description of Holley Airscoop

This particular airscoop was designed for a Douglas DC-3 installation using a Wright G-102 engine and a Holley Model 1375-H carburetor. It was hoped that it might be possible to design some sort of a universal airscoop and, for this reason, it was decided to make the elbow proper as compact as possible and provide the proper height for the scoop by adding a riser which could be varied easily to suit different installations. Since the scoop was designed primarily for airline operation, mechanical reliability was of prime importance, and therefore a majority of the parts of the scoop were made of magnesium castings of heavy section. Since a fair amount of height was necessary, which would mean quite a good deal of cantilever loading at the lower attaching flange of the scoop, it was decided to attach the scoop elbow, and the riser, to the carburetor by means of through studs instead of by using nuts and bolts at the parting surfaces. This permitted the straight riser to be made fairly light, since it was in compression and was subjected to very little stress. The straight riser was held rigidly to the top flange of the carburetor by means of the studs, which screwed onto the regular carburetor studs in the same manner as nuts and which projected through the upper flange of the riser. In order to prevent the necessity for removing any of the parts of the heater muff assembly when removing the scoop elbow, no attaching screws were provided on the front of the scoop riser, where they would be very difficult to reach after the installation was completed. The scoop elbow was therefore attached by six nuts around the sides and rear edge and two cap screws which are located at the extreme front corners. This construction is illustrated in Fig. 10.

The heater valve assembly includes several unique features. Since the only moving parts of the scoop are the heater valve and its control linkage, all of these parts were mounted in a single flange which dovetails interchangeably into the scoop elbow proper. The heater valve is operated by a strut-and-lever mechanism which provides for an over-center lock when the heater valve is in the full-hot position. The strut through which the heater valve is actuated is supplied with a built-in airfoil section which moves into position in the hot-air opening as the heater valve is opened and acts as a vane which helps to guide the air around the sharp turn which exists in the air passage when the heater valve is in the full-hot position. Figs. 11 and 12 show this construction. All shafts are steel, and all bushings are of $\frac{3}{8}$ in. inside diameter oilless type. Freedom from looseness is provided at the hinge points by means of steel thrust washers separated by means of steel wrinkle washers. The heat valve actuating shaft is equipped with a control pulley for use with cable controls and is symmetrical, so that it may be installed end-for-end to provide for right- and left-hand control installations. The lever casting which attaches to the control shaft is provided with a valve arrangement which allows for the escape of hot air when the heater valve is in the full-cold, shut-off position. As soon as the heater valve is opened, the hot-air by-pass valve begins to close and remains closed during all the time that the heater valve is turned on. This construction is illustrated in Fig. 12. The edge of the heater valve is contoured to match its seat in the scoop elbow so that the cold air passage can be shut off to provide a minimum of leakage. This is quite important in operation



■ Fig. 11 – Photograph of Holley airscoop showing heater-valve mechanism



■ Fig. 12 – Assembly drawing of Holley airscoop

through very severe rainstorms, when the quantity of water which is admitted to the engine via the carburetor may become a serious item. The mouth of the hot-air entrance is equipped with a stainless-steel rubbing plate which mates with a welt strip on the heater muff fitting. This provides a flexible, easily disassembled, fairly tight connection with provision for possible relative movement due to vibration. In an actual installation the only other item which needs to be removed in order to replace the scoop elbow is the strip of cowling which encloses the scoop proper. No parts of the heater muff system need to be disturbed in order to replace the scoop. The total weight of the unit as illustrated is approximately 12½ lb in cast magnesium.

■ Conclusion

Reference to the drawing and photographs of the Holley Scoop shows that it follows pretty well the design principles outlined earlier in this paper. Exhaustive laboratory and flight tests have demonstrated that the design satisfactorily does the things expected of it.

Most of the material in this paper may be summarized in the following five rules:

Rule I – Provide the maximum length of straight section of scoop immediately ahead of the carburetor.

Rule II – Use a sharp right angle elbow – not a smooth turn.

Rule III – Maintain a high aspect ratio in the scoop passages and elbow, in so far as possible.

Rule IV – Keep the heater valve as far away from the carburetor as possible and locate it on the far side of the scoop elbow.

Rule V – Maintain at least 75% of the area of the carburetor entrance throughout the scoop and also at the heater valve. Avoid rapidly diverging sections, particularly just ahead of the carburetor entrance.

It is not intended to convey the impression that the principles just summarized are the only ones which will give good results. Problems subject to the number of variables and limitations which enter into the design of an airscoop would be expected to have more than one satisfactory solution, and certainly a large number of unsatisfactory solutions.

The material in this paper is presented with the hope that it may be used as a basis for a more or less standardized design of airscoop for modern aircraft carburetors. An attempt on the part of the various airplane manufacturers to provide airscoops which have certain general characteristics in common would be of great help to everyone concerned with the problem of obtaining satisfactory carburetion, and should pay dividends by reducing the flight-test time required and by making possible better all-around engine performance and economy.

DISCUSSION

Effect of Slipstream on Carburetor Metering

– Frank C. Mock

Bendix Products Division, Bendix Aviation Corp.

FOR over a year we have been testing airscoops from various airplanes in our laboratory under conditions of steady air flow, reporting as to their effect upon carburetor metering. The effect was usually small, in the order of 2% or 3% rich or lean. Our findings usually would be reproduced when the scoop and carburetor were tested on an engine test stand; also, in cruise power in flight at any altitude. But, in many cases at high power at altitude, the flight fuel/air ratio would go definitely rich by as much as 10% to 15%, over the most carefully determined air-box or engine-stand altitude setting.

This result was so regular that our men learned to look for it, and it was so definitely confined to high power that they soon learned quickly to apply a jet change that would correct for it, so far as our limited flight tests were concerned. Such a correction would, of course, test lean on the ground on an engine test stand.

We do not find this in turbo installations, nor where the airscoop opening was well back from the propeller so we decided to demonstrate what the propeller slipstream could do to the metering by the following experiment:

We have a Cyclone G-100 engine mounted on a rigid stand with

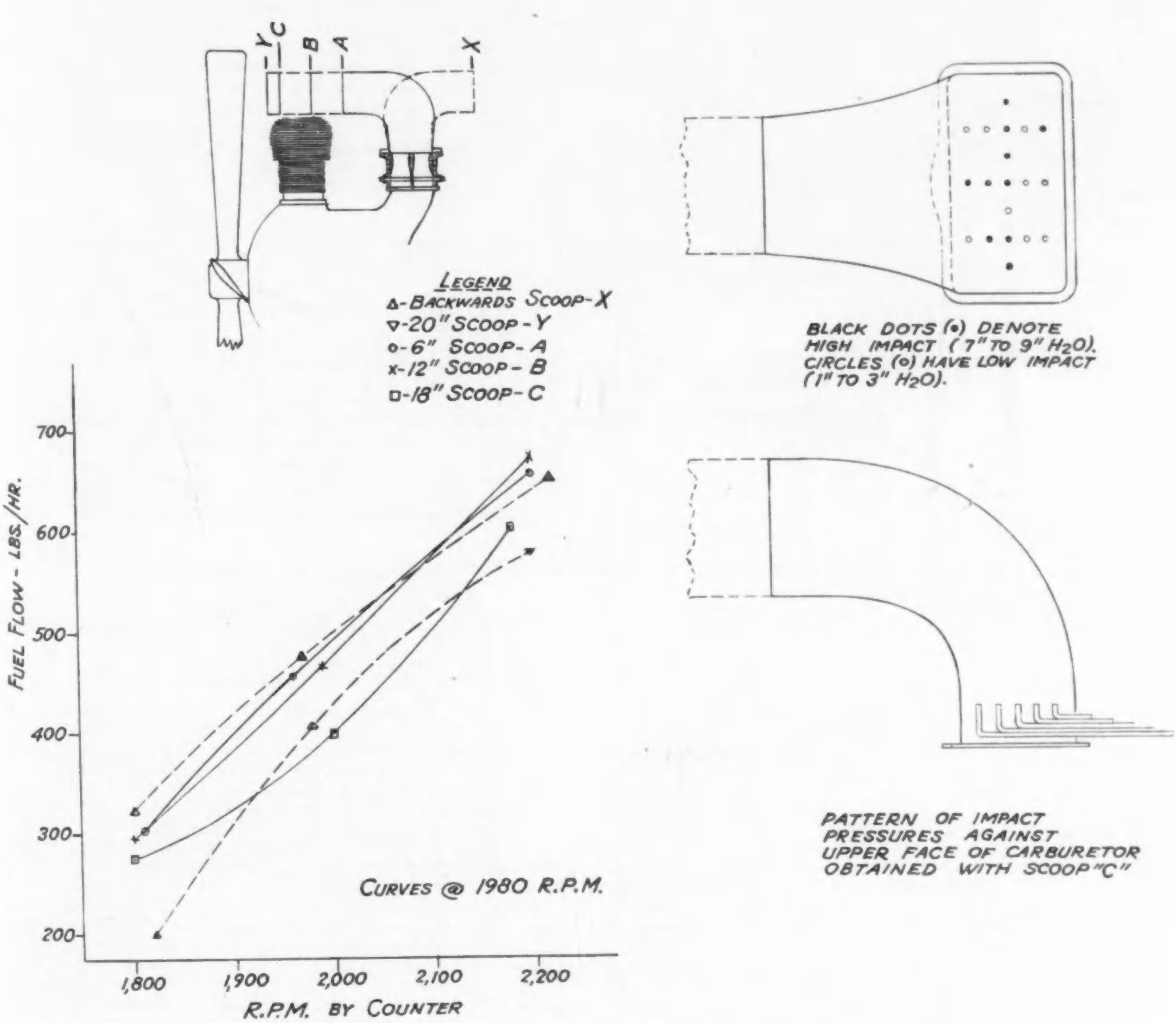
a wooden club, and on this we fitted a scoop elbow, with different length extensions toward the propeller club, as shown on the upper left-hand corner of Fig. A. The left-hand chart of Fig. A shows the effect of these different extensions upon the fuel delivery at various engine speeds with a four-barrel float-type carburetor, which was formerly standard equipment on this engine. It will be noted that there was a very serious variation in metering. Two other carburetor types which have been standard equipment on this engine also were tested. One had a little less than half the variation of the four-barrel; the other one was worse and would not run the engine at several points. It will be noted that we kept the same scoop elbow and only changed the distance of the scoop entrance from the propeller.

To find why the mixture proportions changed, we then placed a series of impact tubes, facing upward, across the face of the carburetor. With no scoop, or with a short elbow, the pressure pattern fore and aft was almost uniform, varying not more than 1 in. H_2O. With the longer scoop extensions a definitely discontinuous pattern developed, as shown on the right-hand section of Fig. A. An irregular squirt of air was developed with impact pressures as much as 8 in. H_2O higher impact than in the surrounding passage. Further measurements are being made to determine how and where this discontinuity develops with several different-shaped scoop elbows.

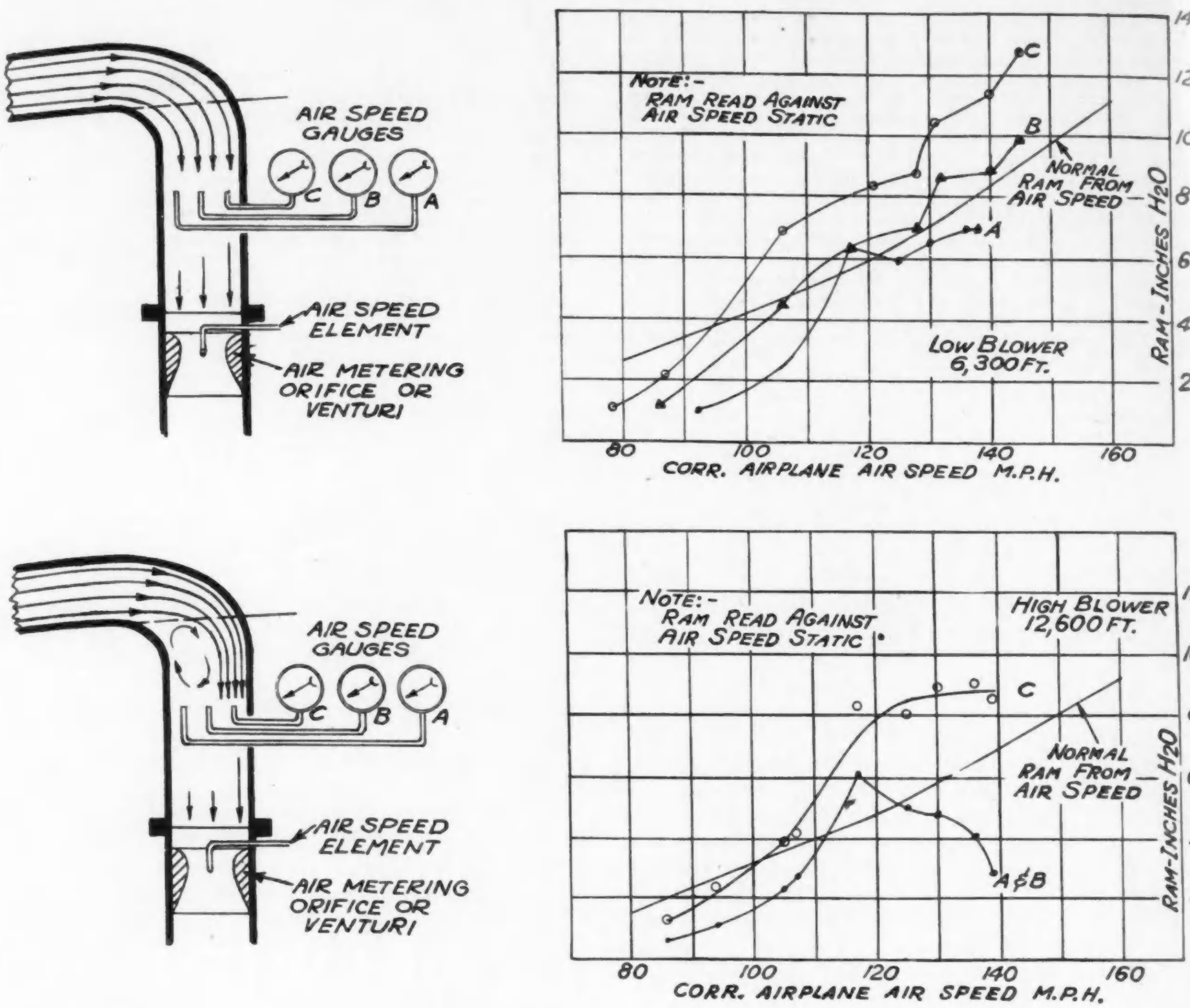
In basic terms, a carburetor meters air by means of an *air-speed metering element* located in an air stream whose area is predetermined by a *metering orifice* or *venturi tube*. If, or when, the air stream does not fill the orifice, the calibration will be incorrect. This will also be the case if the air stream velocity is non-uniform and the air speed metering element is not at a point of mean velocity. It was quite evidently the shift of air stream that disturbed the metering of the carburetor.

We have also had this change of air stream occur in flight, without change of scoop but according to increase of propeller slipstream. Some of our engine companies have also been investigating this subject, and one made a flight test with equipment as shown on Fig. B, in which impact tubes, ranged fore and aft in the scoop passage above the carburetor, were connected to air-speed gages balanced on air speed static pressure.

As shown on the upper right-hand chart of Fig. B, in low blower, about 6000-ft altitude, the impact pressures on the three tubes ran about parallel, the rear one having slightly the higher pressure. I



■ Fig. A (Mock discussion) – Effect of different length airscoop extensions upon fuel delivery and upon pressure pattern across carburetor



■ **Fig. B (Mock discussion) – Results of flight tests showing effect of shift of air stream on ram across carburetor**

have indicated the normal impact pressure due to air speed and it will be noted that these impact pressures followed it very closely.

The lower right-hand chart shows these same pressure readings in high blower at about 12,000-ft altitude. It will be noted that at the lower airplane speeds the impact pressures follow the indicated airplane speed, but that, at about 117 mph a change took place, the front and center impact pressures dropped, while the rear impact pressure rose toward a value which probably expressed the propeller slipstream impact speed so that the actual air flow probably was as shown on the lower left hand view and similar to that experienced with the long scoops in the test of Fig. A.

It will be recalled that it has been long practice to proportion the airscoop entrance to the carburetor for an air velocity of about 100 mph at rated power and to keep the intake passages beyond well below 200 mph in order to avoid loss of engine volumetric efficiency. It is not strange, therefore, that a highly turbulent slipstream of perhaps 275 mph or over, intermittent as the propeller blades pass the scoop opening, should generate this discontinuous flow.

Fig. C shows several ways in which this discontinuity may start. As shown in the upper figure, it may generate at the front edge of the cowl, particularly when this is blunt. We do know that the pressure gradient on an airscoop such as this will vary somewhat with the attitude of the ship, but we do not know under what conditions discontinuity begins.

The center and lower figures depict two sources of variation: at the scoop entrance, and at the duct outlet entering the elbow, of which the latter is the important factor from the standpoint of elbow design. We can do a pretty good job of straightening the air flow around the elbow if the pressure pattern at the entrance to the elbow is uniform: but an elbow design that will transmit a uniform pressure pattern from its entrance to its outlet will generally transmit and maintain any uneven pressure pattern which it receives. Hence, it is quite important that we have a uniform stream filling the opening at the entrance to the elbow.

I want to digress here to mention a phase of this subject that is more important than the engineering one. Our whole airplane procurement system is, at present, set up on the misconception that an engine and carburetor calibrated on the ground in still air will operate exactly the same in flight with the slipstream blowing into the scoop. We have an elaborate procedure for carburetor and engine testing, and have spent weeks trying to get a setting within ± 2% limits, but we have no provision in the program for flight-test determinations nor a regular procedure for flight check. Much effort, time, expense and friction have been wasted because we did not realize the existence of the physical phenomenon just described. The British have taken a more realistic viewpoint: in some cases they have simply moved the metering element of the carburetor over into the air stream coming from the far side of the scoop end and have

selected their jet setting accordingly; however, in this case, the scoop entrance was so far from the propeller that intermittent pulsations of pressure are unlikely.

The information just given was obtained so recently that the picture is still quite incomplete. However, the following recommendations seem clearly indicated:

1. In our talk we should de-emphasize discussion of elbow and duct shapes: instead, speak in terms of impact pressure measurements which will indicate the air flow. (See Fig. D.)

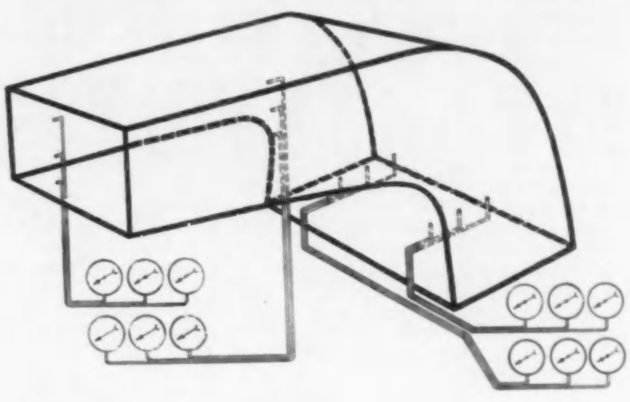
2. Our procurement program must be modified to plan for and appropriate funds for flight tests. For the first six months or so there will be considerable flight development work required. After that the flight tests should be shorter and mainly in the nature of check runs. Meanwhile we should make a fairly thorough flight test to see whether it is permissible to correct for this disturbance by selecting the carburetor jet setting for flight service rather than for ground-test requirements.

3. In planning our future approach on this problem it would seem that the airplane designer should be responsible for selection of the scoop entrance opening at a point where the pressure pattern on it will not change in character in various powers, altitudes, and attitudes. The airplane designer should also see that the duct should give a uniform pressure under all conditions at the elbow entrance.

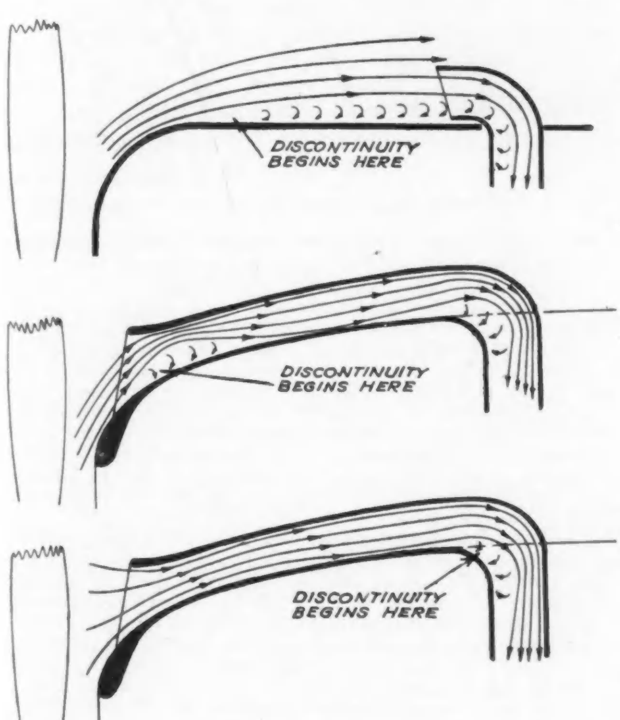
4. Given the foregoing, it should not be difficult to design an elbow that will give a fairly uniform pressure pattern at its outlet, to the carburetor face. Given a definite and uniform pressure pattern of entrance it will not be difficult to handle the carburetor; in fact, the present carburetors would operate properly without change from their ground setting. A converse of this procedure which has been requested, of building a carburetor that will meter consistently regardless of the nature and velocity of the air stream approaching it, does not now seem possible.

5. It is suggested that an effort be made to protect the airscoop entrance from intermittent disturbance by the propeller blades: if at the front of the cowl, by locating it at the blade roots and subtending a large arc of propeller rotation: or by locating it far back from the propeller, providing its pressure is not disturbed by ship attitude. It is felt that a localized intermittent air stream at slipstream velocity can be harmful in other ways besides its effect on carburetor metering: it stresses the cowl attachment, sometimes generates an annoying noise, may affect the fuel distribution in the manifold, and may initiate supercharger surge.

6. It may be worth while for the engine companies to follow the method of test suggested in Fig. A, as a means of ground check that will simulate flight-test conditions as regards scoop duct and elbow.



■ **Fig. D (Mock discussion) – Set-up for obtaining airscoop impact-pressure measurements which will indicate the air flow**



■ **Fig. C (Mock discussion) – Possible forms of scoop airflow disturbance due to slipstream in flight**

Factors Controlling Supercharger Induction

– J. E. Ellor

Chief Experimental Engineer, Rolls-Royce, Ltd.

IN the first place, I agree with the author that too little attention has been paid to airscoop design. The information contained in the paper should be very useful and helpful to the aircraft powerplant engineers. As pointed out by Mr. Kittler, an ideal scoop design might be accommodated in the present type of low-drag powerplants, but exceptionally good results can be attained by compromise, if the factors controlling the efficiency of the induction system to the supercharger are carefully considered. From my experience in the development of Rolls-Royce engines in powerplants, I would say that the three main factors are:

1. The mixture must enter the eye of the supercharger impeller through a passage angular in shape. It must have, as nearly as possible, uniform velocity of flow across this area for good cylinder distribution and there should be a minimum loss of pressure in the supercharger elbow.

2. The passage area through the throttle barrel and chokes of the carburetor must be maximum consistent with good metering control. The entrance shape to the chokes must conform to the known laws of streamline flow and the areas relatively large.

3. The aspect ratio at the supercharger impeller entrance is unity, and our research indicates that in order to get best net results from a complete supercharger inlet assembly, the unit aspect ratio must be departed from as little as possible.

I agree that the cascades or vanes are undesirable for reasons pointed out by Mr. Kittler, but abrupt right-angled turns reduce the energy-conversion efficiency, at the same time introducing an unknown variable in the distribution of flow at the inlet to the chokes. Is it not better to provide for the centrifugal build-up around the outer bend of the scoop and suitably adjust the position of the carburetor air pressure control orifices?

With regard to the scoop inlet parts, our experience shows that there is a definite advantage to be gained in keeping the area as small as possible, both from an overall efficiency point of view and

minimizing of trouble which may occur due to snow entry or ice formation. For the scoop entry area, we find a very satisfactory figure to be 20% in excess of the area calculated, based on the air supply to engine on a climb, assuming I.A.S. velocity at the entrance.

Regarding Mr. Kittler's reference to ram effect: We are able to obtain 90% of the total theoretical head at speeds of 400 mph at 24,000 ft. The velocity head in the passage of the carburetor entrance should not, in my opinion, be taken into account when assessing the total ram, as the engine calibration on the test bed has this already debited.

A further important item is the location of the scoop inlet in relation to the body. For best results, we find that the inner edge of the scoop should be about 1½ in. from the surface to eliminate the boundary-layer effect. The entry should be so shaped as to permit an efficient compression to take place in air stream forward of the scoop, at the same time permitting a streamline non-turbulent flow into the intake itself.

By compromising between scoop shape, carburetor entrance shape, and supercharger elbow, we were able to elevate the full-throttle altitude of the Merlin engine to the extent of 3000 ft without any change in dip speed.

I agree with Mr. Kittler's remarks on warm air admission but I am afraid time will not permit me to expand on this subject. There is one item which has not been referred to by Mr. Kittler, namely, the incorporation of air cleaners with air inlet scoops. For desert or sandy air fields, these cleaners are very necessary to protect the engine from wear and entail careful consideration in scoop design in order to reduce power loss to a minimum.

Comparative Airscoop Data Requested

– A. L. Beall

Research Engineer, Wright Aeronautical Corp.

As Mr. Kittler points out, there is little known on the subject of air-scoop design and its relation to other hydrodynamic problems. He has done an excellent job in reducing the various elements of the problem to rational concepts.

For the individual or organization interested in using the scoop described in Fig. 12, however, a comparison of the unit recommended with other scoops, in service or tested, would be helpful. Data on the actual airflow in pounds per minute under comparable conditions of operation for the recommended and other scoops would save time in the selection of a suitable scoop or the comparison of the merits of several scoops. In brief, the paper could be amplified very materially to increase its usefulness in connection with engines while the data as now shown are most useful to the aircraft manufacturer installing the scoop.

With every effort to insure smooth flow through the scoop, it is doubtful if the airflow at the entrance of the scoop can be expected to be smooth in all types of air and in all flight attitudes. Consequently, carburetor performance tests associated with scoops should contemplate turbulent air at the scoop entrance.

The British type of ice baffle ahead of the scoop entrance might help to insure non-turbulent air in the scoop.

Status of Automatic Carburetion

– Robert E. Johnson

Chief Field Engineer, Wright Aeronautical Corp.

Mr. KITTLER has developed very practical design standards in approaching this very serious airscoop problem, and the progress made to date by his and other companies is very commendable. However, the industry in general must realize the seriousness of the situation and the price it must pay for accurate automatic carburetion.

The industry must recognize the following fundamental facts which exist in the present state of the art:

1. There exist no conclusive design data which will guarantee the realization of desired carburetor metering under all conditions of airplane flight.

2. There exists no conclusive ground-test procedure or equipment by which an airscoop can be tested and guaranteed to realize desired carburetor metering under all conditions of airplane flight.

In view of the foregoing, it is essential that ample flight time be scheduled in each new aircraft engine-carburetor scoop combination to assure that the desired carburetor metering is obtained under all flight conditions. It will be necessary for this procedure to continue until experience has shown that the problems can be correlated between airplane and ground test methods. The carburetor manufacturers and engine manufacturers are cooperating toward this goal, but there is much progress yet to be made.

Aerodynamics of Inlet-Plane Combination

– Abe Silverstein

Aeronautical Engineer, National Advisory Committee for Aeronautics

This paper is an interesting discussion of an important subject. The emphasis is largely on the aerodynamics of the carburetor duct design, and it is believed that, in addition, some consideration should be given to the aerodynamics of the combination of the carburetor inlet and the airplane. Mr. Kittler has given some excellent rules to insure uniform flow in carburetor ducts providing that the velocity distribution at the carburetor duct inlet is uniform. In some cases, particularly when the carburetor scoop is located near the rear of the nacelle, the existence of the nacelle boundary layer may disturb the symmetry of the inlet airflow distribution. This effect is particularly pronounced at angles of attack in which a large adverse pressure gradient exists along the upper surface of the nacelle and, in some cases, airflow separation may actually exist just ahead of the carburetor scoop inlet. Any dissymmetry of air flow at the carburetor inlet would probably be exaggerated within the duct and lead to the type of difficulties that are now experienced. For this reason, the carburetor inlet type shown in Fig. 5 of Mr. Kittler's paper is believed to be somewhat more favorable than that shown in Fig. 4.

The requirements of high carburetor ram pressure and uniform velocity distribution at the carburetor venturis are not contradictory, and it is believed that, by the use of satisfactory turning vanes in the duct corners and carburetor inlets at the stagnation point, satisfactory carburetion can be achieved over the entire flight range.

Factors Influencing Cross-Sectional Area

– David Biermann

Aeronautical Engineer, National Advisory Committee for Aeronautics

This paper is very interesting. It brings to the foreground many of the problems encountered in designing carburetor rams which will be satisfactory from all standpoints. Although Mr. Kittler has discussed some of the aerodynamic problems encountered in taking air into an entrance and piping it to the carburetor, apparently his main concern is with the proper metering of the carburetor. His solution to the difficult problem of obtaining a good pressure distribution at the entrance to the carburetor appears to lie in creating sufficient turbulence at the bend in the ram so that a great portion of the velocity head is dissipated, leaving only the static pressure, which obviously will be uniform across the duct. Turbulence is also an effective way of mixing hot and cold air, and may be the only way, but it is nevertheless a rather expensive way from the standpoint of ram pressure.

Mr. Kittler recommends a cross-sectional area of at least 75% of the carburetor entrance. Although it is desirable to have a large duct with slow-moving air, other requirements may dictate a different answer. For example, we know that the entrance area to a duct must be of such a size that the entrance velocities bear a certain relationship to the forward velocities of the airplane in order that the flow entering will be stable. Too large an entrance will result in air entering one side of the duct and spilling out the other side, thereby resulting in a high drag and a loss of ram. The presence of a boundary layer on the surface of the cowling plays an important part in this problem of maintaining stable entrance flow conditions. It is believed that the best solution, from the standpoint of stable entrance conditions and reduced velocities at the bend of the duct, lies in the use of a ram which extends to the front lip of the cowling. A small entrance is thereby possible, which fulfills the stable entrance conditions, and a sufficiently long pipe is available for efficient expanding of the air before reaching the heater valve and bend in the pipe.

Is It Practical to STREAMLINE for FUEL ECONOMY?

by JAMES C. ZEDER
Chief Engineer, Chrysler Corp.

EVERYBODY knows that you can reduce the power required to drive an automobile through the air by putting a streamlined body on it. Many people think that the advantages to be derived are important only at high speed, and that the main reason for streamlining a car is to make it go faster. Those who do think of streamlining as a penny saver rather than a performance booster may not fully appreciate the fact that the economies made possible by a reduction in air resistance can be largely nullified by failure to adapt the engine and transmission system to the altered road-load requirements of the vehicle. It is practical to streamline for fuel economy if, and only if, the power system is engineered to complement the body.

THIS paper develops the theme that streamlining is practical if, and only if, the power-supply system is engineered to fit the body. The author shows that the reduction in road-load power required brought about by streamlining may be accompanied by a less favorable engine specific fuel rate unless the transmission system is altered to prevent this result. The quantitative importance of this efficiency change is brought out, and the potential gains are set forth.

The present state and future limitations of streamlining as a source of economy are treated, and test-car results are quoted. The economies to be realized are stated to come from the lower horsepower required to maintain a given engine speed.

The importance of car weight and cross-sectional area are brought out, as well as the necessity for maintaining the accelerating ability of the streamlined car along with its improved economy.

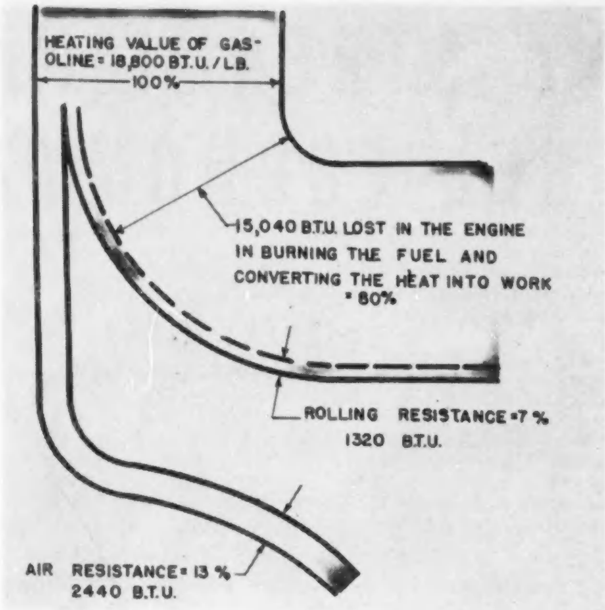
The last decade has seen a gradual trend established toward lowered air resistance through modifications in body and sheet-metal design. Every change in this direction increases the practicability of streamlining for economy by making less drastic the changes necessary to transform the "conventional" into the "streamline." We have taken a good many little steps along this path. They have had to be little, for in the past all major attempts at speeding up the progress of streamlining have met with the determined opposition of competitors, and tardy public acceptance. The frame of mind which once made it necessary to equip horseless carriages with whip sockets is still present, and a force to be reckoned with. It will tolerate no change too rapid for its limited powers of assimilation. Sun visors retreated within the body ten years ago. Windshields began to tilt, and have acquired more rake year by year. The spare wheel was first boxed in; then the luggage rack was thrown away and bodies were designed to accommodate both the spare wheel and baggage space. Fenders gradually evolved from wind slicers to guiding surfaces for the air stream. They are becoming virtually part of the body. Headlamps developed streamlined tails and finally disappeared into the fenders. Bumpers are being admitted from associate to full membership with the car. The tendency to widen the front seat is placing the maximum diameter of the car farther forward, which is in accord with good streamlining principles.

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 4, 1941.]

I – Heat Balance

So much has been written about streamlining that a good deal of familiar ground has to be covered nowadays to make a paper dealing with any phase of the subject coherent. In order to increase fuel economy through streamlining, for example, we must first see what we have to work with.

Every pound of gasoline that goes down the gasoline tank spout is capable of giving up almost 19,000 Btu of heat energy. We see in Fig. 1 what a typical 1941 sedan traveling 60 mph does with all that energy. First of all the engine takes up the big job of changing heating value into horsepower, and finds the going so tough that only 20 Btu in every 100 show up at the flywheel as pound-feet and rpm. We still have to get those 20 Btu down to the road surface before they can bore the holes through the air for the car to pass through. By the time they get there 7 more Btu are missing, having been expended in warming up the



■ Fig. 1 – Heat balance at 60 mph – 1941 model sedan

transmission, the rear axle, and the tires. Out of the original 100 only 13 Btu ever get to the place where they can push the car through the air. Any saving which we are going to make must come from our operation on this 13% of the energy we bought at the filling stations. At first glance we seem to be dealing in pretty small potatoes. If we are fortunate enough to be able to do a streamlining job that will cut the air resistance in half it looks as if we stand to save 6½%. That is about 1 mpg in 16. If the job is to be worth doing, we must get into that lost battalion of 80 Btu that never reached the flywheel and save some of them. That is exactly what happens. We are trying to move the car down the road. The wheels must roll and the car must push aside the air. The 20 Btu rolled the wheels and pushed the car down the road a certain distance. With a streamlined body of half the former resistance, 13½ Btu will move it that same distance, and for every one of them four will be lost as before, *provided the engine efficiency can be kept unchanged.* Altogether, then, we must put five times 13½ or 67½ Btu into the tank of the streamlined car and 100 Btu into the conventional car to travel the same distance. Taking it the other way, the hypothetical streamlined car would travel half again as far as the conventional on a gallon of gasoline. The fuel economy, therefore, has been increased 50% for a 50% reduction in air resistance. The potatoes aren't so small after all.

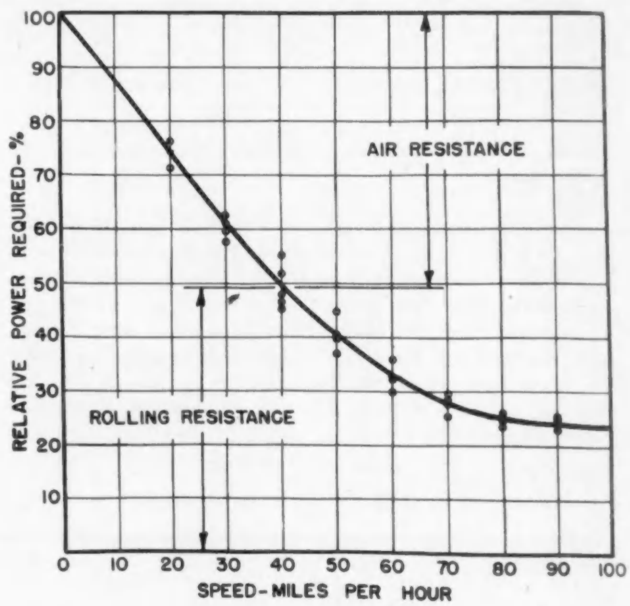
	CONVENTIONAL CAR	STREAMLINED CAR WITH 50% LESS AIR RESISTANCE
ROLLING RESISTANCE	1320 B.T.U.	1320 B.T.U.
AIR RESISTANCE	2440 B.T.U.	1220 B.T.U.
TOTAL RESISTANCE	3760 B.T.U.	2540 B.T.U.
ENGINE LOSS FOR 20% THERMAL EFF.	15040 B.T.U.	10160 B.T.U.
TOTAL	18800 B.T.U.	12700 B.T.U.
WEIGHT OF FUEL REQUIRED	1.00 LB.	0.67 LB.
MILES PER POUND	2.58	3.87
MILES PER GALLON	16.0	24.0

■ Fig. 2 – Economy comparison on Btu basis

Fig. 2 shows the energy comparison on the basis of the pound of fuel of Fig. 1.

II – Power Required

Fig. 3 shows the relative importance of air resistance and rolling resistance throughout the speed range. The circles mark test values obtained on 1941 model cars of different makes. We generally think of air resistance as being of importance only at high speed, but the chart shows that, even at 30 mph, the air resistance of a conventional, 1941 model car amounts to about 40% of the total power required. Again assuming that the engine thermal efficiency can be maintained, reducing the air resistance 50% would reduce the power required by about 20%, which would mean a 25% improvement in fuel economy.



■ Fig. 3 – Varying proportions of air and rolling resistance with speed

Fig. 4 is the familiar curve sheet showing test curves of rolling resistance, total resistance, and engine brake horsepower all plotted against car speed. The difference between any corresponding pair of ordinates of the total and rolling-resistance curves represents the air resistance. The difference between corresponding ordinates of the engine bhp and total-resistance curves represents power available for acceleration. This quantity becomes zero where the total resistance and bhp curves cross, and the speed at which this intersection occurs will, therefore, be the top speed of the car.

Fig. 5 shows the lowering of the total resistance of a

test car to which streamlining was applied. Incidental to the lowered resistance, the original axle ratio of 3.89 increased the power available for acceleration and top speed. Changing the effective axle ratio to 2.74:1 reduced engine speeds and cut down the engine friction accordingly. This was an important step in realizing the gains in economy made possible by the lowered air resistance. Except for the general conclusion concerning the beneficial effect of reducing the engine speed and friction, it will be seen that Fig. 5 gives us no guarantee of improved road load economy for the range, say, from 40 to 80 mph in which the engine is operating at part throttle. We must consult the part-throttle specific fuel curves to make sure we aren't getting less horsepower for more gasoline.

III – Air Resistance

First, however, the subject of air resistance deserves further attention. The aerodynamic drag coefficient K has become the popular measure of streamline body design. The usual formulas for drag and air resistance horsepower are:

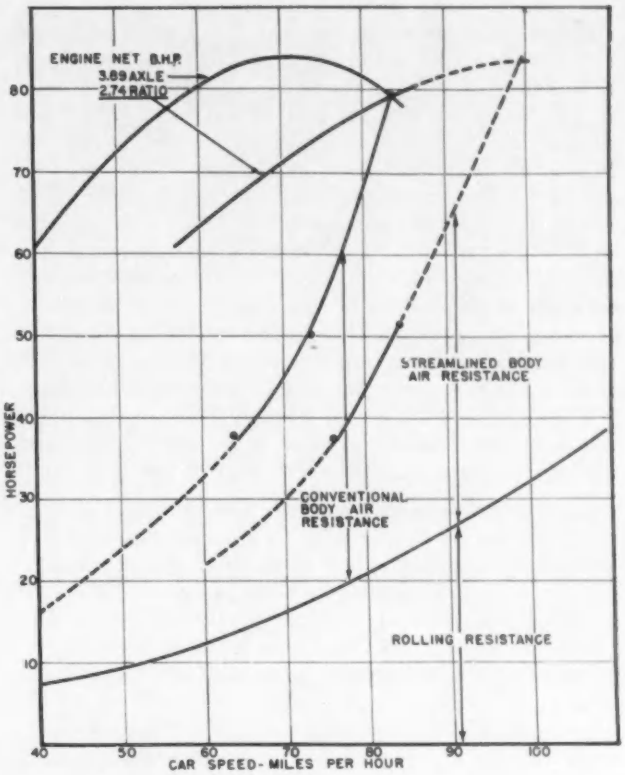
$$D = KAV^2$$

and

$$P = \frac{KAV^3}{375}$$

where K = Aerodynamic drag coefficient, lb sq ft (mph)^2
A = Frontal area of the car, sq ft
V = Car speed relative to the air, mph

An idea of the position of our present cars between the



■ Fig. 5 – Power and resistance curves for conventional and streamlined bodies

■ Fig. 4 – Curves of power available and power required

limiting extremes of air resistance is given by Fig. 6. The traditional flat plate provides the upper boundary of $K = 0.00320$, and the streamlined body of revolution sets a minimum in the neighborhood of $K = 0.00008$. In the early days, automobile designers used the flat-plate coefficient in calculating wind resistance on the few occasions when they considered it at all. By 1932 the K value had got down to about 0.0017. Current models fall in the range $K = 0.00105$ to 0.00120. A DeSoto Airflow test car to which a series of streamlining refinements were added, most important of which was a long tail, attained a coefficient of $K = 0.00061$, the average of tests made at 76 and 85 mph.

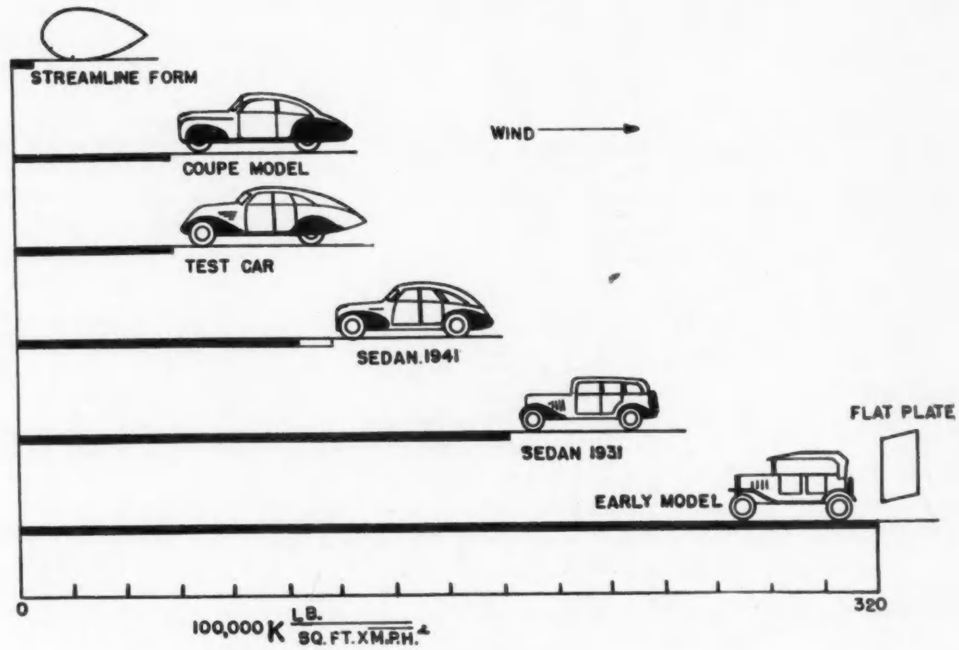
One of the biggest obstacles to efficient streamlining of a car body is the necessity for carrying passengers in it. The more passengers that have to be carried, the greater the compromises which must be made between the streamlined and the practical. Unfortunately, the human body has never developed streamlining. The man who buys the car is going to insist on emulating the flat plate and facing broadside into the wind. Moreover, he insists on disposing the upper half of his length pretty much vertically in the body whether he sits in the front or the back seat, so the sedan body cannot be tapered in the best streamlined fashion either in profile or in plan view without adding a good deal of length to the car. The design of the Airflow car was built around the then-new idea of a car which could carry six persons in comfort. In the face of the handicap of this increased passenger space, wind-tunnel model tests showed the DeSoto Airflow sedan to have only 80% of the total drag of a contemporaneous conventional model. It was because of the limitations imposed by the

enlarged passenger space, however, that the overall length of the Airflow car used in the streamlining tests previously mentioned had to be increased by 44 in. (from 179 to 223 in.) to obtain the tapering rear section necessary for best streamlining results. By making the results obtained on this car the basis for a series of wind-tunnel studies, a car design has been developed which, according to the wind-tunnel results, should approach the K value of the modified Airflow with no increase of overall length over 1941 production models. This design is still "in the mill" so that no road-test information is available. It can be said, however, that the car is of the coupe type which has the advantage of carrying fewer passengers and is therefore capable of more complete development of streamlining principles.

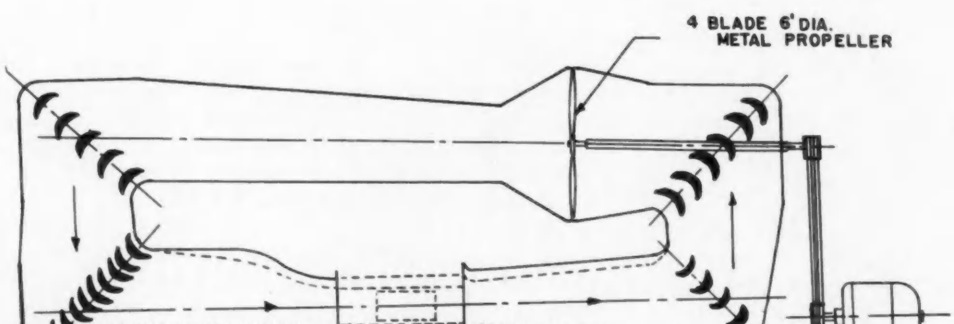
The Chrysler wind tunnel (Fig. 7) in which this design was developed measures 30 in. by 45 in. in the throat cross-section and has a maximum air speed of 120 mph. It is regularly used as a means of comparing the aerodynamic characteristics of new car designs with those of production models on which corresponding road-test results are available. Thus it points the way for future design, enabling the engineer to test a wide variety of designs quickly and economically.

The tests are run on wooden models scaled down to 1/10 full size. All small projections, such as radiator ornaments, door handles, and the like are omitted because the drag they produce is greatly magnified on the small scale model. A 1/10 scale model tested at 50 mph has a Reynolds' number of about 770,000 while the full-size car at the same speed will have a number in the neighborhood of 7,700,000.[1] It would be necessary to make the test at 500 mph, or to increase the air density to compensate for this scale effect. It is sufficient for comparison purposes to omit the small projections, smooth the bottom surfaces of the models, and correct the lowered drag figures so obtained

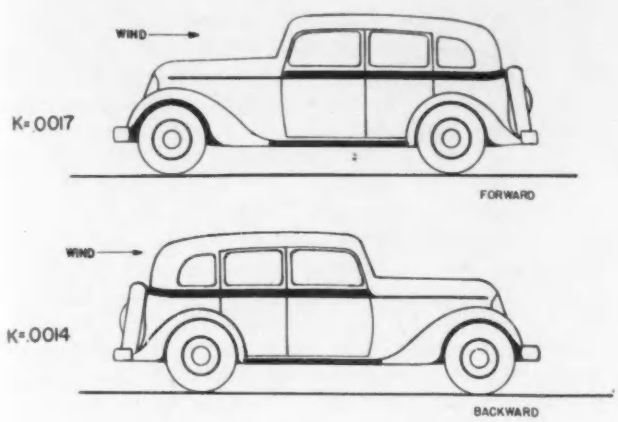
[1] Based on 200 in. overall length. See Appendix.



■ Fig. 6 – Comparative resistance coefficients



■ Fig. 7 – Chrysler 30 x 45-in. wind tunnel



■ Fig. 8 – Car model reversed in wind tunnel

by correlation with road-test results on models for which these are known.

Road Tests – The road tests of air resistance to which the wind-tunnel work must be correlated are of two types:

(1) Speed runs made with a series of different throttle stops, followed by measurement on the chassis dynamometer of the power available with the same throttle stops and the same speeds.

(2) Direct measurement of road-load torque at various speeds by means of a torque indicator built into the transmission, followed by friction horsepower measurements at these speeds on the chassis dynamometer.

The first of the foregoing methods is the one which has been in most general use to date. With this method, the chassis dynamometer measures the power supplied by the engine (at the road test speed and throttle setting) in excess of that required to drive the car on the rolls. Thus the dynamometer measures directly the horsepower available for overcoming air resistance.

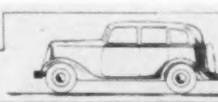
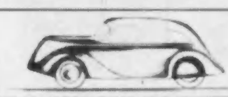
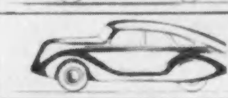
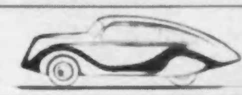
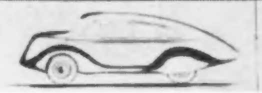
The torque indicator has been developed only recently. It measures the torque supplied to the propeller shaft while the car is being driven on the road. This torque includes air resistance, tire rolling resistance, and rear-axle universal-joint friction. The power required to drive the car on the rolls with the clutch released is measured on the chassis dynamometer and subtracted from the corresponding torque-meter results to obtain the net figure of air resistance. Either method may be used to check the other. The deceleration method is also used as a check when weather conditions permit.

IV – Tests of Detail Changes

The observation has sometimes been made that the conventional automobile with the engine hood projecting forward from a box-like body is, aerodynamically speaking, traveling backwards, since its greatest bulk is to the rear and what taper it displays is toward the front. Wind-tunnel tests of a model of a 1933 conventional sedan showed that the drag was decreased about 20% when the model was placed with its back into the wind. Fig. 8 shows the outline of this car and the *K* values obtained.

The results obtained on a DeSoto Airflow test car in which a series of detail changes were made to reduce air resistance already have been referred to. Fig. 9 gives a condensed survey of the changes made and tested in this program. A contemporary car of conventional design is included at the head of the table to show the true baseline of the comparison. The sum total of the changes made on the test car reduced its air resistance at top speed by 44% and increased its economy at 80 mph, 57%. The intermediate test conditions are listed in the figure in the order in which the tests were made. The effect of a particular change might, of course, be modified in combination with

■ Fig. 9 – Studies in streamlining

	TEST CONDITION		FUEL ECON. M.P.G.	AT M.P.H.	TOP SPEED M.P.H.	DRAG. COEF. K LB./SQ.FT.(M.P.H)²
	1933 CONVENTIONAL SEDAN		9.5	76	76.0	.00170
1	DE SOTO AIRFLOW		11.3	80	84.0	.00140
2	BUMPERS REMOVED FLUSH WINDOWS & REAR WHEEL SHIELDS ADDED				86.3	.00128
3	NEW FRONT ADDED				88.0	.00120
4	FRONT SECTION OF TAIL ADDED				91.1	.00107
5	UNDERPAN & SILL FAIRING ADDED				94.4	.00094
6	REAR SECTION OF TAIL ADDED				98.8	.00081
7	FAIRED WINDSHIELD ADDED		17.7	80	99.4	.00079

other parts. Direct fuel economy figures are available only for the initial and final conditions of the series, but the effect of individual items may be estimated from their effect on the top speed of the car. The corresponding tabulation of K values is based on these top speeds and the power available at each speed, estimated from the 2.74:1 ratio power curve at the test speed, of 99.4 mph.

It will be noted that the K values listed for these top speeds are higher than the values obtained from the lower speed test points. This is the result of the departure of the test results from the cubic curve assumed in the formula:

$$\text{Hp required} = \frac{KAV^3}{375}$$

V – Design Considerations

Improved streamlining of the body tends to move forward the center of pressure of a side wind on the car. In some cases the change may be sufficient to produce directional instability at high speed. The addition of a rear fin would bring the center of pressure back to a more favorable location, but the designer would probably choose to compromise his design to avoid the condition rather than risk offending a public taste not yet educated to the acceptance of the fin idea.

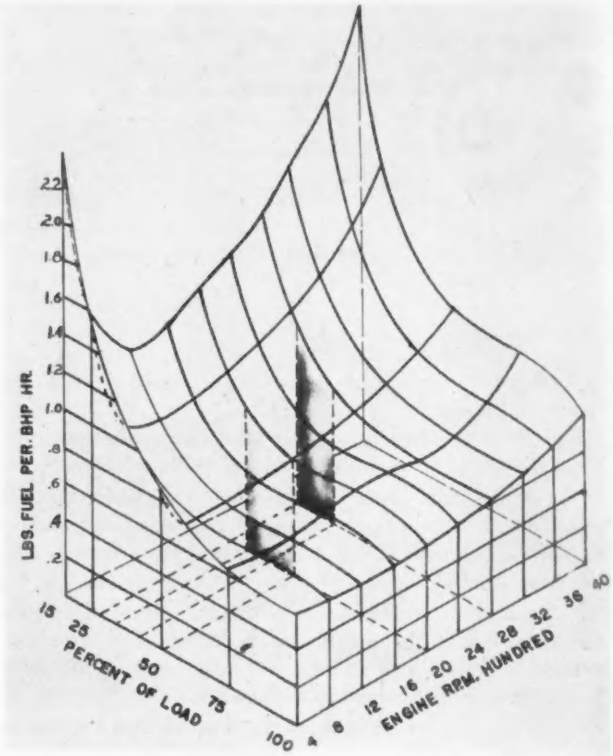
The economies to be realized by streamlining come from the lower horsepower required to maintain any given speed. Power required to accelerate is not affected directly since, for constant acceleration, it is the product of the average speed during the accelerating period, the mass of the loaded car, and the acceleration.[2] A streamlined design involving increased weight, therefore, might easily fail to show the expected improvement in tank mileage because of additional power required to handle the greater mass during periods of acceleration. Any weight saving which can be obtained in a new design will be well worth while. In any case, the designer should be certain that his streamlined body design is no heavier than the body it replaces.

VI – Engineering the Power System

Assuming now that we have succeeded in designing a car of the same weight as the present conventional model, and with 30% to 40% lower wind resistance, due to improved streamline form, we have yet to supply it with a satisfactory engine and drive train.

If we simply switched bodies on the present conventional car, leaving the engine, transmission and axle ratio unchanged, the lowered power requirements of the streamlined body would: (1) increase the accelerating ability of the car, and (2) reduce the percentage of full load under which the engine would operate at any given cruising speed.

Fig. 10 recalls the familiar fact that specific fuel rate is not constant either for load or speed changes. What this meant in the streamlining tests of the Airflow can be seen from Fig. 11. At 60 mph the total power required was reduced about 1/3 by streamlining but, with the 3.89 axle (engine speed 2800 rpm) the engine dropped from 41% to 27% of full load, with an increase in specific fuel rate from 0.74 to 0.93 lb per bhp-hr. The reduction in total fuel used per hour, therefore, was only 4 lb in over 24 for an economy gain of only about 20%. The same analysis carried through for the 2.74:1 drive ratio shows a greater economy gain, namely, 26%. The increased saving is due to the fact that the full-throttle power is decreased by the 2.74 ratio, since the engine speed at 60 mph has dropped to 1970 from 2800. The power required is a greater percentage of full-load power, and the unfavorable change in the specific fuel rate with streamlining is reduced. The load conditions are



■ Fig. 10 – Specific fuel consumption versus speed and load

such[3] that the fuel rate for the streamlined body with the 2.74 drive is the same as that for the conventional body with the 3.89 drive. Hence the combined effect of changing the body and the drive ratio is to cut the air resistance in half *and maintain the thermal efficiency unchanged,* and the increase in economy is about 50%.

It will be noted that changing the drive ratio to 2.74 without streamlining makes the same improvement in economy as streamlining without changing the ratio. The horsepower available for acceleration at 60 mph is reduced, however, from 47 to 31, a loss of about 1/3, and the proportionate loss in activity is too great to be practical except in combination with a transmission having a lower gear instantly available when high acceleration is needed.

The acceleration horsepower at 60 mph with the 2.74 ratio and the streamlined body was 42 as compared with 47 for the original car. At lower speeds, where the gain due to lowered air resistance is a smaller percentage of the power required, the reduction in activity is more pronounced. It is important, therefore, that a streamlined car be provided not only with a top gear ratio which will realize the economy gains possible at open road cruising speeds, but also with an accelerating ratio available in the medium- and low-speed range without special manipulation on the part of the driver.

[2] See Appendix for explicit derivation.

Conclusion

The function of streamlining, then, is to reduce the air resistance and hence the power required to drive the car. To translate this change into increased fuel economy requires: (1) that the weight of the car be kept down to normal, and (2) that a transmission be used which maintains the engine thermal efficiency at a satisfactory level in the high-speed range and yet does not reduce car activity at lower speeds.

Certain factors now developing should accelerate the trend toward streamlining in the coming decade. The need for military roads will stimulate the construction of through highways adapted to faster traffic. The competition of faster trains, faster buses, and increasing volume of air travel will force the automobile to higher cruising speeds. At the same time, with battleships, airplanes, and tanks to pay for, "miles per gallon" is going to be more and more important. The individual driver will be cutting down on all his living expenses in order to pay his income tax. He is going to insist on getting the economy he has been bragging about to his friends.

The effect of improved streamlining in practice will depend on, first, what the designer does with it and, second, how the driver uses it. If the designer takes firm hold on a low K value and hangs a greatly increased cross-sectional area around its neck, the driver may be able to stretch out full length for a nap on the front seat, but he won't have much to rejoice over in miles per gallon. On the other hand, if the designer, awake to the signs of the times, turns out a job with a minimum KA value, and the proper correlation of engine, transmission, and rear axle, the economies achieved will lie in the hands of the driver. He can continue to cruise at his accustomed speed of say, 60 or 65 mph and enjoy the benefit of 22 mpg instead of 16 or 18, or he can fall under the spell of high speeds attained with seemingly little effort, increase his cruising speed to 75 or 80 mph and buy his miles at the same old figure. No matter which he does, we may be sure he is going to like the change.

AXLE RATIO	3.89		2.74	
CRUISING SPEED	60		60	
ENGINE SPEED	2800		1970	
FULL LOAD H.P.	80		64	
	CONVENTIONAL BODY	STREAMLINED BODY	CONVENTIONAL BODY	STREAMLINED BODY
ROAD LOAD POWER REQ'D	33 H.P.	22 H.P.	33 H.P.	22 H.P.
RD. LD.% OF FULL LOAD	41.2%	27.5%	51.5%	34.3%
SPECIFIC FUEL RATE, LB./B.HP.-HR.	.74	.93	.62	.74
FUEL USED, LB./HR. AT ROAD LOAD	24.5	20.5	20.5	16.3
FUEL ECONOMY, MILES/GALLON	15.1	18.1	18.1	22.8

GAINS IN ECONOMY:	
(1) BY STREAMLINING WITH 3.89 RATIO	19.9%
(2) BY CHANGING THE RATIO FROM 3.89 TO 2.74 WITHOUT STREAMLINING	19.9%
(3) BY STREAMLINING WITH THE 2.74 RATIO	26.0%
(4) BY STREAMLINING AND CHANGING RATIO	51.0%

■ Fig. 11 – Effect of drive ratio on streamlining economy

APPENDIX

Derivation of Formula for Acceleration Horsepower:

Symbols defined.

a = a constant acceleration, ft/sec²

V_1 = speed at the beginning of the acceleration period, fps

V_2 = speed at the end of the acceleration period, fps

d = distance traveled during accelerating period, ft

F = the force accelerating the car, lb

M = the mass of the loaded car, lb(sec)²/ft

t = time in seconds required to increase the speed from V_1 to V_2

W = work done in producing the speed change, ft-lb

Hp = horsepower required to do the work W in the time t

The time required to accelerate from V_1 to V_2 is

$$t = \frac{V_2 - V_1}{a}$$

The average speed during this time is

$$\frac{V_1 + V_2}{2}$$

The distance covered is

$$d = \left(\frac{V_1 + V_2}{2}\right) t = \left(\frac{V_1 + V_2}{2}\right)\left(\frac{V_2 - V_1}{a}\right)$$

The accelerating force is

$$F = Ma$$

The work done by the force is

$$Fd = Mad = M\left(\frac{V_1 + V_2}{2}\right)(V_2 - V_1)$$

The horsepower required to do this work in the time t is

$$\text{Hp} = \frac{Fd}{550t} = \frac{\frac{1}{2}M(V_1 + V_2)(V_2 - V_1)}{550\left(\frac{V_2 - V_1}{a}\right)}$$

Hence

$$\text{Hp} = \frac{Ma}{550}\left(\frac{V_1 + V_2}{2}\right)$$

From this result it appears that the power required to produce a constant acceleration is equal to the product of the mass of the loaded car, the mean velocity during the accelerating period, and the acceleration.

Reynolds' Number

The exact mathematical expression for drag replaces the "constant," K, in the formula

$$D = KAV^2$$

with a variable expression which includes as factors the air density, a function of the ratio of the speed of sound to the air velocity, and another function usually written

$$f\left(\frac{VL}{\nu}\right)$$

The quantity within the parentheses is called the "Reynolds' Number." The numerator is the product of the air speed by a linear dimension of the body. The denominator is the "kinematic viscosity" (ratio of the absolute viscosity to the density) of the fluid. In the ordinary range of densities and velocities the expression may be written approximately[3]

Reynolds' Number = $9230\,LV$

where L is in feet and V is in mph.

[3] See "Airplane Design – Aerodynamics," by E. P. Warner, McGraw-Hill Book Co., 1936.

Development of the

by FRED E. WEICK

Chief Engineer, Engineering and Research Corp.

THIS paper describes the development of one particular two-place airplane intended for private use. The aim of its design was to make it particularly suitable for the private owner by having it unusually simple and easy to fly, quick to learn to fly, and free from the difficulties associated with stalling and spinning. It was also to have a good field of view for the pilot and a cruising speed of 100 mph with a low-powered engine.

The development of the plane began ten years ago when a small group of engineers at the NACA laboratories at Langley Field started a private study along these lines. This study included computations, tests on flying models, and the construction and testing of the W-1 and W-1A experimental airplanes. The latter is shown in Fig. 1. The results of this experimenting were published in a previous SAE paper given by the writer in January, 1936.[1]

The endeavor to obtain an airplane having these special characteristics resulted in the development of the W-1 and W-1A of the following unconventional features:

A. The tricycle landing gear with castering nose wheel, steerable if desired.

B. Suitable longitudinal and lateral stability with definitely limited upward elevator travel to prevent loss of control due to stalling and spinning.

C. A glide-control flap.

D. Two-control operation using pitching and rolling controls.

Our conclusions regarding these features following the various tests and trials were stated in the 1936 SAE paper, and are repeated as follows:

A. The stable long-travel, three-wheel landing-gear arrangement enables satisfactory landings to be made almost regardless of the wind direction, the air speed at contact, or the manner in which the airplane is flared at contact or guided to the ground.

B. Freedom from the danger of stalling was obtained by

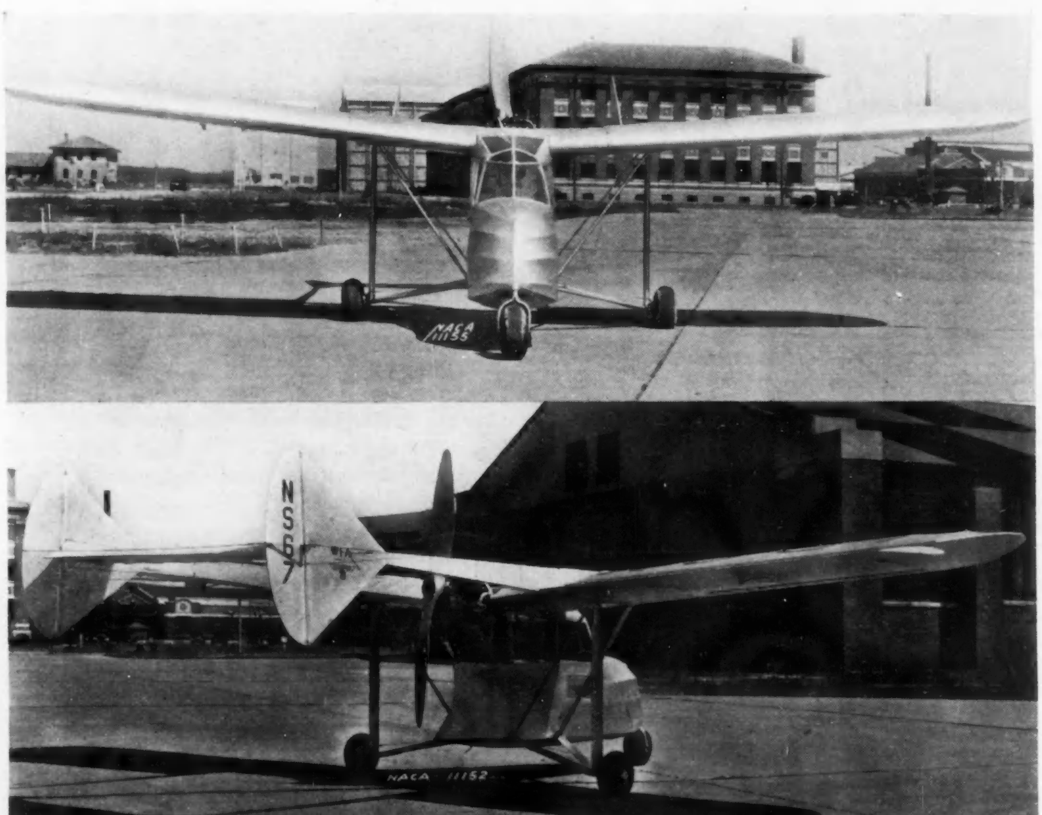


Fig. 1 — W-1A experimental airplane

[This paper was presented at the National Aeronautic Meeting of the Society, Washington, D. C., March 13, 1941.]

[1] See SAE Transactions, May, 1936, pp. 177-188: "Everyman's Airplane – A Move Toward Simpler Flying," by Fred E. Weick.

ERCOUPE . . .

An Airplane for Simplified Private Flying

■ Fig. 2 – Experimental Ercoupe as first flown

■ Fig. 3 – Experimental Ercoupe in its final form

DEVELOPMENT of a two-place airplane particularly suited to the needs of the private flyer was the aim of the design of the "Ercoupe." This was done by making it unusually simple and easy to fly, quick to learn to fly, and free from the difficulties associated with stalling and spinning; it was also to have a good field of view for the pilot, and a cruising speed of 100 mph with a low-powered engine.

The development of such a plane began ten years ago, Mr. Weick reports, when a small group of engineers at the NACA Laboratories at Langley Field started a private study that resulted in the construction of the W-1 and W-1A experimental airplanes having the following unconventional features:

1. The tricycle landing gear with castering nose wheel, steerable if desired.
2. Suitable longitudinal and lateral stability with definitely limited upward elevator travel to prevent loss of control due to stalling and spinning.
3. A glide-control flap.
4. Two-control operation using pitching and rolling controls.

The problem in the case of the Ercoupe, he points out, was to produce a salable low-powered airplane of good performance and attractive appearance that incorporated the ease of handling and other special characteristics that had been developed in the previous experimental planes. Characteristics of the final design provided with a 65-hp Continental motor were obtained, he brings out, by a long succession of flight tests, modifications, and more flight tests.

In the remainder of his presentation, Mr. Weick describes some of the trials and the changes that have been made in the design in order to obtain the flying and handling qualities desired.

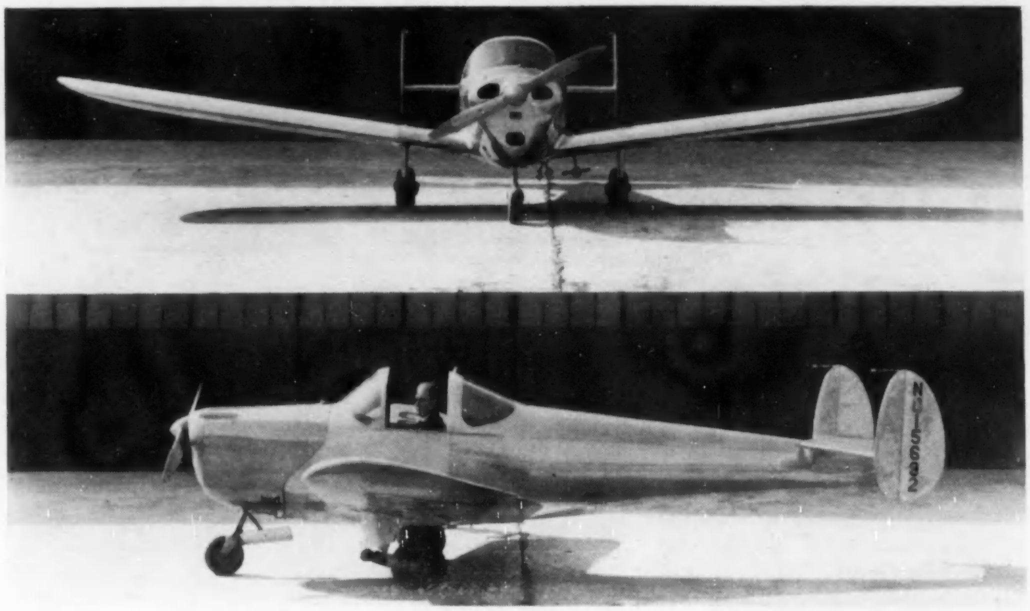
having the longitudinal stability and the available elevator control so related that the airplane, or at least the outer portions of the wing, could not be stalled. As long as high-lift wings are used, it seems a basic condition that this relation must hold in any completely successful solution of the problem.

C. The glide-control flap is very helpful in enabling an unskilled pilot to approach and to make contact at a desired point with greater accuracy than a good pilot with a conventional airplane. (Sufficient improvement in power-plant reliability may eliminate the necessity for extra controls of this nature.)

D. With the W-1A and its stable landing-gear, two-control operation using the elevator and slot-lip ailerons is quite satisfactory and sufficient for ordinary flying, including making sharp turns, holding a course in gusty air, and making take-offs and landings in all directions with respect to the wind, with straight or with curved approaches. It appears possible to us and we hope that the elimination of rudder control in the manner described may be a definite stride toward reducing the skill, training, practice, and general keenness required to fly safely.

In the case of the Ercoupe the problem was to produce a salable low-powered airplane of good performance and attractive appearance that incorporated the ease of handling and the other special characteristics that had been developed in the previous experimental airplanes. A low-wing tractor arrangement was decided upon for reasons that will be brought out later, and an experimental model was designed and built during 1937. This airplane is shown in Fig. 2 as it first appeared. It was constructed entirely of metal except that the outer panels of the wings and the control surfaces were covered with fabric.

An engine of approximately 60 hp was desired, but none was available at the time, and so a 40-hp Continental engine was used in the first airplane with the idea of getting it into the air as soon as possible and working out its handling characteristics. There being no 60-hp engine nor any



■ Fig. 5 – Production Ercoupe with Continental A-65 engine

promise of one, the Engineering and Research Corp. undertook the design and construction of one and obtained an approved type certificate from the Civil Aeronautics Authority. A 4-cyl inverted in-line design was chosen, largely with the view to obtaining good forward vision for the pilot. The original experimental plane was finally fitted with this Erco engine and was also modified in many other respects. It is shown in its final form in Fig. 3, and is, in fact, still in that form and in flying condition.

This design, which had been crystallized by many trials and changes, was then reworked for production and the first production model, which received its A.T.C. in January, 1940, is shown in Fig. 4. By that time commercial light plane engines of greater power were available and, in fact, were in sufficiently wide use that they could be purchased for less than the cost of the manufacture of our own engine in the relatively small quantities that were feasible. The airplane was therefore fitted with a 65-hp Continental engine and another A.T.C. obtained. It is in this form, shown in Figs. 5 and 6, that it has finally been put in production as the Ercoupe.

The design and development of this airplane, as is probably the case with many other light airplanes, has been accomplished without model wind-tunnel tests. The results of previous wind-tunnel research, published largely by the NACA, were, of course, used throughout the design. The present characteristics were obtained, however, by a long succession of flight tests, modifications, and more flight tests. The flight tests have the advantage of having certain conditions such as the air turbulence, the Reynolds number, and the dynamics of the problem in exactly the correct magnitudes, a matter of importance when stability and control near the stall are involved. In fact, the flight tests give, I believe, the next to the last word; the last comes from the user of the airplane.

With this rather lengthy introduction, the remainder of the paper will be devoted to a description of some of the trials and the changes that have been made in the design in order to obtain the flying and handling qualities desired. The changes have been made both as the result of our own testing and as the result of the as-yet limited experience of the users of the airplane.

■ Landing Gear

In designing the W-1, the following specifications were laid out as desirable for the landing gear:

(1) The landing should be safe practically regardless of how, or if, the flight path is leveled off at contact.

■ Fig. 4 – Ercoupe as first approved with Erco engine

(2) Landing should be satisfactory if contact is made at any speed up to at least twice the minimum, and the airplane should have no tendency to leave the ground after contact has been made.

(3) The landing gear should have stable-taxiing and easy-steering characteristics, being entirely free from ground-looping tendencies.

(4) Drift landings should be possible without incurring excessive side force on the landing gear structure.

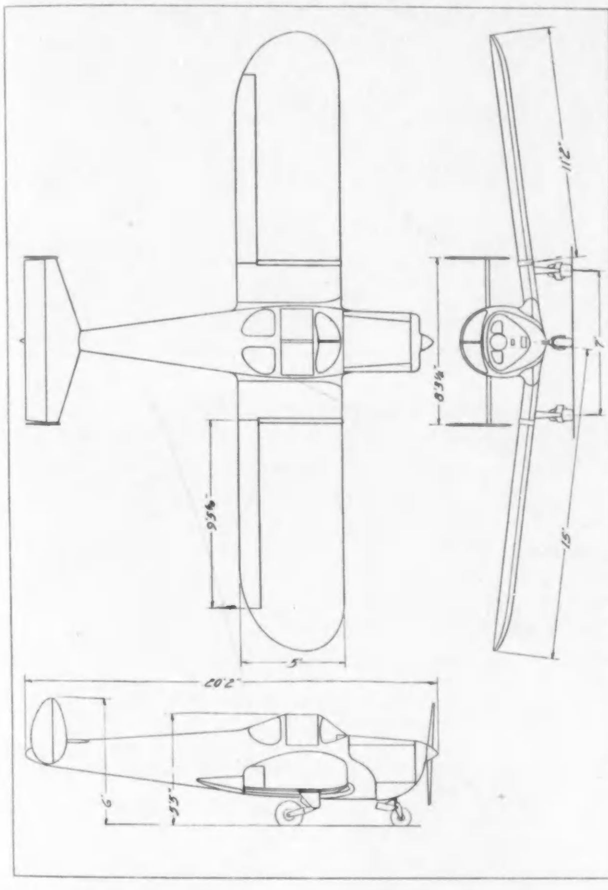
(5) The airplane should be practically impossible to nose over, even with poor terrain and full continuous application of the brakes.

To meet these specifications, a landing gear that we later referred to as the tricycle type was designed and constructed. At the time, it was my impression that this was the first gear to have a castering nose wheel and therefore the first of the present tricycle type, for the well-known earlier ships, such as the Curtiss Pusher, which had the same general wheel locations, had fixed axis nose wheels which did not caster or steer. I have found recently, however, that we were not actually the first to use a castering and steerable nose wheel, for one was included in a patent that has been brought to my attention which was applied

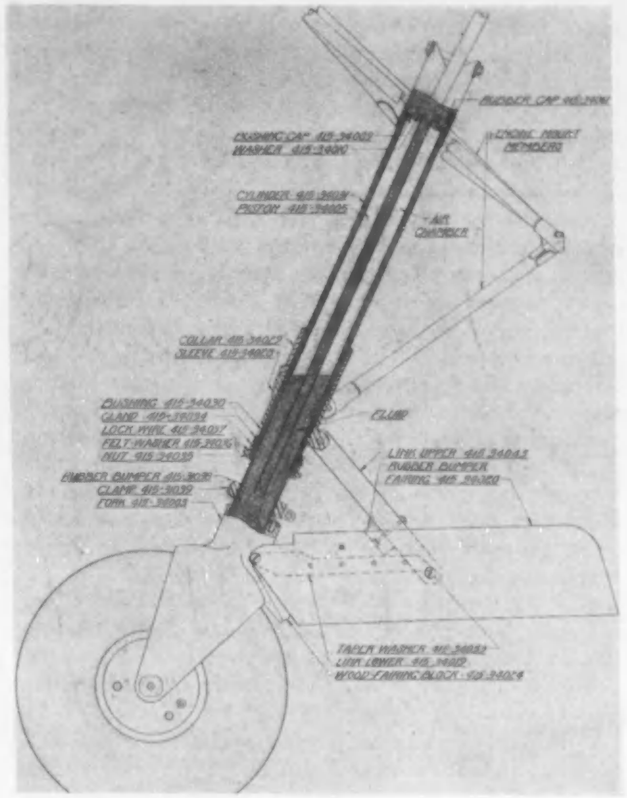
for in 1909 by the aeronautical experimental association headed by Alexander Graham Bell, and including Curtiss, Baldwin, Selfridge, and McCurdy, and reliable information has come to light that an airplane of the design patented was flown.

The tricycle gear used on the Ercoupe is essentially the same as that of the W-1, but the details of construction have, of course, been changed to fit the different general arrangement of the airplane. The wheel base is somewhat smaller because the forward position of the nose-wheel is limited by the tractor propeller, but it seems to be ample for practical purposes. This belief is substantiated by one actual experience that occurred during the first flight trials. The front oleo strut was not yet functioning properly in that it could compress only a small part of its total travel, and an extremely hard nose-down forced landing caused the front fork and oleo to bend back flat under the nose of the ship. The plane just slid to a stop on its nose and the two rear wheels, with no apparent tendency to nose over, and nothing else was damaged.

In this airplane the nose wheel is steered by means of the control wheel which also moves the ailerons and, in case of two-control operation, the rudders as well. No difficulty has been experienced with nose wheel shimmy except in an occasional case in which the steering link has worn loose and, in those cases, the shimmy has been eliminated by taking the slack out of the control linkage.



■ Fig. 6 – Three-view drawing of production Ercoupe



■ Fig. 7 – Diagrammatic view of nose-wheel installation

All three wheels have hydraulic shock-absorbing units and have 1 ft of vertical travel. The nose-wheel installation (Fig. 7) has an oleo cylinder of special design using hydraulic brake fluid for the shock-absorber element and using compressed air to form a spring for taxiing. Torque is transmitted to the wheel for steering purposes by means of a nutcracker linkage, and a sheet-metal fairing is attached to the lower member of the nutcracker in such a manner as to form a streamline trailing edge for the oleo tube when the wheel is all the way down in flying position. Fig. 8 is a view of the front of the airplane with the nose wheel down and the fairing in place.

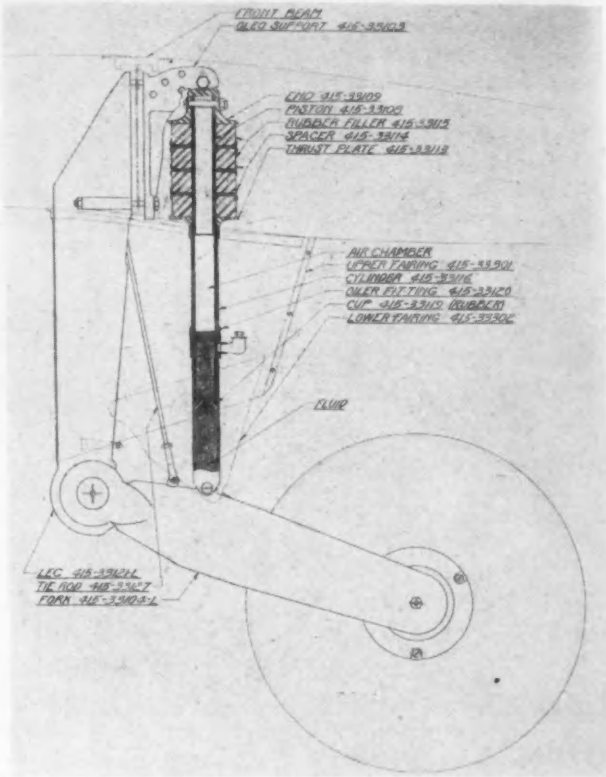
Although compressed air is used as a taxiing spring, it

■ Fig. 8 – Nose wheel in flight position showing fairing in place

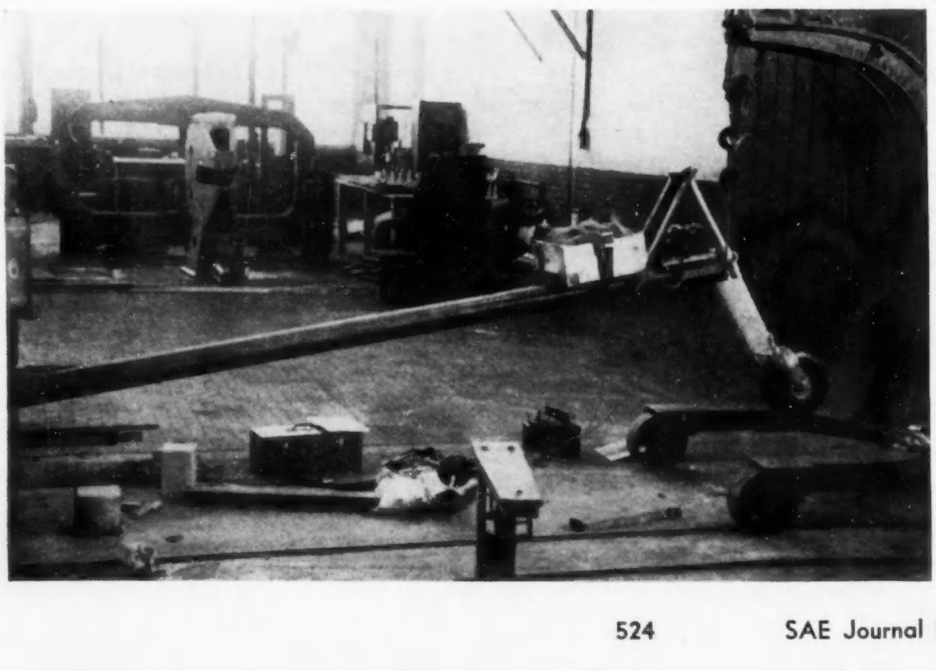
is not necessary to pump air into the strut in the usual manner. Air at atmospheric pressure is trapped in the piston when the oleo strut is filled with fluid and, as the load is put on the wheel and the piston moves into the cylinder, this air compresses to the point where it will support the nose of the airplane and form an air cushion for ground running.

Although the present nose oleo with its taxiing cushion of trapped atmospheric air requires no inflation and operates satisfactorily when in good condition, it does have air under pressure at all times that the airplane is on the ground, and it requires a perfect seal in order that the pressure is not lost. The strut operated without difficulty for nearly three years of experimental operation but, on account of slight, almost unavoidable, imperfection, it recently has caused some annoyance in the hands of several of the users. We have, therefore, constructed and tested a nose oleo unit which has the same internal dimensions but which uses a steel spring for taxiing purposes, and which should cause no difficulty because of leakage. The rig for drop-testing these nose wheel units is shown in Fig. 9. An NACA accelerometer was mounted above the nose wheel and entire unit was hinged as shown at the left of the picture. This unit is not yet in production, but it is under service test and will be fitted as standard equipment after CAA approval is obtained.

A diagram of the rear wheel installation is given in Fig. 10 and the installation is also shown in Figs. 5 and 8. The arrangement of this unit, and the design of many of the structural features throughout the airplane, were the work of the project engineer on the job, Frank B. Lane. The landing-gear forks and the posts on which they are hinged are high-strength heat-treated No. 220 aluminum-alloy castings. The post is fastened directly to the main wing spar at the outer end of the center section. An oleo shock absorber is located to the rear of the post, and springing for taxiing is obtained from rubber discs in compression at the top of the oleo tube. In the original plane the rubber discs were at the center of the oleo strut as shown in Fig. 3, and it was found almost impossible to enclose them in a reasonable fairing. In the production model, the discs are inside the wing contour which permits the use of a narrow low-drag fairing, housing both the post and the oleo structure. The cast-aluminum-alloy forks are designed to fit the form of the tires snugly. This was done primarily in order to obtain low drag but, in addition, it was hoped that the close fit with only about ¼ in. clearance from the tire would serve to scrape the mud off and not bind or jam. The fork was given more clearance in the center than at the edges so that the scraping or cutting



■ Fig. 10 – Diagrammatic view of rear wheel installation



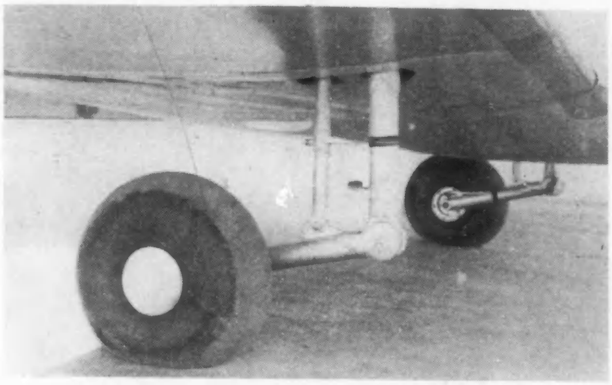
■ Fig. 9 – Nose wheel drop-testing unit

action would be relieved as in the case of a lathe tool. During the course of the experimental operation the forks seemed to operate satisfactorily in this respect even though extreme mud conditions were sought and found for testing purposes. When the production planes got out into commercial use, however, difficulty with mud retarding the wheels was reported occasionally. It seems that our worst trial had been made in soft wet mud which the forks scraped off easily, but a less moist and stickier mud, which is found under certain conditions, retards the wheels to an objectionable extent. A modified form of rear landing gear in which the fork has been eliminated is now under service test. The new type mounts on the wing in the same way and has the same shock-absorber unit, but is constructed of straight steel tubing with the wheel located to one side. It is shown without fairing in Fig. 11.

On the whole, the landing gear as used to date seems quite husky and able to stand up under exceptionally harsh treatment. Originally, the wearing parts were fitted with metal bushings which wore and became loose in a shorter time than was considered satisfactory, probably because they were difficult to keep clean. They were replaced early by reinforced bakelite material impregnated with graphite, and the bearings now seem to wear satisfactorily.

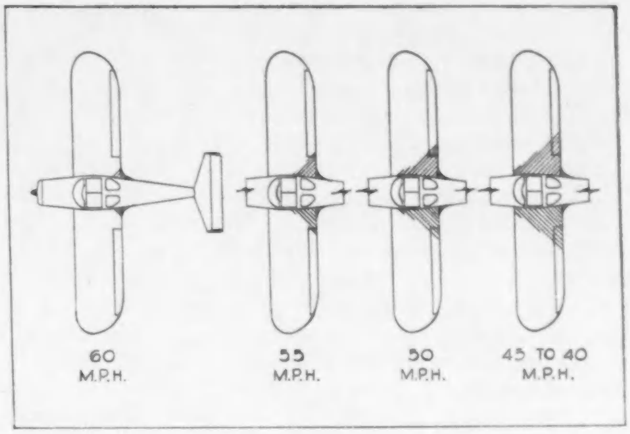
■ Stalling and Spinning

The way to avoid stalling difficulties is to avoid stalling the wings, or at least their outer or tip portions. This can be done in the old conventional manner by relying on the pilot not to stall, but the doubtful success of the method is indicated by the accident records of the CAA, which show that most of the fatal accidents that occurred in private flying during the past few years involved stalling or spinning. The stalling difficulty also can be avoided in a different manner by designing and building the airplane in such a way that it will not balance in flight at a high enough angle of attack so that the wings, or at least their outer portions, are stalled. In the W-1 this was accomplished by providing a longitudinally stable airplane and limiting the upward travel of the elevator surface. In the W-1A it was aided in addition by the slots of the slot-lip ailerons, which gave satisfactory control and stability but too great a drag to be used where performance is important.



■ Fig. 11 – Experimental rear gear without fork

In the present Ercoupe design the use of the limited upward elevator travel is continued and, in addition, the exact manner in which the wings approach the stalled condition has been treated carefully. A straight wing is used, the stalling of which naturally starts at the middle of the span. With the low-wing monoplane arrangement used, the stalling or breaking away of the airflow is sure to start at the juncture of the wing with the fuselage if no fillet is used, and the air speed and angle of attack at which the breaking away of the flow starts can be controlled by proper filleting.



■ Fig. 12 – Variation of burbled flow at center of wing with decrease in airspeed (shown by shaded areas)

A series of different fillets was tried in flight tests that extended over a couple of months, the upper surfaces of the wing being fitted with wool tufts to show the nature of the airflow. With the fillet finally selected, the flow starts burbling at the rear of the fillet when the speed of the airplane is reduced to a point a few miles per hour above the minimum speed, and the stalled portion gradually spreads as the speed is reduced, as shown by the portions of the wing shown shaded on Fig. 12. The outer portion of the wing does not stall even when the usual amount of maximum upward elevator is used, and so the stability and control are satisfactory in a glide with the control wheel full back even when the elevator travel is not limited below the conventional amount. Three factors contribute to the attainment of this condition. One is that the portion of the wing shown on the diagram stalls so definitely that the spanwise lift distribution is broken down in the center, and the outer panels act somewhat after the manner of two separate low-aspect-ratio airfoils, and can go up to a higher angle of attack before stalling. Another factor is that the pitching moment of the main wing changes as the center portion stalls in such a manner as to tend to decrease the angle of attack. Third, the loss of lift in the center of the wing reduces the downwash on the tail, thus reducing the effectiveness of the elevator in forcing the tail down and increasing the angle of attack. With all three of these factors at work, the usual amount of elevator control is insufficient to cause the outer portions of the wing, which in this case comprise most of it, to stall. Lateral stability and control are therefore maintained throughout the entire range of speed and angle of attack in gliding flight.

The general action is somewhat similar to that with the wing tip slots. A reasonable minimum speed is obtained with the outer portions of the wing about 6 deg below

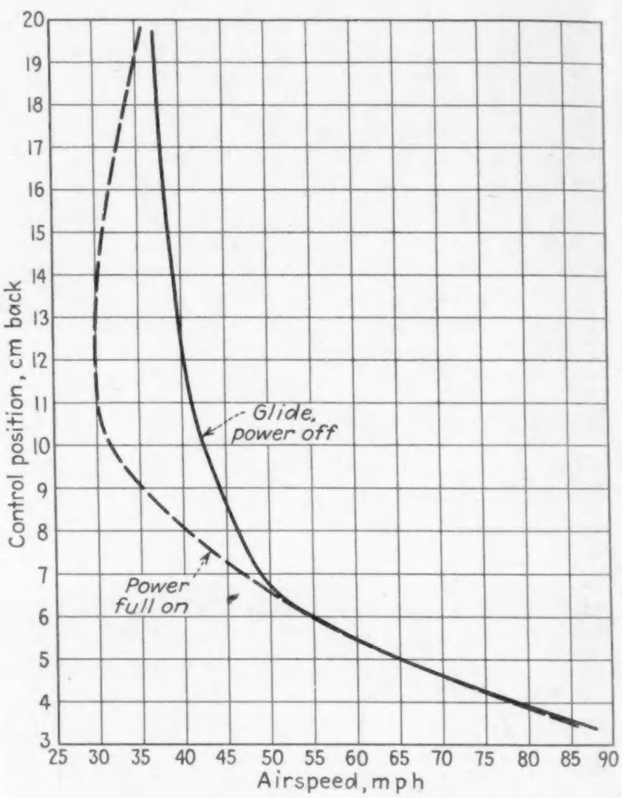
maximum lift by using an unusually high lift wing section for the Reynolds Number range in which it is working, the NACA 43013.

The burbled flow over the central portion of the wing serves another useful purpose in that, as the speed in a glide is reduced to about 5 mph above the minimum, a mild form of buffeting occurs which increases somewhat as the speed is reduced further. This buffeting automatically warns the pilot that he has little reserve speed or kinetic energy left for such purposes as overcoming the effect of gusts or flaring off the flight path for landing.

The problem of avoiding stalling with power on entails additional difficulties in the case of a tractor airplane. In the W-1 pusher, it was easy to place the thrust line of the propeller so as to give balance at the same speed, either in a glide with the throttle closed or in flight at any throttle setting, and so the same limited elevator position prevented stalling throughout the entire power range. With a conventional tractor arrangement, however, the slipstream passes over the center of the wing, increasing the downwash by a large amount. This added downwash increases the downward load on the tail and thus increases the angle of attack at which the airplane trims by an amount of the order of 10 deg or more. On this account if the up travel of the elevator of a conventional tractor airplane is limited to the point where the airplane cannot be maintained in stalled flight with power full on, the plane will not fly at all with power off except in a very steep dive. Yet it is important to avoid stalling dangers with power on as well as in gliding flight, for a large proportion of all accidents involving stalls occur with power on.

In order to obtain a satisfactory condition in this respect, the Ercoupe, which is, of course, a tractor, was arranged to have the propeller thrust give as large a nosing-down moment as possible. The low-wing arrangement and an engine having the propeller shaft near the top help in this matter, and the effect is increased further by inclining the engine axis downward toward the nose so that the line of thrust has a larger moment arm about the center of gravity of the airplane. With this arrangement, the airplane trims at approximately the same speed for any constant elevator position at any throttle setting within the range of speeds ordinarily used in flight, as is shown in Fig. 13. Being a tractor, however, the propeller slipstream prevents the carefully worked out breaking-away of the flow at the center of the wing as the speed is decreased. In fact, the lift over the center portion, and therefore the downwash on the tail, is increased with power on, and the airplane will fly at a lower speed than with power off. At the lowest speeds a single position of the elevator therefore gives a different airplane speed for each different throttle setting. The conventional amount of elevator control was sufficient to stall the wing with power on and for this reason a limitation to the up travel of the elevator was necessary.

Flight tests with the original experimental Ercoupe with the 40-hp Continental engine showed that, when the elevator up travel was restricted to a point at which the plane could not be maintained in stalled flight with power full on, it still had sufficient elevator control for all necessary flight maneuvers including taking off and landing. This was still true when the Erco in-line engine was installed in the plane and 55 to 60 hp taken from it. When the 65-hp Continental engine was installed in the production model, however, the increased slipstream effect required that the upward elevator travel be limited to such an extent if satisfactory power-on stalling characteristics were obtained that the tail could not always be lowered as much as desired during take-off or landing. This difficulty was overcome by giving the leading edges of the wings a sharply pointed contour for a distance of a few inches adjacent to the fuselage. These cause the air passing over the wings close to the fuselage to burble at high angles of attack even with the slipstream blast full on. Thus sufficient up elevator travel for landing and taking off can



■ **Fig. 13 – Elevator control position versus airspeed – power on and power off**

be used without giving unsatisfactory power-on stalling characteristics.

Observations of the results of a large number of crashes lead to the conclusion that, in the most serious ones, the nose of the airplane has, in practically every case, come in contact with the ground at a rather steep downward angle. The airplane inherently moves rapidly in the direction toward which the nose points, but it cannot move at a high speed in a direction perpendicular to the wing surface. In our airplane we have tried to provide a combination of control and stability such that the plane is, in no circumstances, likely inadvertently to approach the ground with the nose down.

The limited elevator control not only insures stable and controllable flight conditions throughout the low-speed flight range, but it prevents spinning as well. In addition to this precaution against spinning, the airplane has been designed in such a manner that it will not balance or trim in a spin. Beside the company pilots, expert test pilots and stunt pilots have attempted to force the experimental airplane to spin by all kinds of maneuvers, both power on and power off, some even in the inverted condition, but it

never has shown any signs of spinning. And certain of these trials were made under extreme conditions with the full unrestricted elevator travel and with 20 lb of lead in the tail of the ship to make certain that it could be stalled. Finally after thorough tests by the Civil Aeronautics Authority, the production plane was officially rated "Characteristically Incapable of Spinning."

■ Elimination of Rudder Pedals

In the flight trials of the W-1A it was found that two-control operation including taking-off and landing under all sorts of conditions could be carried out very well with the elevator and the slot-lip ailerons only, but the control with the elevators and rudders only was relatively unsatisfactory. In both cases successful handling in cross-wind landings depended on the stability of the tricycle gear in taxiing and running along the ground.

A theoretical analysis by R. T. Jones of the NACA staff indicates that a turning control giving pure rolling moments and no yawing moments is close to the optimum for two-control operation, although a small and varying amount of yawing moment is required to produce perfect entries into and recoveries from turns with no trace of skidding or slipping. Ordinarily conventional ailerons can be used reasonably well, but the effect of their inherent adverse yawing moments must be overcome by the natural weathercock stability of the airplane, and a rather large amount of slipping and skidding in turns is likely to result. The use of conventional ailerons alone as the turning control in a two-control system has the disadvantage also that their effectiveness is likely to be lost as the stall is approached. The slot-lip ailerons used on the W-1A had no yawing moments of any importance and they gave satisfactory control, but as mentioned previously, their drag was excessive.

The first production Ercoupes were constructed in such a manner that the pilot can take his choice as to whether he flies it with two controls or with the conventional three controls. The system can be changed from the conventional three-control to the two-control arrangement in a few minutes by adding a link connecting the ailerons and rudders together and removing the rudder pedals. Turning the control wheel then moves the ailerons, rudders, and nose wheel simultaneously, and turns the airplane whether it is flying in the air or rolling on the ground.

In the first experimental Ercoupe we had the nose wheel steered by the rudder pedals when the conventional three-control system was in use but, to our surprise, even experienced pilots when taxiing would try to turn the airplane by turning the control wheel as in an automobile, even though it moved only the ailerons, and those in the opposite direction from that which they were trained to use to aid in making a turn on the ground. We have therefore provided for the nose wheel to be turned by the control wheel whether the two- or the three-control arrangement is in use.

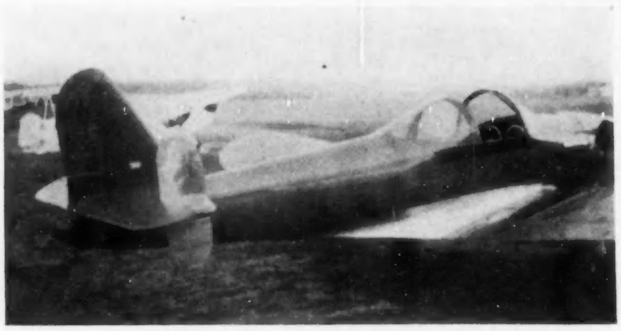
By linking the rudders to the ailerons in the two-control arrangement, the optimum relation between rolling and yawing moments can be obtained for one flight condition. In our case we have chosen that this relation should occur at a low speed because that is the most difficult condition in which to obtain satisfactory control. At high speeds the system therefore provides slightly larger yawing moments than would be ideal, but this can be noticed in flight only if an extremely sharp turn is made suddenly at high speed. Such a turn is made but rarely, and a bit of slipping and skidding accompanying it is of no great importance. A number of skilled pilots have flown the plane both with the two-control and the three-control systems, and it is an interesting observation that the slipping and skidding during the flights, as indicated by the ball in the bank indicator, have been noticeably less with the two-control than with the conventional three-control arrangement in use, unless the pilot paid special attention to flying carefully with the three controls.

One able test pilot, Melvin N. Gough of the NACA staff, who is experienced in flying and testing all types and sizes of airplanes, flew the experimental plane alternately with two, three, two, and three controls on an extremely gusty day, and he was considerably surprised to find that he noticed less bumpiness with the two- than with the conventional three-control arrangement. It seems that, with only two controls, he realized that he had incomplete control over the attitude of the plane and he did not try so hard to overcome the effects of the gustiness. When he let the stability do the work for him, he apparently had a smoother ride.

An interesting set of trials that brought out one of the advantages of two-control operation was made in 1939 by one of the company pilots, Robert Sanders. On a day with a high wind he made a series of take-offs from the College Park Flying Field, each time simulating sudden engine failure and a forced landing by closing the throttle at a definite point and then making a landing back in the field. During the trials he cut off his power at enough different points along the take-off path to cover the entire possible range. In some cases he had to make fairly sharp downwind turns and land with the wind in order to get back in the field and, in making these turns, it was apparent to him that the airplane was often flying at an entirely different attitude than that which he would have maintained had he had the conventional three controls. The two-control airplane was flying correctly with respect to the air, but the pilot could not see the air and had to fly on the basis of his view of objects on the ground and his estimate of wind velocity. He had originally intended to repeat the trials with all three controls in use for comparison, but he changed his mind and cancelled that part of the program on the basis that he was sure to slip and thereby lose altitude so that he could not complete his turn from the same height.

As in the case of obtaining freedom from stalling difficulties, the problem of obtaining satisfactory two-control operation is more complex with a tractor airplane than with a pusher. The W-1 had two sets of fins and rudders, one on each boom and, being outside of the propeller slipstream, there was no noticeable difference in directional trim with variation of power. In the case of the conventional tractor, however, the twisting slipstream passing the body and the vertical surfaces cause the directional trim to change as the power is increased. With a right-hand propeller the vertical surfaces, which are ordinarily above the center of the slipstream, receive in effect an increasing blast from the left, and the rudder must be deflected toward the right to compensate for the effect and maintain the plane on a straight course. This was the case with the first experimental Ercoupe shown in Fig. 2, which had been made with a single vertical tail to be as conventional in appearance as possible.

In two-control operation it is desirable that the directional trim remain unchanged throughout the entire range of speeds and throttle setting. In order to eliminate as much as possible the change of directional trim with variations in throttle setting and airspeed, two modifications were tried. One of these was to cant the propeller axis to the right various amounts, and the other was to add various amounts of vertical fin area to the tail outside of the central portion of the propeller slipstream. One of the latter arrangements is shown in Fig. 14. These experiments



■ Fig. 14 -- Experimental Ercoupe with additional vertical fins

showed that greatly improved power-on and power-off directional trim could be obtained with the proper arrangement. Finally, the tail was redesigned to have two fins and rudders at the outer extremes of the horizontal tail surfaces as in the case of the W-1. This is the arrangement shown in Fig. 3 and used later in the production model. Even with this tail arrangement, having the vertical surfaces entirely outside of the slipstream, it was found advisable to cant the propeller shaft axis 3 deg to the right to overcome the unsymmetrical effect of the slipstream over the nose of the fuselage. With this arrangement the directional trim is practically unchanged by throttle setting throughout the entire speed range of the airplane.

The reception of the idea of flight without rudder pedals has been very encouraging. The airplane, as stated previously, is available with either the conventional three-control arrangement or with the two-control arrangement having no rudder pedals. It was thought that many pilots with experience on conventional airplanes would be reluctant to give up the separate rudder control. It has come as a very agreeable surprise, then, that, except for four planes delivered to the CAA, only one plane has been delivered with rudder pedals. And after about two weeks of use the two-control arrangement was tried in that one and the rudder pedals have never been put back in.

■ Longitudinal Control

In the first experimental Erco airplane, shown in Fig. 2, the elevator control was found to be too sensitive, and a very slight fore-and-aft movement of the control wheel would produce a disproportionately large change in the attitude of the plane. This was immediately improved to a reasonable condition by changing the linkage and cutting down the elevator travel about one-third, but maintaining the same total control-wheel movement. In this condition the upward travel of the elevator was still sufficient for all flight runs and sufficient to stall the airplane with power full on. This arrangement was reasonably satisfactory but the control was still a bit sensitive and gave pilots the immediate response that is associated with pursuit planes. It was decided that more damping in pitch was desired and, in order to obtain it, the production model was given a 12 in. greater tail length. This accomplished the desired purpose.

■ Lateral Control

The aileron control on the Erco experimental plane as first flown was not entirely satisfactory at low air speeds, mainly because, with the linkage used, it was necessary to turn the control wheel too far to get the desired response in roll, and sufficient movement could not be made quickly enough under gusty air conditions. With the ailerons fully deflected, they gave an average amount of rolling control. They were large, covering 20% of the chord, nearly all of the span and having an equal up and down deflection of about 12 deg. The control did not seem entirely sufficient at the lowest air speeds, however, for, inasmuch as the airplane could be flown at speeds all the way down to the minimum without fear of difficulty due to stalling, it seemed desirable to have a quick response in lateral control even at minimum speed.

At the lowest speeds the aileron control was obviously hindered by an undesirable amount of adverse yawing. This was thought to be associated with the fact that the airflow over the center portion of the wing burbled at the lowest flight speeds and the vertical fin, which is relied upon to act against the aileron yaw, was operating in this burbling wake of retarded air. The adverse yawing was substantially reduced and the aileron control at low speeds definitely improved by the extra fin area shown in Fig. 14. This improvement was retained in the final detail design shown in Fig. 3.

Even though the aileron control is reasonably satisfactory in the final arrangement of the first experimental airplane, it seemed that even better control in the low-speed range would be desirable. The production design was, therefore, modified further in three ways. The deflections were increased, and extreme differential linkage was used, and a Frise-type balance lip was added. The large deflection and extreme differential movement fit in well with the steering of the nose wheel by the aileron wheel. In order to steer easily while taxiing slowly, it was found desirable to use a maximum deflection of the control wheel of 180 deg in each direction. In normal flying, however, it was found desirable not to have to turn the control wheel beyond 90 deg. With the Ercoupe linkage, a 90-deg movement of the control wheel moves one aileron up 25 deg and the other down 10 deg with an ordinary differential motion. Turning the control wheel to the limit of 180 deg moves the up aileron up to 50 deg, but the down aileron back to 0 deg. This latter half of the deflection is not required in ordinary flying, for the 90-deg control-wheel deflection gives as much rolling control as is ordinarily available in airplanes of this general category. Under special conditions of gusty air at low speeds, the extra 90-deg movement of the control wheel gives a large reserve of rolling control which is particularly advantageous in that, with maximum control deflection, there is no down aileron to encourage the stall at the low-speed conditions of flying.

It is the opinion of the writer that safe operation is aided by abundant control and maneuverability to overcome the effects of gusts and misjudgments. The Ercoupe, because of its unusually powerful aileron control, is ex-

ceptionally maneuverable. It is not in the least sluggish, as might be inferred from the term, "limited control."

Landing Approach

If two-control operation (without rudder pedals) is being used, the conventional sideslip is not available for steepening the path momentarily to avoid overshooting. In the W-1A, a glide-control flap solved this problem very satisfactorily, but an extra control was involved. In the design of the Ercoupe an attempt was made to have a minimum number of controls for the pilot to operate.

The original experimental plane was designed with ailerons covering practically the entire wing span with the idea in mind that, if a glide-steepening device was considered essential after some flight experience had been obtained, the inner portions could be cut off and converted into flaps. It was hoped, however, that this would not be found necessary.

After a certain amount of flying had been accomplished, it was decided that the flaps would not be required because any of several means for losing altitude could be used with a bit of practice. The first obvious one is merely to nose the airplane down and land it at a higher speed, and this works quite satisfactorily if the altitude to be lost is not too great and if the landing field is smooth and not too small. If the excess height in the approach is noticed at an early enough stage, a relatively steep glide can be obtained by reducing the speed to the point at which the center portion of the wing is well burbled (see Fig. 12) and the drag is high. In that case it is well to nose down at an altitude of about 200 ft and pick up sufficient speed to flare off the flight path and reduce the vertical velocity before contact with the ground is made. Altitude can be lost even more quickly in the low-speed glide if the wings are rocked from side to side about 15 deg and rather quickly. A certain amount of yawing occurs if the wing is rolled back from a bank before the turn has developed. Finally, one of the best ways of adjusting the approach nicely seems to be to make the time-honored 90-deg turn which can be cut short or stretched as required.

Of course, all of these maneuvers require a certain amount of piloting skill, and there has been an indication from some users of the airplane, mostly instructors, that some means for steepening the glide path while flying straight and at normal approach speed would be desirable. Trials with spoilers used for this purpose are now scheduled and should be made in the near future. It is possible that either spoilers or flaps will be added if experience indicates this action desirable, but the desire for a glide steepener of some kind seems to be diminishing as the users are becoming familiar with their airplane.

Piloting Suggestions

A number of special features have been built into the Ercoupe to make it particularly useful and easy to fly safely. It will not fall out of control or into a spin. It can be landed at any speed up to twice the minimum without ballooning off the ground, and it is free from the dangers of ground-looping or nosing-over. This does not mean, however, that the airplane is foolproof. Safe operation requires that the pilot know the capabilities of the airplane and operate within them, allowing a reasonable margin for unforeseen contingencies.

Taxiing – Maneuvering on the ground is accomplished by merely opening the throttle sufficiently to cause the desired forward motion, and steering the nose wheel with the control wheel. To stop, the wheel brakes are applied by means of a hand grip just below the throttle handle. The brake handle can be locked for parking by turning the grip to a horizontal position.

If, with an extremely high wind, the airplane should have a tendency to weathercock into the wind, skidding the nose wheel somewhat against the pilot's control, improved traction and steering control can be obtained by keeping the control wheel forward, applying the brakes a small amount and turning the handle to lock them if desired, and using greater engine power for taxiing.

Taking Off – With its tricycle gear and nearly level wing the Ercoupe can be run along the ground at high speeds. The start of the take-off run usually is made with the wheel in the neutral or medium position. To take off the ground it is necessary to increase the angle of attack of the wing by lowering the tail, and this is usually done by moving the control wheel back gently after the minimum take-off speed has been exceeded by a comfortable margin.

It is possible to have the airplane take off by itself, with no force applied to the control wheel, by setting the longitudinal trim adjustment to a nose up (or low-speed trim) condition.

The shortest take-off run is ordinarily obtained by holding the control wheel full back throughout the entire ground run. The tail will not come down until flying speed has been attained. The wheel should be eased forward as soon as the plane leaves the ground, however, or the nose will point up too steeply for good climb and it may drop again momentarily with some loss of altitude.

It is advisable not to climb steeply after taking off until an airspeed reading of at least 45 mph has been reached, because the airplane will fly at lower speed with full power than without power and, in the event of engine failure, the airplane would be caught at less than its minimum flying speed without power. (See Fig. 13.)

In taking off cross-wind or in gusty air it is advisable to keep the control wheel well forward, which holds the nose wheel firmly on the ground, until the desired take-off speed has been reached, and then to take off decisively and without hesitation. If the airplane is being operated without rudder pedals, it may weathercock into the wind just after it leaves the ground, but this need cause no concern. It is merely adjusting itself to true flight with respect to the air, and a straight course of travel is maintained without difficulty.

This brings up a change in point of view that it seems an experienced pilot of conventional airplanes must pass through before he can be satisfied with two-control operation (without rudder pedals). He has been accustomed to controlling the attitude of his airplane about all three axes as well as the flight path and the speed. With two-control operation he must be willing to sacrifice his control in yaw and to rely upon the stability of the landing gear to handle the drift in a cross-wind take-off or landing. Not until he feels fully confident that the airplane itself will take care of these items satisfactorily and without strain can he be expected to fly a two-control airplane with a feeling of comfort and pleasure.

In flight without rudder pedals, turns are made by simply turning the control wheel until the proper bank is reached and at the same time keeping the nose in the position desired by adjusting the fore-and-aft position of the control

wheel. The airspeed required in a steady turn is higher than in straight flight because the lift is not vertical and only the vertical component of the lift will support the airplane against gravity. For example, the Ercoupe in a gliding turn with a 60-deg bank and the control wheel full back will show an airspeed reading of about 60 mph as compared with about 40 mph in a straight glide with the control wheel full back. Sharp turns naturally require steep banks with the control wheel well back.

If the airplane is in a power-off glide and the speed is gradually reduced by easing the wheel back, a mild jouncing or buffeting will be noticed at about 5 mph above the minimum speed. This is caused by the burbling of the air flow at the juncture of the wing and fuselage, and has been designed into the ship as an active warning that the minimum speed is being approached. The airplane will fly satisfactorily at minimum speed with the wheel all the way back in a glide, but the practice is not recommended at low altitude because no reserve energy is available to overcome the effects of gusty air or misjudgment.

If the control wheel is eased back gradually with power full on, the airplane will reach an uncomfortably nose-high attitude. In this condition the flying will not be smooth or steady, but control is maintained.

Landing – A good airspeed reading during the approach to a landing is one between 55 and 60 mph. As the ground is approached, the flight path is leveled off so as to reduce the vertical velocity, and then usually the plane is immediately set on the ground. It need not be held off in the conventional way until minimum speed has been reached, except in the case of rough terrain. After contacting the ground at any speed above the minimum, the wheel should not be pulled back or the airplane may take off again. The wheel should, therefore, either be held still or eased forward to a neutral position, preferably the latter.

The airplane can be held off in the conventional manner until it loses its flying speed. This practice is preferred by some pilots in order to reduce the ground run, and is always advisable in case of rough terrain.

After landing, the brakes may be used as desired. In an emergency they may be applied before the landing is made, but this procedure is not recommended as standard practice on account of the tire wear involved. During the landing run the brakes should be released if slippery terrain or loose gravel is encountered and a tendency to skid is noticeable, as in the case of an automobile.

If in the approach to landing the pilot finds that he is overshooting slightly, he can nose the airplane down and put it on the ground immediately at a relatively high speed. With immediate application of the brakes the landing will require decidedly less overall distance than it would if the airplane were held off the ground until minimum speed had been reached. Also if the approach has been made at too high an altitude the flight path can be steepened by any of the methods previously mentioned under the heading "Landing Approach."

If the airplane without rudder pedals is being landed in a strong cross-wind, it must be pointed up-wind sufficiently to keep the flight path in line with the runway, and it will approach the ground with the wings level but with considerable drift. This should not alarm the pilot for, with the stable tricycle gear, contact with the ground made in this manner is quite satisfactory because the stability of the gear will automatically change the heading of the airplane so that it continues down the runway. In making cross-wind landings it is well to set the airplane on the ground very definitely, and to move the control wheel forward somewhat in order to hold the nose down and reduce the lift on the wings as contact is made. Under strong cross-wind conditions some pilots prefer to glide straight to the ground without pulling the control wheel back to flare off the glide path before contacting the ground. This insures making a definite contact. It is also helpful in some cases to set the brake on about half way before making contact, and possibly to apply it strongly just after contact is made. This tends to hold the nose down and the wing at a low angle of attack and also cuts down the speed to that of a reasonable taxiing value as quickly as feasible. Another important point in a cross-wind landing is that the airplane should be given its head with very light if any lateral pressure on the control wheel at the moment that contact is made. This permits the stable castering tendency of the nose wheel to act and the airplane will change its heading slightly so that it lines up with the direction of its motion along the ground.

In a few cases pilots inadvertently have turned the airplanes off the runways in cross-wind landings while trying to keep the windward wing down. A pilot familiar with conventional airplanes is inclined to try to make the airplane line up with its direction of motion at the time of contact. If he has a cross-wind from the left, for example, in the approach the airplane will be headed somewhat into the wind or to the left of the course being followed if the approach is made in line with the runway. As the airplane approaches the ground, the pilot may turn the control wheel to the right in order to make the airplane head along the runway. With the two-control arrangement the airplane will naturally bank to the right and the left wing on the windward side will be higher than the other. To get the left wing down again the pilot turns the control wheel to the left, bringing the wings approximately level as the airplane touches the ground. At the instant of contact, however, the control wheel is held to the left and the nose wheel is, therefore, also held to the left. Thus the airplane immediately starts a left turn on the ground, which sets up a centrifugal force tending to raise the left wing again. The pilot cannot see the nose wheel, but is conscious of the tendency of the left wing to rise, and therefore, turns the control wheel still farther left turning the airplane more sharply and aggravating the situation rather than relieving it. In an extreme case the right wing tip may even touch the ground during a maneuver of this nature, and the pilot may feel that the plane has ground-looped to the left, whereas actually he has unconsciously steered it into a sharp left turn while trying to keep the wings level.

Flight tests and demonstrations under cross-wind conditions have shown that, if the pilot will be careful not to over-control the stable castering tendency of the nose wheel at the moment of contact, the airplane will follow a straight path. In certain cases, particularly where exceptionally fast landings were made in gusty cross-winds, the windward wing would tend to rise somewhat and stayed up for a certain distance. The wheel was not turned, however, and the plane was allowed to follow a straight course even though the windward wing was uncomfortably high, probably about two feet above normal at the tip. After the speed had dropped somewhat, the wing in each case came down to normal and the rest of the landing run was un-

eventful. These experiences show that, as would be expected, there is undoubtedly a limit to the magnitude of cross-wind that the airplane should be landed with, particularly unless the pilot is very skillful in handling it. On the other hand, under all ordinary cross-wind landing conditions, if contact is made very definitely, say without flaring off the glide path, at an airspeed of about 60 mph, and no effort is made to turn or hold the control wheel at the moment of contact, a satisfactory landing should result.

In gusty air or in high winds in general, it is usually advisable to approach and land at a somewhat higher speed than in still air and to have the airplane at all times either definitely in the air or definitely on the ground.

■ Acknowledgment

As is evident from the contents of this paper many sources have contributed to the development of the Ercoupe. These include the previous projects in the private airplane field, such as in particular the Stout Sky Car, the research work of the NACA, the small group of engineers who designed and constructed the W-1, the Department of Commerce which encouraged the development through the purchase of the W-1A when modified by the Fairchild Co., and finally the personnel of the Engineering and Research Corp. which has transformed the experimental product into a commercial private-owner's airplane.

Earth-Moving Equipment in National Defense

UNDOUBTEDLY the problem of earth moving is more important to the present national defense program than most of us realize, for in its scope is included not only that excavating which is inherently necessary for highways, cantonments, airports, ammunition dumps, new factories and all the other construction work, but also mining of iron ore, copper, nitrates, coal, mercury, tin and other raw materials. To begin with the House Roads Committee has reported H-5110, the "Defense Highway Act of 1941," authorizing appropriations of $287,000,000 for specific purposes as follows: $100,000,000 for immediate correction of critical deficiencies in roads and bridges of approximately 78,000 miles on the strategic military system of primary roads; $150,000,000 for construction and improvement of access roads to military and naval reservations, defense industry sites, and sources of essential raw materials; $2,000,000 for experimental flight trips along public highways; $10,000,000 to cooperate with the states in making advance highway engineering surveys; and $25,000,000 to reimburse states for repairing roads damaged by the Army and Navy.

This immense highway program is only a small part of the "earth-moving" program required generally throughout the entire national defense scheme. When we consider, for instance, that approximately 60% of all iron ore mined today is done in open pits with power shovels and tractor-drawn units and high percentages of other raw materials are produced the same way, we can readily visualize the great and increasing importance of the tractor, truck, shovel, and other excavating and haulage equipments and the economics of each.

The excavating machinery and allied industries have progressed rapidly during the past few years as a result of the continuous quest for lower costs of "dirt moving" as well as because of the unusually great progress made in the development of new construction materials and welding procedures. We find ourselves today with machines which are much faster, lighter and stronger than they have been. As an example, a ½-yd power shovel which met obsolescence in 1933 weighed 45,400 lb. The same size shovel obsoleted in 1936 weighed 31,000 lb. The modern ½-yd power shovel introduced this year weighs only 26,100 lb.

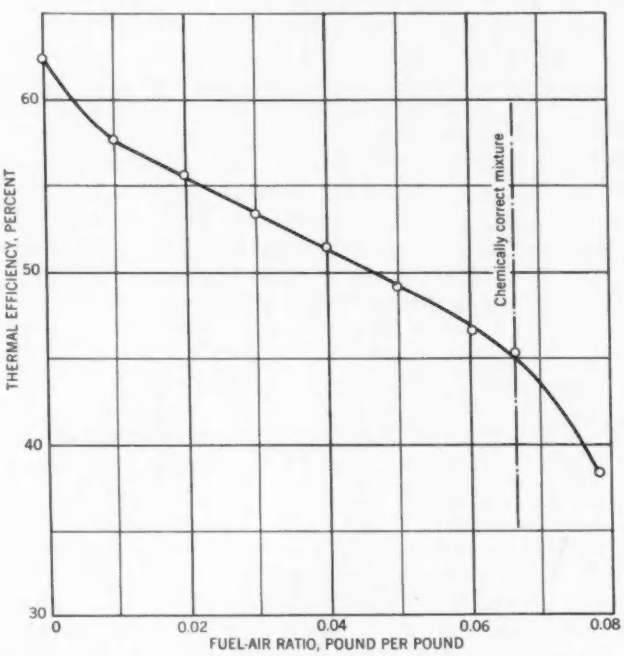
The advent and perfecting of the crawler-type tractor have revolutionized the dirt-moving industry, whether it be highway or mining, within the past few years. As the design of the diesel engine has progressed so also has the tractor. Higher speed industrial-type diesel engines of much less weight have been one of the primary factors in the development of today's industrial tractor. New uses of crawler tractors now become more and more the fact as new allied equipments for them are built and their applications broadened. As open pit mines become deeper, locomotive and train haulage is replaced by truck haulage in order to reduce costs and speed up production. Roads for these trucks must be built and maintained. The tractor has now entered the field of gold mining in Alaska. Hardly a strip mine, especially where trucks are used for haulage, is without tractors for cleaning the surface of the coal and for building and maintaining roads. Tractors and scrapers also have been introduced and are being used for certain coal stripping operations where they can be used. The national defense airports which number into the hundreds are being constructed primarily with tractors and tractor equipment. Tractors have entered the field of land clearing and reclamation work. Tractors and bulldozers now excavate basements for dwellings. They lay thousands of miles of pipe line and load sand, gravel, cinders and other materials on construction jobs and in industrial plants. The logging industry in the West as in other parts of the country has also found the tractor. The preservation of our forests is made possible by skidding the logs out with tractors rather than with destructive high-line methods which methods we all hope will become more and more a part of the past. Tractors will save our forests.

As exemplified by the "Defense Highway Act" previously referred to and its immense appropriation the construction of new roads for civilian and military purposes is an immediate problem. The tractor with its various scraper, bulldozer and other attachments has broken into this field merely because dirt under many conditions can be moved much cheaper by these means than it can with shovels and trucks. Many of our old highways were purposely made long and crooked in order to get around and eliminate expensive rock cuts. Today we find this different. Rock cuts or not, the highway must be kept as straight as possible, and the power shovel with truck haulage continues to be the best tool available for this work.

Excerpts from the paper: "Comparisons of Methods and Costs of Earth Moving," by George W. Mork, engineer in charge of Tractor Equipment Division, Bucyrus-Erie Co., presented at the National Tractor Meeting of the Society, Milwaukee, Wis., Sept. 26, 1941.

A Rational Basis for Correlating Data on
COMPRESSION-IGNITION
at Different Intake

IN connection with studies by the Bureau of Mines of the hazards that might attend the use of diesel engines underground, consideration was given to the possible effects of changes in the temperature and pressure of the intake air on the quantity of harmful and objectionable gases produced by such engines. In preparing for this phase of the Bureau's investigation, a study was made of published data[2-7 inc.] on the performance of compression-ignition engines at different intake and exhaust conditions. This study disclosed that correlation of the indicated efficiency with fuel-air ratio had not been examined in analyzing such data. Inasmuch as thermodynamic considerations show that indicated efficiency should be related to fuel-air ratio, it appeared desirable to analyze existing data on this basis. Accordingly this was done, and the striking correlations obtained emphasized the fundamental nature of the relation and suggested its use in correlating data on engine



▪ **Fig. 1 – Relation between computed thermal efficiency and fuel-air ratio for a limited-pressure fuel-air cycle with octane fuel and an assumed compression ratio of 13.9:1**

THIS paper discusses first the theoretical relation between thermal efficiency and fuel-air ratio for assumed limited-pressure fuel-air cycles. Data on actual cycles are then presented showing the relation of indicated efficiency to fuel-air ratio. This and other relationships are then used in deriving equations for relating the power output of engines operated at constant speed and throttle setting to ambient conditions.

The fundamental nature of the relation between thermal efficiency and fuel-air ratio is emphasized by showing the variation with fuel-air ratio of computed efficiency of assumed limited-pressure fuel-air cycles. This computed efficiency for a given cycle and compression ratio is determined solely by the thermodynamic properties of the fuel-air medium and is the maximum efficiency theoretically possible for the assumed conditions.

The paper also includes an analysis of published data on compression-ignition engine performance showing the relation between actual indicated efficiency and fuel-air ratio. The data available indicate that this relation is not affected significantly by changes in ambient conditions, provided that the pressure at the intake equals the pressure at the final exhaust. Because this basic relation is independent of ambient conditions, it may be used in predicting the effect of changes in ambient conditions on the power output of compression-ignition engines operated at constant speed and throttle setting. This can be done only if the ef-

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 5, 1941.]

[1] Published by permission of the Director, Bureau of Mines, Department of the Interior.

[2] See SAE Transactions, 1935, pp. 210-214: "A Rational Basis for Comparing Diesel-Engine Performances," by E. S. Dennison.

[3] See ASME Transactions, Vol. 63, 1941, pp. 91-95: "Effect of Variations in Atmospheric Conditions on Diesel-Engine Performance," by J. S. Doolittle.

[4] See *Diesel Power*, Vol. 11, 1933, pp. 541-545: "Reducing the Performance of a Solid-Injection Diesel Engine to Standard Conditions," by H. A. Everett.

[5] See SAE Transactions, 1934, pp. 217-223: "Altitude Performance of Aircraft Engines Equipped with Gear-Driven Superchargers," by R. F. Gagg and E. V. Farrar.

[6] See NACA Technical Note No. 619, 1937: "Compression-Ignition Engine Performance at Altitudes and at Various Air Pressures and Temperatures," see also SAE Transactions, 1937, pp. 263-272: "Compression-Ignition Engine Performance at Altitude;" both by C. S. Moore and J. H. Collins, Jr.

[7] See SAE Transactions, 1937, pp. 312-314: "Correcting Diesel Performance to Standard Atmospheric Conditions," by C. F. Taylor.

ENGINE PERFORMANCE and Exhaust Conditions[1]

by MARTIN A. ELLIOTT

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performance at different atmospheric or ambient temperatures and pressures.

It is the object of this paper: (a) to show that the relation of indicated efficiency to fuel-air ratio offers a rational basis for correlating data on engine performance throughout a wide range of conditions; and (b) to suggest methods

fect of changes in ambient conditions on fuel-air ratio is known or can be computed. Inasmuch as the quantity of fuel supplied to a compression-ignition engine depends only on the speed and throttle setting and is not affected by changes in ambient conditions, the effect on fuel-air ratio of such changes can be determined when the quantity of air entering the engine at each condition is known. This latter factor is determined by the density of the air and by the volumetric efficiency; both of these factors depend on ambient temperature and pressure. Equations relating all of these factors are presented herein. These equations furnish a rational basis for computing the power output of a compression-ignition engine at a given set of conditions from the power output at some other set of conditions. The precision with which the equations represent test results is shown.

The effect of changes in ambient conditions on the power output of engines in which the fuel-air ratio is constant for operation at constant speed and throttle setting is discussed briefly. This discussion applies to the gasoline or more precisely the spark-ignition engine equipped with a carburetor. It is pointed out that knowledge of the relation between fuel-air ratio and indicated efficiency is not required in formulating equations for computing the effect of ambient conditions on the power output of gasoline engines because fuel-air ratio and, consequently, indicated efficiency are more or less constant, depending on the operating characteristics of the carburetor.

for utilizing this and other relations in correcting to standard ambient conditions the power output of compression-ignition engines operated at constant speed and throttle setting.

For the purposes of this report "engine performance" at a given speed and throttle setting will be defined completely by data on power output, fuel consumption, volumetric efficiency, and mechanical efficiency.

Table 1 – Computed Pressures, Volumes, Temperatures, Internal Energies, and Indicated Efficiencies for Limited Pressure Fuel-Air Cycles with Octane as the Fuel

Fuel-air ratio, lb per lb	0.0	0.01	0.02	0.03	0.04	0.05	0.0605	0.0665	0.0782
p_1[a]		14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7
V_1[b]		14.14	14.24	14.44	14.47	14.52	14.72	14.72	14.75
T_1[c]		558	562	570	571	573	581	581	581
E_1[d]		6.6	7.4	8.7	8.9	9.3	10.6	10.6	10.6
p_2		546	548	548	548	548	546	546	546
$V_2 = V_3$		1.017	1.024	1.039	1.041	1.045	1.059	1.059	1.061
T_2		1490	1507	1529	1532	1538	1553	1553	1552
E_2		178.2	181.3	185.7	185.9	187.5	190.6	190.6	191
$p_{3a} = p_3$		570	640	720	810	845	850	860	870
V_{3a}		1.416	1.646	1.788	1.868	2.005	2.19	2.24	2.27
T_{3a}		2143	2766	3343	3888	4308	4675	4805	4720
E_{3a}		322.5	478.7	646.8	815.6	984	1148	1258	1472
p_4		26.3	37.8	49.0	59.8	71.9	82.5	92	83
$V_4 = V_1$		14.14	14.24	14.44	14.47	14.52	14.72	14.72	14.75
T_4		1000	1414	1838	2231	2643	3070	3280	3000
E_4		87.3	172.3	270.8	373.3	490	615	695	920
$p_5 = p_1$		14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7
V_5		21.9	29.3	36.6	42.7	51.8	55.5	61.5	59.5
T_5		855	1131	1397	1613	1936	2100	2260	2070
E_5		60.6	115.2	173.4	226.7	310	360	403	658
η[e]	0.623[f]	0.576	0.556	0.533	0.514	0.491	0.465	0.453	0.383

[a] – Pressures expressed as psi.
[b] – Volume of mixture, cu ft per lb. of air.
[c] – Temperatures in deg R.
[d] – Total internal energy, Btu per lb of air
[e] – Efficiency expressed as a ratio.
[f] – Limiting efficiency = efficiency of constant-volume air cycle.

Relation to Fuel-Air Ratio of Computed Thermal Efficiency

The theoretical or maximum possible thermal efficiency of a given internal-combustion-engine cycle can be computed from the thermodynamic properties of the fuel-air medium. Inasmuch as these properties are related to fuel-

air ratio, it follows that thermal efficiency must also be related to fuel-air ratio. The basic nature of this relation is therefore apparent.

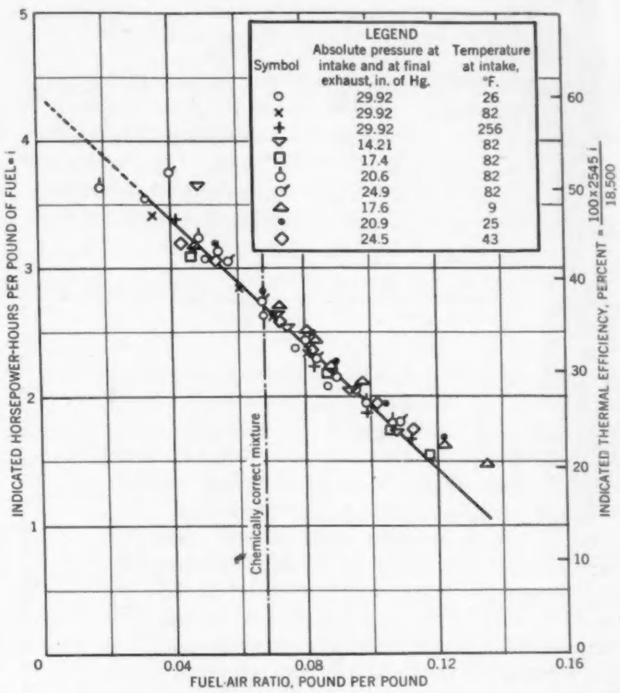
Computation of the thermal efficiency of limited-pressure fuel-air cycles is discussed briefly in the following. This fuel-air cycle was selected because it is a rational criterion for the actual cycle of the high-speed compression-ignition engine, in which the maximum pressure is limited by the design and method of operating the engine. The designation "fuel-air cycle" is the same as that used by Taylor and Taylor[8] and signifies a so-called "ideal cycle" in which all computations are based upon the thermodynamic properties of the fuel-air medium.

In calculating the thermal efficiency of a limited-pressure fuel-air cycle, it is necessary to assume a value for the maximum pressure. Inasmuch as this pressure is affected by fuel-air ratio, the relation between maximum pressure and fuel-air ratio must be known before the variation of computed thermal efficiency with fuel-air ratio can be determined. In the calculations summarized in this report the data of Rothrock and Waldron[9] were used to establish the relation between maximum or limited pressure and fuel-air ratio. The values selected from the results of these investigators are shown as p_{3a} in Table 1. Inasmuch as these results were obtained with a single-cylinder test engine having a compression ratio of 13.9:1, this compression ratio was assumed in calculating the thermal efficiencies.

The computation of thermal efficiencies at the higher fuel-air ratios was facilitated by using the thermydynamic charts developed by Hershey, Eberhardt, and Hottel[10] and applied as outlined by Taylor and Taylor[8]. At fuel-air ratios lower than those for which the charts were constructed, a slight modification in previous methods of calculation was used. In all computations a fuel having the same chemical composition as octane was assumed because the thermodynamic charts were constructed for mixtures of octane and air. From the standpoint of the thermodynamic calculations the slight difference between the chemical composition of octane and diesel fuel has no significance[11]. In all calculations the temperature of the intake air was assumed to be 80 F, and the pressure at the intake and at the final exhaust was assumed to be 14.7 psi absolute.

The computed thermal efficiencies at different fuel-air ratios are shown in Table 1, along with other pertinent computed results. The subscripts have the following significance: 1 denotes the beginning of the compression stroke; 2, the end of the compression stroke; 3, the end of the addition of heat at constant volume; 3a, the end of the addition of heat at constant pressure and the beginning of the expansion stroke; 4, the end of the expansion stroke; and 5, the condition after the exhaust valve opens and expansion to final exhaust pressure occurs.

The relation between computed thermal efficiency and fuel-air ratio for the assumed limited-pressure fuel-air cycles is shown in Fig. 1. At fuel-air ratios within the normal operating range of compression-ignition engines, the relation can be represented without significant error by a straight line. At fuel-air ratios outside this range the relation appears to be curvilinear. The relation shown in Fig. 1 is in general similar to that of either the constant-volume or constant-pressure fuel-air cycle[11]. This result would be expected because, at a given compression ratio and quantity of heat added, the efficiency of the limited-pressure cycle is less than that of the constant-volume fuel-air cycle and greater than that of the constant-pressure fuel-air cycle.



■ **Fig. 2 – Relation between indicated efficiency and fuel-air ratio shown by results of Moore and Collins' tests[6] of a compression-ignition engine at different intake conditions**

It is of interest to note that, for a given compression ratio, the efficiency of the limited-pressure cycle approaches that of the constant-volume cycle as the fuel-air ratio approaches zero. This can be determined from the expression for the efficiency of the limited-pressure cycle which is given by Taylor and Taylor as

$$\eta = 1 - \left(\frac{1}{r}\right)^{K-1} \left[\frac{\alpha\beta^K - 1}{(\alpha - 1) + K\alpha(\beta - 1)}\right], \tag{1}$$

in which η = thermal efficiency of fuel-air cycle, expressed as a ratio,

K = ratio of specific heat at constant pressure to specific heat at constant volume,

r = compression ratio,

$$\alpha = \frac{P_3}{P_2}, \tag{2}$$

$$\beta = \frac{V_{3a}}{V_3}, \tag{3}$$

p = pressure, psi,

V = volume, cu ft per lb of air.

The subscripts have the same significance as stated previously. As the fuel-air ratio approaches zero, both α and β approach 1, and the fraction in brackets approaches 1, as

[8] See "The Internal Combustion Engine," by C. F. Taylor and E. S. Taylor, International Textbook Co., Scranton, Pa., 1938.

[9] See NACA Technical Report No. 545, 1936: "Effects of Air-Fuel Ratio on Fuel Spray and Flame Formation in a Compression-Ignition Engine," by A. M. Rothrock and C. D. Waldron.

[10] See SAE Transactions, 1936, pp. 409-424: "Thermodynamic Properties of the Working Fluid in Internal-Combustion Engines," by R. L. Hershey, J. E. Eberhardt, and H. C. Hottel.

[11] See University of Illinois Engineering Experiment Station Bulletin 160, 1927: "A Thermodynamic Analysis of Internal-Combustion Engine Cycles," by G. A. Goodenough and J. B. Baker.

indicated by taking the first derivative of both numerator and denominator.

It should be pointed out that the computed efficiency may be in error at fuel-air ratios greater than the chemically correct value, because the charts of Hershey, Eberhardt, and Hottel[10] are based upon the assumption that no free carbon is present in the equilibrium mixture, whereas comparatively large quantities of free carbon are present during combustion under over-rich conditions in the compression-ignition engine[12].

Relation of Indicated Efficiency to Fuel-Air Ratio When Pressure at Intake Equals Pressure at Final Exhaust – Moore and Collins[6] studied the performance of a single-cylinder compression-ignition engine under conditions carefully controlled to simulate operation in atmospheres whose temperatures ranged from −3 to 256 F and whose absolute pressures ranged from 14.21 to 29.92 in. hg. In several series of experiments the pressure at the intake was equal to the pressure at the final exhaust. Unfortunately these investigators did not tabulate their results; however, it was possible to reproduce their numerical data by scaling their curves. From these numerical data efficiencies and fuel-air ratios were computed. Unless otherwise noted, indicated efficiency will, for convenience, be expressed as indicated horsepower-hours per pound of fuel. In computing fuel-air ratios it was necessary to assume that volumetric efficiency was independent of fuel-air ratio. This assumption should not introduce an appreciable error. Sample numerical data of Moore and Collins[6] and results computed from these data are presented in Table 2.

Table 2 – Performance of Compression-Ignition Engine[a] Observed by Moore and Collins[6] in Tests at Different Intake Conditions

Absolute Pressure in Intake and at Final Exhaust, in. hg	Temperature Intake, F.	Fuel per Cycle,[b] lb x 10⁴	Air per Cycle,[b] lb x 10³	Imep,[b] psi	Ihp-Hr per Cycle,[c] x 10⁴	Fuel-Air Ratio,[c] lb per lb	Ihp-Hr[c] per Lb of Fuel
29.92	256	1.73	4.16	101.8	5.89	0.0416	3.40
29.92	256	2.86	4.16	131.7	7.61	.0687	2.66
29.92	256	4.65	4.16	134.0	7.75	.1117	1.68
29.92	82	1.82	5.23	108.0	6.24	.0348	3.43
29.92	82	3.20	5.23	157.9	9.13	.0612	2.85
29.92	82	4.24	5.23	172.6	9.98	.0810	2.35
29.92	26	1.96	5.91	120.1	6.94	.0332	3.54
29.92	26	3.99	5.91	182.9	10.57	.0675	2.65
29.92	26	5.20	5.91	187.9	10.86	.0880	2.09
20.90	25	1.98	3.68	109.1	6.31	.0538	3.19
20.90	25	2.51	3.68	122.6	7.09	.0682	2.82
20.90	25	4.48	3.68	129.4	7.48	.1217	1.67
17.57	9	1.44	3.10	79.7	4.60	.0465	3.19
17.57	9	2.14	3.10	101.2	5.85	.0691	2.73
17.57	9	3.78	3.10	106.0	6.12	.1220	1.62
14.21	82	1.13	2.32	71.2	4.12	.0487	3.65
14.21	82	1.66	2.32	76.1	4.40	.0716	2.65
14.21	82	2.50	2.32	75.3	4.35	.1077	1.74

[a] Single-cylinder, four-stroke cycle engine having a 5-in. bore and 7-in. stroke with a piston displacement of 137.5 cu in.; compression ratio, 14.5; engine speed, 2000 rpm.
[b] Scaled from curves.
[c] Computed from data of Moore and Collins.

Relation to Fuel-Air Ratio of Indicated Efficiency

General – The foregoing discussion makes it apparent that the computed efficiency of the assumed limited-pressure cycles was related to fuel-air ratio because, for a given set of assumptions, the efficiency was determined by the thermodynamic properties of the fuel-air medium which, in turn, are determined by fuel-air ratio. Inasmuch as the same considerations would apply to actual cycles, a similar relation between indicated efficiency and fuel-air ratio would be expected, although the relation for an actual cycle would be modified by engine design and by heat losses occurring during the cycle.

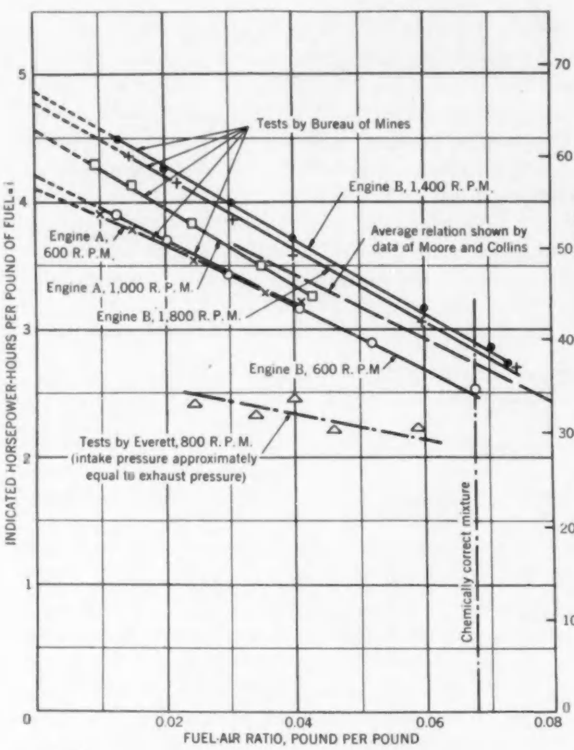
Because of its basic nature, the relation between indicated efficiency and fuel-air ratio of an actual cycle may offer a means for comparing performance data of engines obtained at different intake conditions, provided that changes in the intake conditions were without significant effect on the relation between indicated efficiency and fuel-air ratio. To determine this, available experimental data were analyzed as described in the following:

The indicated efficiency and fuel-air ratio computed from the data of Moore and Collins[6] are correlated in Fig. 2. For comparative purposes indicated thermal efficiency calculated by assuming a value of 18,500 Btu per lb for the lower heating value of the oil is shown also as the ordinate.

Fig. 2 demonstrates clearly that the relation between indicated efficiency and fuel-air ratio shown by the data of Moore and Collins[6] generally was not affected significantly by changes in the ambient temperature[13] and pressure when the pressure at the intake equaled the pressure at the final exhaust. The excellent correlation of the results in Fig. 2 emphasizes the fundamental nature of the relation shown and indicates the importance of this correlation in analyzing data on the performance of compression-ignition engines. The application of this relation to the correction

[12] See ASME Transactions, Vol. 63, 1941, pp. 97-105: "The Significance of Diesel Exhaust-Gas Analysis," by J. C. Holtz and M. A. Elliott.

[13] Computed efficiencies of fuel-air cycles for a limited range of intake temperatures have shown only minor effects of this factor.



■ **Fig. 3 – Relation between indicated efficiency and fuel-air ratio shown by tests of four-stroke-cycle compression-ignition engines**

of the performance of such engines for changes in intake conditions will be discussed in a subsequent section of this report.

The relation shown in Fig. 2 can be represented by a straight line having the general form

$$e = e_0 - af, \tag{4}$$

in which e = ihp-hr per lb of fuel,

f = fuel-air ratio, lb per lb,

e_0 and a = constants for a given engine operated at a given speed.

Inserting the values of e_0 and a indicated by the data of Moore and Collins[6], Equation (4) becomes

$$e = 4.39 - 24.6f. \tag{5}$$

In connection with the linear relation indicated by these results, it is of interest to compare relations observed for other compression-ignition engines. Data for determining the relation between indicated thermal efficiency and fuel-air ratio for each of two commercial four-stroke-cycle compression-ignition engines were available in the results of the Bureau of Mines investigation of the composition of exhaust gases produced by Diesel engines[14]. To utilize the results of this study it was necessary to develop a method for estimating indicated horsepower from measurements of brake horsepower and fuel consumption because, in the Bureau of Mines investigation, facilities were not available for testing engines without accessories or for measuring indicated horsepower. The method for estimating indicated horsepower depends upon an extrapolation to zero fuel consumption of the rate of change of horsepower with fuel consumption. Indicated horsepower estimated by this method agreed closely with indicated horsepower shown by performance data published by the manufacturer of one of the engines.

The relation between indicated horsepower-hours per pound of fuel and fuel-air ratio for the engines studied by the Bureau of Mines and for an engine studied by Everett[4] is shown in Fig. 3. For comparative purposes the relation indicated by the data of Moore and Collins[6] is reproduced in Fig. 3.

Fig. 3 shows that the relation between indicated efficiency and fuel-air ratio for both engine A and engine B (studied by the Bureau of Mines) might be represented without significant error by a straight line throughout a wide range of fuel-air ratios. It appears that the results of Everett might also be approximated by a straight line. It is of interest to compare Fig. 3 with Fig. 1 and also to note the similarity between the relations shown for the actual cycle which represents data from four different type engines and includes results obtained at different engine speeds. However, it is apparent that both engine design and engine speed have an effect on the slope and on the y intercept, consequently the constants e_0 and a in Equation (4) would be affected by these factors. The values of these constants for the engines for which data were available are summarized in Table 3. The results shown in Table 3 appear to indicate a relation between these constants and engine speed, although more experimental data would be required to verify this observation.

It should be pointed out that the relation between indicated efficiency and fuel-air ratio may not be linear at the lower fuel-air ratios if the fuel is not burned completely. Evidence of this deviation from a linear relation at lower fuel-air ratios may be indicated by data published by Taylor[7]. Unfortunately, the original data from this investigation were not readily available although, from the published curves, it was possible to compute the ratio of indicated mean effective pressure to fuel-air ratio. This factor is directly proportional to indicated efficiency and is shown in relation to fuel-air ratio in Fig. 4 for three different intake temperatures. The observed relation might be approximated by a straight line at fuel-air ratios between 0.03 and 0.06 lb per lb but, at fuel-air ratios less than 0.03 lb per lb the factor proportional to indicated thermal efficiency was less than that which would be expected if the relation were linear. Such a decrease in indicated efficiency at lower fuel-air ratios might occur because fuel in locally over-lean regions failed to react completely with

Table 3 – Numerical Value of Constants e_0 and a in Equation, $e = e_0 - af$

Engine	Engine Speed, rpm	Value of e_0, (intercept on y-axis)	Value of a, (slope)
Engine A – Bureau of Mines tests	600	4.11	22.8
	800	4.61	29.2
	1000	4.56	30.3
	1200	4.33	24.4
	1400	4.12	22.6
Engine B – Bureau of Mines tests	600	4.21	25.6
	1000	4.65	29.0
	1400	4.87	29.2
	1800	4.77	29.1
Engine studies by Moore and Collins[6]	2000	4.39	24.6
Engine Studied by Everett[4]	800	2.7	10

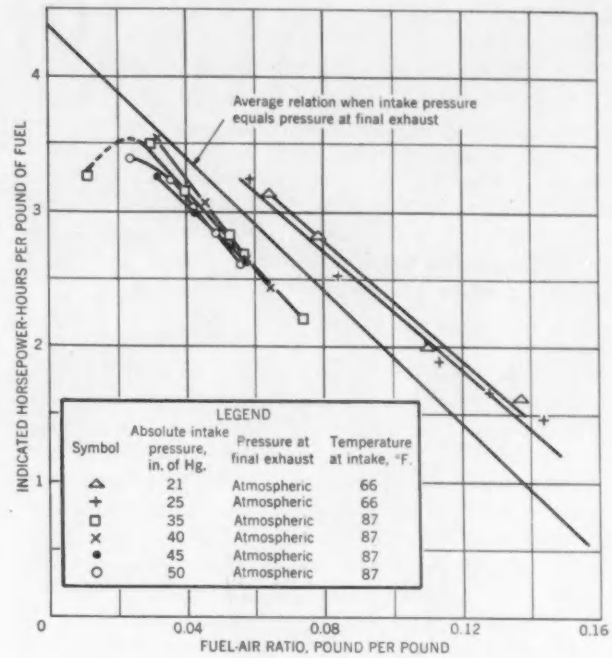
[14] See "Diesel Engines Underground. I – Composition of Exhaust Gas from Engines in Proper Mechanical Condition," by J. C. Holtz, L. B. Berger, M. A. Elliott, and H. H. Schrenk, Bureau of Mines, Report of Investigations 3508, 1940.

oxygen. In locally over-lean regions the concentration of fuel is below that corresponding to the lower inflammable limit of the fuel at the pressure and temperature existing in the combustion space. Therefore complete inflammation does not occur in such regions although some fuel may be partly oxidized. The existence of locally over-lean regions during combustion in the compression-ignition engine and the possible importance of the lower limit of inflammability in relation to the combustion process are discussed in a forthcoming report[15].

Relation of Indicated Efficiency to Fuel-Air Ratio when Pressure at Intake is Different from Pressure at Final Exhaust – The results discussed in the foregoing were obtained under conditions such that the engine exhausted into an atmosphere in which the pressure was equal to that at the intake. Such conditions probably are of most general interest because they apply to operation of compression-ignition engines that are not supercharged. However, Moore and Collins[6] in their studies of compression-ignition engine performance at altitudes and at various air pressures and temperatures obtained data on the operation of the test engine under conditions such that the pressure of the atmosphere at the intake was different from that at the final exhaust. To determine whether operation under such conditions had an effect on the relation between indicated efficiency and fuel-air ratio, the results of Moore and Collins[6] were used to compute the data presented in Fig. 5.

Fig. 5 shows that, when the pressure at the intake was greater than that at the final exhaust, the indicated efficiency at a given fuel-air ratio was less than that observed when the pressures at intake and exhaust were equal. This decrease in efficiency would not be expected because, in the tests of Moore and Collins[6], air at increased intake pressure was supplied by an auxiliary compressor that was not driven by the engine. Because of this, an increase in efficiency might be predicted owing to a tendency toward a reduced pumping loss as the intake pressure increases. This tendency toward a reduced negative work-area with increasing intake pressure has been observed in light-spring diagrams taken from a single-cylinder engine[16].

A reasonable explanation for the apparent effect of an increased intake pressure on the indicated efficiency at a given fuel-air ratio and final exhaust pressure can be offered when it is realized that the intake and exhaust valves were both open for the period from 27 deg before top-center to 41 deg after top-center. During this period of valve overlap air flowing into the cylinder from the intake also could flow out through the exhaust valve. Consequently, all of the air coming through the intake passages might not remain in the cylinder and act as working fluid, and therefore the fuel-air ratio indicated by measuring the fuel and metering the air would be less than that actually existing in the cylinder during the working cycle. If this were the case, the lines in Fig. 5 representing the relation between indicated efficiency and fuel-air ratio when the intake pressure was greater than the exhaust pressure should be moved to the right because the fuel-air ratio actually existing during the working cycle is greater than that shown and is the proper ratio with which to correlate data on indicated efficiency. If these lines are moved to the right, they approach the line representing the relation between indicated efficiency and fuel-air ratio when intake and exhaust pressures were equal. This suggests that, in the range of pressures studied, increases in intake pressure at a given exhaust pressure might not have a great effect on the relation between indicated efficiency and fuel-air ratio provided the fuel-air ratio existing during the working cycle is used as the basis of the correlation. Further experi-



Symbol	Absolute intake pressure, in. of Hg.	Pressure at final exhaust	Temperature at intake, °F.
△	21	Atmospheric	66
+	25	Atmospheric	66
□	35	Atmospheric	87
×	40	Atmospheric	87
•	45	Atmospheric	87
○	50	Atmospheric	87

Fig. 5 – Relation between indicated efficiency and fuel-air ratio shown by results of Moore and Collins' tests[6] of a compression-ignition engine when pressure at intake was different from pressure at final exhaust

[15] See "Diesel Engines Underground. III – Effect on Exhaust-Gas Composition of Operating Engines in Mixtures of Normal Air and Natural Gas," by M. A. Elliott, J. C. Holtz, L. B. Berger, and H. H. Schrenk, Bureau of Mines Report of Investigations 3584, 1941.

[16] See "Some Factors Affecting Friction in an Internal-Combustion Engine," by A. E. Hale and E. H. Olmstead, Thesis, Massachusetts Institute of Technology, 1937.

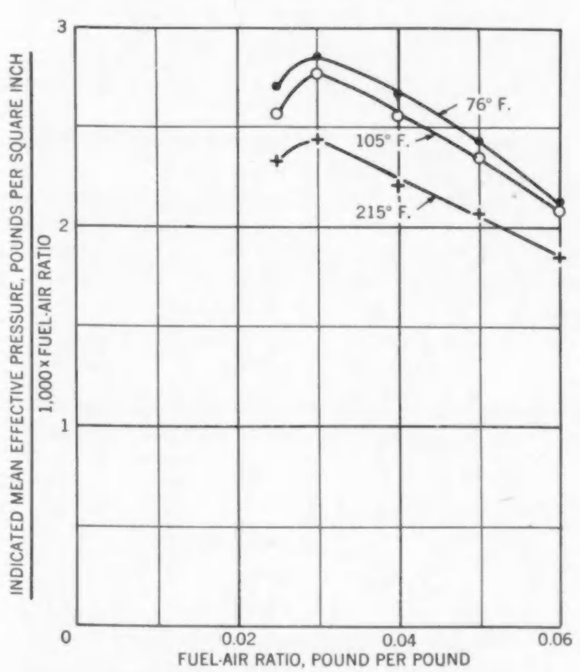


Fig. 4 – Relation to fuel-air ratio of a factor directly proportional to indicated efficiency computed from data of Taylor[7]

mental work would be necessary to determine the validity of this presumption.

Fig. 5 shows also that, when the intake pressure was less than the exhaust pressure, the indicated efficiency at given fuel-air ratio was greater than that observed when intake and exhaust pressures were equal. This increase in efficiency would not be expected[16] and reasoning similar to that offered in the discussion immediately preceding shows that the fuel-air ratio determined by measuring the fuel and metering the air does not represent the ratio of fuel to the medium present prior to combustion.

When the pressure at the final exhaust is greater than that at the intake the fuel-air ratio is not the proper ratio with which to correlate data on indicated efficiency because, under such conditions, significant quantities of exhaust gas might flow into the cylinder during the period of valve overlap. Because of this the medium present before combustion would be a mixture of air and exhaust gas, and therefore the ratio of fuel to this mixture is of greater significance than the fuel-air ratio. Obviously, the former ratio is less than the fuel-air ratio determined by measuring the fuel and air because of the presence of significant quantities of exhaust gas. It is apparent, therefore, that the relation shown in Fig. 5 when the pressure at the intake was less than the pressure at the final exhaust would approach the relation for equal intake and exhaust pressures if the ratio of fuel to the medium present before combustion were used as the basis of the correlation in both instances.

Although beyond the scope of this report, it nevertheless may be of interest to mention the possibility that scavenging efficiency might be estimated from measurements of fuel-air ratio determined by weighing the fuel and metering the air and from determinations of the fuel-air ratio existing during the working cycle. The use of these two fuel-air ratios in such an estimate might overcome some of the objections to using chemical analyses of the cylinder gases in attempting to determine scavenging efficiency[17].

Correction of Performance for Changes in Intake Conditions

The foregoing analysis of the results of tests made by Moore and Collins[6] in studying the operation of a single-cylinder compression-ignition engine throughout a wide range of intake conditions has shown that the relation between indicated efficiency and fuel-air ratio was not affected significantly by changes in the pressure and temperature of the intake, provided that the pressure at the intake equaled that at the final exhaust. Because this relation is independent of the density of the intake air it can be used as pointed out in the following for developing a rational procedure for correcting engine performance for changes in ambient conditions.

The following discussion is confined to operation of engines under conditions such that the pressure at the final exhaust is the same as the pressure at the intake.

Basis for Comparing Performance of Internal-Combustion Engines at Different Ambient Conditions – The fundamental objective of correcting the performance of internal-combustion engines for changes in ambient conditions is to make possible comparison of the performance of an engine at constant speed and throttle setting but at different pressures and temperatures of the atmosphere at the intake and final exhaust. In operation of a compression-ignition engine at constant speed and throttle setting the quantity of fuel injected per unit time is constant. In contrast with this, at constant speed and throttle setting a spark-ignition engine equipped with a carburetor operates at approximately a constant fuel-air ratio depending on ability of the carburetor to maintain this ratio constant over the range of operating conditions. The following discussion will be confined principally to the development of a method for correcting the performance of compression-ignition engines for changes in ambient temperatures and pressures[18]. However, some of the relations and methods are applicable also to correction of the performance of spark-ignition engines equipped with carburetors.

Variation of Indicated Horsepower of Compression-Ignition Engines with Density of Intake Air at Constant Speed and Throttle Setting – The indicated horsepower of any internal combustion engine may be computed from the following basic relation:

$$i = eF, \tag{6}$$

in which i = ihp,
e = indicated efficiency expressed as ihp-hr per lb of fuel,
F = fuel used, lb per hr

If the pressure at the intake equals that at the final exhaust, the indicated efficiency is a function of only the fuel-air ratio as demonstrated in a preceding section of this report. Under such conditions the value of e is given by the relation:

$$e = \phi(f), \tag{7}$$

in which f = fuel-air ratio, lb per lb,
ϕ = any function expressing the relation between e and f (this function was linear for the engines discussed in this report).

The value of f may be determined from the relation:

$$f = \frac{F}{Qv\rho}, \tag{8}$$

n which Q = piston displacement of the engine on the intake stroke, cu ft per hr,
v = volumetric efficiency expressed as a ratio,
ρ = density of intake air at temperature and pressure existing in intake, lb per cu ft.

If subscripts 1 and 2 are used to denote two different conditions of operation in so far as density of intake air is concerned, then the ratio of indicated horsepower at condition 1 to indicated horsepower at condition 2 will be given by the following relation which is applicable for all internal-combustion engines and for all operating conditions:

$$\frac{i_1}{i_2} = \frac{e_1F_1}{e_2F_2}. \tag{9}$$

In the compression-ignition engine operated at a constant speed and throttle setting $F_1 = F_2$ and Equation (9) becomes

$$\left(\frac{i_1}{i_2}\right)_F = \frac{e_1}{e_2}, \tag{10}$$

in which

$\left(\frac{i_1}{i_2}\right)_F$ = ratio of i_1 to i_2 when F is maintained constant

If the relation between e and f is linear, as it was over a wide range of fuel-air ratios for the engines examined in

[17] See p. 252: "The Internal Combustion Engine," by C. F. Taylor and E. S. Taylor, International Textbook Co., Scranton, Pa., 1938.
[18] Changes in relative humidity have not been considered because of their minor effect. [3, 4]

this report, the value of e_1 is given by Equation (4) and Equation (10) can be expressed as follows:

$$\left(\frac{i_1}{i_2}\right)_F = \frac{e_0 - af_1}{e_2} = \frac{e_0 - \left(\dfrac{aF_1}{Qv_1\rho_1}\right)}{e_2}. \qquad (11)$$

For a given compression-ignition engine operated at given speed and throttle setting

$$\frac{aF_1}{Q} = \frac{aF_2}{Q}. \qquad (12)$$

If the value of $\dfrac{aF_2}{Q}$ computed from the relation

$$e_2 = e_0 - \frac{aF_2}{Qv_2\rho_2} \qquad (13)$$

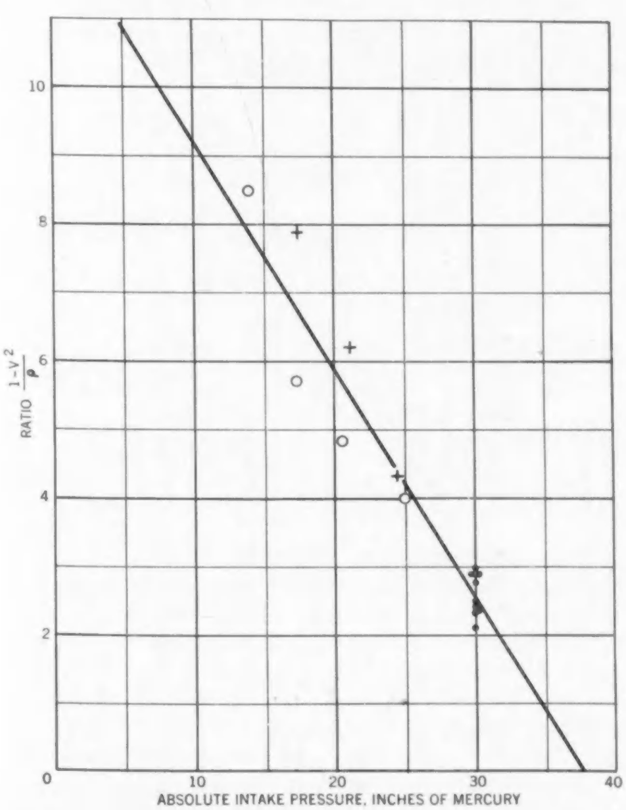
is substituted for $\dfrac{aF_1}{Q}$ in Equation (11), then

$$\left(\frac{i_1}{i_2}\right)_F = \left(\frac{e_0}{e_2} - 1\right)\left(1 - \frac{v_2\rho_2}{v_1\rho_1}\right) + 1. \qquad (14)$$

Multiplying both sides of Equation (14) by i_2 and substituting for $\dfrac{i_2}{e_2}$ its value F_2, then

$$\left(i_1\right)_F = \left(e_0F_2 - i_2\right)\left(1 - \frac{v_2\rho_2}{v_1\rho_1}\right) + i_2. \qquad (15)$$

In applying Equation (14) or (15), when the differences between ρ_1 and ρ_2 are large, it is necessary to know the relation between the volumetric efficiency v and the pressure and temperature of the intake, because v is affected by these factors[6]. Examination of the results of Moore and Collins[6] has shown that a linear relation is obtained by plotting $\dfrac{1-v^2}{\rho}$ as ordinates against the absolute pressure at the intake P as abscissas. The relation indicated by the



■ Fig. 6 – Variation of $\dfrac{1-v^2}{\rho}$ with absolute intake pressure, indicated by results of Moore and Collins[6]

Table 4 – Comparison of Volumetric Efficiency Observed by Moore and Collins[6] with Volumetric Efficiency Computed from Empirical Equations

Absolute Pressure in Intake in. of hg	Temperature of Intake F	Temperature of Intake R	Density of Intake lb per cu ft	Volumetric Efficiency, Expressed as a Ratio: Observed	Calculated	Relative Error in Calculated Value, %
Constant Pressure, Variable Temperature						
29.92	256	716	0.0555	0.94	0.92	−2.1
29.92	148	608	.0653	0.90	0.91	+1.1
29.92	70	530	.0750	0.89	0.90	+1.1
29.92	26	486	.0817	0.90	0.89	−1.1
29.92	−3	457	.0870	0.89	0.88	−1.1
Constant Temperature, Variable Pressure						
30.2	82	542	.0735	0.91	0.90	−1.1
24.9	82	542	.0610	0.86	0.87	+1.2
20.6	82	542	.0505	0.87	0.84	−3.4
17.4	82	542	.0426	0.87	0.84	−3.4
14.2	82	542	.0348	0.84	0.85	+1.2
Variable Temperature and Pressure						
29.92	66	526	.0746	0.88	0.90	+2.3
24.5	43	503	.0646	0.85	0.85	0.0
20.9	25	485	.0572	0.80	0.82	+2.5
17.6	9	469	.0498	0.78	0.82	+5.1

data of Moore and Collins[6] is shown in Fig. 6 and may be expressed[19] by the equation

$$v^2 = 1 - 12.6\rho + 0.334P\rho, \qquad (16)$$

in which P = absolute intake pressure, in in. of hg and the other terms have the significance previously stated.

If the value of ρ for air, determined from the relation

$$\rho = \frac{1.327P}{T}, \qquad (17)$$

in which T = absolute temperature deg R, is substituted in Equation (16) then

$$v^2 = 1 - \frac{16.7P}{T} + \frac{0.443P^2}{T}. \qquad (18)$$

A comparison of actual volumetric efficiency observed by Moore and Collins[6] with volumetric efficiency, computed from Equations (16) or (18), is shown in Table 4 and indicates reasonably close agreement.

The foregoing method of correlating data on volumetric efficiency is presented because it may also be applicable to other engines. Although Equation (16) is empirical, it appears to have some theoretical basis, because at any given pressure $v = 1$ when $\rho = 0$. Such a limiting condition seems reasonable because the inertia of a given volume of air becomes less as the density is reduced, and consequently a given force acting for a given time will impart a greater velocity to a given volume when the density is low. Although further discussion of the subject is beyond the scope of the present paper, it is believed that this and other boundary conditions indicated by Equation (16) should receive further theoretical consideration.

If the relation between volumetric efficiency and density of intake mixture is not known and it is assumed that $v_1 = v_2$, then errors in the calculated value of $\left(\dfrac{i_1}{i_2}\right)_F$

[19] See "Graphical and Mechanical Computation. Part II – Experimental Data," by Joseph Lipka, John Wiley & Sons, Inc., New York, N. Y., 1921.

Table 5 – Comparison of Horsepower Ratios Observed by Moore and Collins[6] with Ratios Computed from Rational and from Empirical Equations

									$\left(\frac{i_1}{i_2}\right)_F$					$\left(\frac{b_1}{b_2}\right)_F$				
Absolute Pressure at Intake and at Final Exhaust, in. hg	Temperature of Intake F	Temperature of Intake R	Density of Intake, lb per cu ft	Fuel Consumption, lb per cycle	Fuel-Air Ratio,[a] lb per lb	Ihp-Hr per Lb of Fuel,[a] e	Mechanical Efficiency,[a] m	Bph-Hr per Lb of Fuel,[a] ω	Actual[a]	Calc. from Eq. (14)[b]	Error, %	Calc. from Eq. (14)[c] with $v_1 = v_2$	Error, %	Actual[a]	Calc. from Eq. (30)[b]	Error, %	Calc. from Eq. (32)[c]	Error, %
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
								Constant Pressure, Variable Temperature										
29.92	256	716	0.0555	4.0×10^{-4}	0.096	1.96	0.69	1.35	1.29	1.30	+.8	1.32	+2.3	1.42	1.43	+0.7	1.38	−2.9
29.92	204	664	0.0598	4.0×10^{-4}	0.092	2.16	0.72	1.56	1.17	1.19	+1.7	1.21	+3.4	1.23	1.25	+1.6	1.23	0.0
29.92	148	608	0.0653	4.0×10^{-4}	0.085	2.33	0.74	1.72	1.09	1.10	+.9	1.11	+1.8	1.12	1.14	+1.8	1.12	0.0
29.92	116	576	0.0690	4.0×10^{-4}	0.078	2.46	0.75	1.85	1.03	1.05	+1.9	1.06	+2.9	1.04	1.07	+2.9	1.06	+1.9
29.92	70	530	0.0750	4.0×10^{-4}	0.074	2.53	0.76	1.92	1.00	1.00	[d]	1.00	[d]	1.00	1.00	[d]	1.00	[d]
29.92	45	505	0.0787	4.0×10^{-4}	0.072	2.61	0.77	2.01	.97	.97	0.0	.97	0.0	.96	.97	+1.0	.97	+1.0
29.92	26	486	0.0817	4.0×10^{-4}	0.068	2.66	0.77	2.05	.95	.95	0.0	.94	−1.1	.94	.93	−1.1	.94	0.0
29.92	−3	457	0.0869	4.0×10^{-4}	0.065	2.65	0.77	2.04	.95	.91	−4.2	.90	−5.3	.94	.88	−6.4	.90	−4.3
								Constant Temperature, Variable Pressure										
30.2	82	542	0.0735	3.0×10^{-4}	0.057	2.95	0.73	2.15	1.00	1.00	[d]	1.00	[d]	1.00	1.00	[d]	1.00	[d]
24.9	82	542	0.0610	3.0×10^{-4}	0.070	2.73	0.72	1.97	1.08	1.12	+3.7	1.11	+2.8	1.09	1.14	+4.6	1.11	+1.8
20.6	82	542	0.0505	3.0×10^{-4}	0.084	2.31	0.70	1.62	1.28	1.32	+3.1	1.29	+0.8	1.33	1.40	+5.3	1.31	−1.5
17.4	82	542	0.0425	3.0×10^{-4}	0.103	1.83	0.64	1.17	1.61	1.64	+1.9	1.61	0.0	1.84	1.88	+2.2	1.73	−6.0
								Variable Temperature and Pressure										
29.92	66	522	0.0761	4.0×10^{-4}	0.075	2.51	0.76	1.91	1.00	1.00	[d]	1.00	[d]	1.00	1.00	[d]	1.00	[d]
24.5	43	503	0.0646	4.0×10^{-4}	0.092	2.18	0.73	1.59	1.15	1.20	+4.3	1.15	0.0	1.20	1.25	+4.2	1.17	−2.5
20.9	25	485	0.0572	4.0×10^{-4}	0.109	1.89	0.71	1.34	1.33	1.42	+6.8	1.33	0.0	1.43	1.53	+7.0	1.37	−4.2
17.6	9	469	0.0498	4.0×10^{-4}	0.129	1.55	0.67	1.04	1.62	1.74	+7.4	1.63	+0.6	1.84	1.99	+8.2	1.76	−4.3

[a] Computed from data scaled from curves of Moore and Collins[6].
[b] Rational equation.
[c] Empirical equation.
[d] Assumed as reference.

may result if the difference between ρ_1 and ρ_2 is large. However, the data of Moore and Collins[6] indicate that, for the engine tested by them, large errors in $\left(\frac{i_1}{i_2}\right)_F$ would not occur by assuming $v_1 = v_2$. This is demonstrated by the results given in columns 10 to 14 of Table 5, which show the error in $\left(\frac{i_1}{i_2}\right)_F$ computed from Equation (14): (a) when the relation between v, ρ, and P was known (rational equation); and (b) when it was assumed that $v_1 = v_2$ (empirical equation). The errors were determined by comparing the computed values with the actual values calculated from the data of Moore and Collins[6]. Frequently the error in $\left(\frac{i_1}{i_2}\right)_F$ computed from the empirical equation was less than that computed from the rational equation. This appears to be fortuitous because the errors in the assumption that $v_1 = v_2$ compensate for the deviation of the values of e from the average relation given by Equation (5). However the applicability of this empirical relation in correlating data obtained in tests of other engines should receive further consideration because this relation is simpler and requires a knowledge of fewer factors than the rational equation.

Variation of Brake Horsepower of Compression-Ignition Engines with Density of Intake Air at Constant Speed and Throttle Setting – Brake horsepower may be computed from the relation:

$$b = mi \tag{19}$$

in which b = brake horsepower,

m = mechanical efficiency expressed as a ratio.

The relation between brake horsepower at different intake conditions is given by the expression

$$\frac{b_1}{b_2} = \frac{m_1 i_1}{m_2 i_2}, \tag{20}$$

or at a constant quantity of fuel per unit time

$$\left(\frac{b_1}{b_2}\right)_F = \frac{m_1}{m_2}\left(\frac{i_1}{i_2}\right)_F. \tag{21}$$

Substituting the value of $\left(\frac{i_1}{i_2}\right)_F$ from Equation (14) or (15)

$$\left(\frac{b_1}{b_2}\right)_F = m_1\left(\frac{e_0}{w_2} - \frac{1}{m_2}\right)\left(1 - \frac{v_2\rho_2}{v_1\rho_1}\right) + \frac{m_1}{m_2}, \tag{22}$$

or

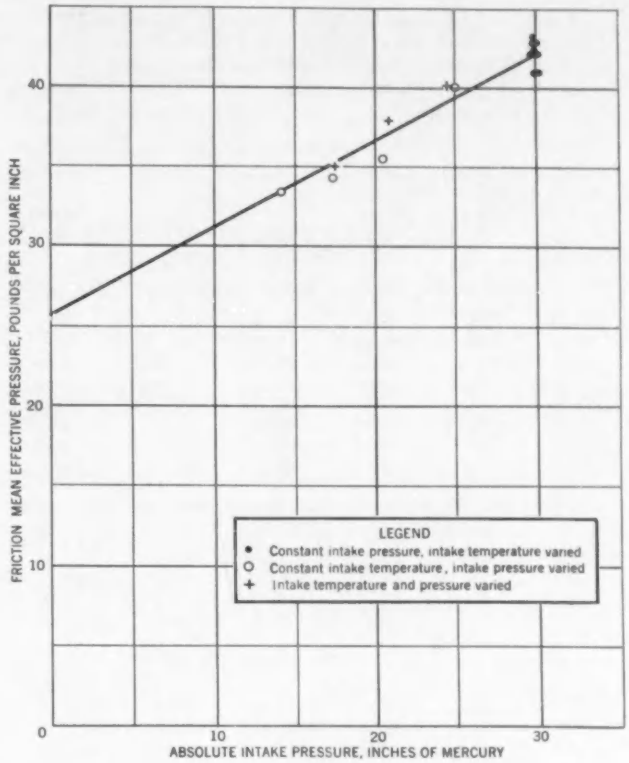
$$\left(b_1\right)_F = m_1\left(e_0 F_2 - \frac{b_2}{m_2}\right)\left(1 - \frac{v_2\rho_2}{v_1\rho_1}\right) + \frac{m_1 b_2}{m_2}, \tag{23}$$

in which w = bhp-hr per lb of fuel.

In applying Equations (22) and (23) it is necessary to know the relation between m and ρ as well as between v and ρ.

The variation of mechanical efficiency with intake density depends on many factors and therefore a general relation between m and ρ cannot be given. However, in the tests of Moore and Collins[6] data were obtained on the friction mean effective pressure (fmep) at different intake densities. Analysis of these data has disclosed that the variation of fmep with absolute intake pressure might be represented satisfactorily by a straight line, as shown in Fig. 7. Such a linear relation has been observed in tests of another engine[20] and therefore this correlation may be applicable to many engines. In fact consideration of the

[20] See NACA Technical Report No. 262, 1927: "Friction of Aviation Engines," by S. W. Sparrow and M. A. Thorne.



■ Fig. 7 – Variation of friction mean effective pressure with absolute intake pressure, indicated by data of Moore and Collins[6]

relation shown in Fig. 7 suggests that it may have some theoretical significance because the intercept on the y-axis may be mechanical friction and the difference between this intercept and any value on the curve may represent the pumping loss at the pressure in question.

The equation of the straight line[19] shown in Fig. 7 is

$$y' = 25.6 + 0.55P, \tag{24}$$

in which y' = friction mean effective pressure, psi. A comparison of fmep calculated from Equation (24) with observed fmep is shown in Table 6 and indicates that the equation represents the data with reasonable precision. Equation (24) applies only when the pressure at the intake equals the pressure at the final exhaust.

If the fmep can be expressed as a function of absolute intake pressure, then the friction horsepower-hours per pound of fuel, y, can be calculated from the relation:

$$y = \frac{y'Q}{60 \times 33{,}000F}. \tag{25}$$

The value of y is given also by the relation

$$y = \frac{(1-m)w}{m} = \frac{(1-m)b}{mF}. \tag{26}$$

Eliminating y from equations (25) and (26) and solving for b then

$$b = \frac{my'Q}{60 \times 33{,}000\,(1-m)}. \tag{27}$$

From equation (27) it follows that

$$\frac{b_1}{b_2} = \frac{m_1 y'_1\,(1-m_2)}{m_2 y'_2\,(1-m_1)}. \tag{28}$$

Letting $\frac{y'_1}{y'_2} = R$ and solving equation (28) for $\frac{m_1}{m_2}$ then

$$\frac{m_1}{m_2} = \frac{(1-m_1)}{R}\,\frac{b_1}{b_2} + m_1 \tag{29}$$

Substituting for $\frac{m_1}{m_2}$ in Equation (22), its value given by

Table 6 – Comparison of Friction Mean Effective Pressure Observed by Moore and Collins[6] with Friction Mean Effective Pressure Computed from Empirical Equation

Absolute Intake Pressure, in. hg	Temperature of Intake, F	Density of Intake lb per cu ft	Friction Mean Effective Pressure, psi: Observed	Calculated	Error in Calculated Value, %
Constant Intake Pressure, Intake Temperature Varied					
29.92	256	.0555	42.0	42.1	+0.2
29.92	204	.0598	43.1	42.1	−2.3
29.92	148	.0653	42.8	42.1	−1.6
29.92	116	.0690	42.2	42.1	−0.2
29.92	70	.0750	40.8	42.1	+3.2
29.92	45	.0787	42.0	42.1	+0.2
29.92	26	.0817	43.1	42.1	−2.3
29.92	−3	.0870	42.0	42.1	+0.2
Constant Intake Temperature, Intake Pressure Varied					
30.2	82	.0735	40.8	42.2	+3.4
24.9	82	.0610	40.0	39.3	−1.8
20.6	82	.0505	35.4	36.9	+4.2
17.4	82	.0426	34.3	35.2	+2.6
14.2	82	.0348	33.3	33.4	+0.3
Intake Pressure and Temperature Varied					
29.92	66	.0761	42.2	42.1	−0.2
24.5	43	.0646	40.1	39.1	−2.5
20.9	25	.0572	37.9	37.1	−2.1
17.6	9	.0498	35.0	35.3	+0.9

Equation (29) and solving for $\left(\frac{b_1}{b_2}\right)_F$ the following relation is obtained

$$\left(\frac{b_1}{b_2}\right)_F = \frac{\frac{m_1 e_0}{w_2}\left(\frac{v_1\rho_1}{v_2\rho_2} - 1\right) + m_1}{\frac{v_1\rho_1}{v_2\rho_2} - \frac{1 - m_1}{R}}. \quad (30)$$

If condition 1 is assumed to be the standard condition, then Equation (30) permits calculation of the ratio of brake horsepower at the two conditions from a measurement of fuel consumption and brake horsepower at condition 2 provided that m_1, R, e_0, and $\frac{v_1}{v_2}$ are known. If these factors can be evaluated, Equation (30) can be applied over a wide range of intake densities with the further restrictions that: (a) the variation of indicated efficiency with fuel-air ratio is linear; and (b) the pressure at the intake equals the pressure at the final exhaust.

If it is desired to compute the horsepower at condition 2 from the fuel consumption and horsepower at the standard condition, 1, then Equation (30) may be rewritten:

$$\left(\frac{b_2}{b_1}\right)_F = \frac{1}{m_1}\left(1 - \frac{1-m_1}{R}\right) - \left(\frac{e_0}{w_1} - \frac{1}{m_1}\right)\left(\frac{v_1\rho_1}{v_2\rho_2} - 1\right). \quad (31)$$

Equation (30) is necessarily complex because it attempts to correct for all factors affecting $\left(\frac{b_1}{b_2}\right)_F$. If the difference between ρ_1 and ρ_2 is not great, as is frequently the case in correcting the performance of engines for ordinary variations in atmospheric conditions, then two simplifying assumptions may be made, namely that $v_1 = v_2$ and $m_1 = m_2$. With these assumptions, Equation (22) becomes

$$\left(\frac{b_1}{b_2}\right)_F \doteq \left(\frac{m_1 e_0}{w_2} - 1\right)\left(1 - \frac{\rho_2}{\rho_1}\right) + 1. \quad (32)$$

If ρ is expressed in terms of P and T as shown in Equation (17) then,

$$\left(\frac{b_1}{b_2}\right)_F \doteq \left(\frac{m_1 e_0}{w_2} - 1\right)\left(1 - \frac{P_2 T_1}{P_1 T_2}\right) + 1. \quad (33)$$

A comparison of values of $\left(\frac{b_1}{b_2}\right)_F$ observed by Moore and Collins with values calculated from Equation (30) and from the simplified Equations (32) or (33) is shown in columns 15 to 19 of Table 5. The agreement between observed and calculated values is generally good, although in some instances the errors in the values calculated from Equation (30) were relatively large. As pointed out previously, these large errors occurred because the test results were not represented accurately by the linear relation between indicated efficiency and fuel-air ratio. The largest error, 8.2%, occurred in correcting the results of a test in which the fuel-air ratio was approximately 0.13 lb per lb, a condition that has no practical value.

It is of interest to note that, throughout the entire range of densities, the approximate empirical equation on the average represents the ratio, $\left(\frac{b_1}{b_2}\right)_F$, with greater precision than the rational equation. As stated previously, this may be true only of the engine tested by Moore and Collins, although it is possible that the approximate equation may also be satisfactory for other engines.

Further experimental work on other engines would be required to determine the equation most suitable for the correction as well as the most suitable form of the equation. It is believed, however, that the foregoing considerations may offer a rational basis for the formulation of equations and methods for correcting to standard intake conditions the performance of compression-ignition engines.

Correction Formula for Constant Fuel-Air Ratio Engines

Although it is not within the scope of this paper to discuss the correction of the performance of gasoline engines for changes in intake conditions, nevertheless it is of interest for comparative purposes to show the ratio, $\left(\frac{b_2}{b_1}\right)_f$, assuming a constant fuel-air ratio and a variation of friction horsepower with intake pressure similar to that indicated by the data of Moore and Collins[6].

The general expression for the brake horsepower of an internal-combustion engine is

$$b = meF. \quad (34)$$

Substituting in Equation (34) the value of F given by Equation (8),

$$b = meQv\rho f. \quad (35)$$

From Equation (35) it follows that

$$\frac{b_2}{b_1} = \frac{m_2 e_2 v_2 \rho_2 f_2}{m_1 e_1 v_1 \rho_1 f_1}. \quad (36)$$

At a constant fuel-air ratio $e_1 = e_2$ and $f_1 = f_2$ therefore

$$\left(\frac{b_2}{b_1}\right)_f = \frac{m_2 v_2 \rho_2}{m_1 v_1 \rho_1}. \quad (37)$$

Substituting in Equation (37) the value of $\frac{m_2}{m_1}$ given by Equation (29) and solving for $\left(\frac{b_2}{b_1}\right)_f$, then

$$\left(\frac{b_2}{b_1}\right)_f = \frac{1}{m_1}\,\frac{v_2\rho_2}{v_1\rho_1} - \frac{1}{R}\left(\frac{1}{m_1} - 1\right). \quad (38)$$

Letting $\left(\frac{1}{m_1} - 1\right) = c'$ then Equation (38) becomes

$$\left(\frac{b_2}{b_1}\right)_f = \left(1 + c'\right)\frac{v_2\rho_2}{v_1\rho_1} - \frac{c'}{R}. \qquad (39)$$

Equation (39), which has a rational basis, is similar to the empirical relation

$$\frac{b_2}{b_1} = \left(1 + c\right)\left(\frac{\rho_2}{\rho_1}\right) - c \qquad (40)$$

given by Taylor and Taylor[8] and Gagg and Farrar[5] for representing the comparative performance of engines at different atmospheric densities. The similarity between these equations is even more striking when it is realized: (a) that the value of c can be estimated from the mechanical efficiency at sea level (condition 1 in the Equations (39) and (40); and (b) that the values of R and of $\frac{v_2}{v_1}$ always are in the vicinity of unity.

It is of interest to compare qualitatively the foregoing rational relations with existing relations for correcting the power output of gasoline engines to standard intake conditions. From Equation (37) it is apparent that $\left(\frac{b_2}{b_1}\right)_f$ can be computed when the values of ρ, v and m at the two conditions are known. Obviously, ρ is a function of P and T only. Furthermore the data analyzed in this paper indicate that, for a given engine, v is related to P and T and m is related to P. It appears therefore that a knowledge of the ambient pressure and temperature, P and T, is sufficient to determine the ratio $\left(\frac{b_2}{b_1}\right)_f$ when a gasoline engine is operated at a constant speed and throttle setting. The validity of this reasoning is indicated by the fact that existing formulas for correcting for changes in intake conditions the performance at constant speed and throttle setting of spark-ignition engines equipped with carburetors contain only P and T as factors. In contrast with this, formulas for correcting to standard intake conditions the performance of compression-ignition engines at constant speed and throttle setting require a knowledge of the fuel consumption per unit time in addition to pressures and temperatures at the two intake conditions. This additional requirement is imposed because indicated efficiency varies with intake density when a compression-ignition engine is operated at a constant throttle setting whereas, in the spark-ignition engine equipped with a carburetor, indicated efficiency is not affected greatly by density of the intake, provided the carburetor maintains a reasonably constant fuel-air ratio.

■ Summary

A study of published data on the performance of compression-ignition engines under a variety of operating conditions has shown that the relation of indicated efficiency to fuel-air ratio is a fundamental correlation of great assistance in analyzing, interpreting, and generalizing test results. According to performance data for four different compression-ignition engines, the relation appeared to be linear throughout a wide range of fuel-air ratios. Furthermore, published results of tests of a compression-ignition engine at different intake and exhaust conditions indicated that, within the range of conditions investigated, the relation between indicated efficiency and fuel-air ratio was not affected significantly by changes in intake pressure or temperature, provided the pressure at the intake was the same as that at the final exhaust. Published data obtained in tests in which these pressures were not the same appeared to indicate that the relation of indicated efficiency to fuel-air ratio was different under such conditions. However, further consideration of these data revealed that the fuel-air ratio determined by metering the air and measuring the fuel was not representative of the ratio of fuel to the medium present in the cylinder during the working cycle owing to the difference between exhaust and intake pressures which, during the period of valve overlap, permitted air to flow out of or exhaust gas to flow into the cylinder, depending on which pressure was greater.

Because of its basic nature and because it did not appear to be affected by changes in ambient conditions, the relation between indicated efficiency and fuel-air ratio offered a rational basis for correcting the performance of compression-ignition engines for changes in ambient conditions. Accordingly, rational equations were derived for computing either the indicated or brake horsepower at some standard condition of intake temperature and pressure from the fuel consumption and horsepower determined at any other intake temperature and pressure. Simplified approximate equations are also presented. A comparison of horsepower ratios computed from the equation with horsepower ratios determined experimentally indicates satisfactory agreement throughout a wide range of intake conditions.

In applying the more complex rational equation, a knowledge of the variation with intake conditions of mechanical efficiency and volumetric efficiency is required. Accordingly, published data were examined, and satisfactory methods for correlating the data developed. Although these methods are empirical they appear to have some theoretical significance and may warrant further study.

The fundamental objectives in correcting the performance of internal-combustion engines for changes in intake conditions at constant speed and throttle setting are discussed briefly. It is pointed out that operation of compression-ignition engines at constant speed and throttle setting is characterized by a constant rate of fuel input and therefore comparisons should be made on this basis. In contrast with this, at constant speed and throttle setting, a spark-ignition engine equipped with a carburetor approaches operation at a constant fuel-air ratio and therefore with such an engine comparisons should be made on this basis. Formulas applying to this latter condition are discussed briefly.

■ Acknowledgments

The author is indebted to A. C. Fieldner, chief, Technologic Branch; and Wilbert J. Huff, consulting explosives chemist, Explosives Division, Bureau of Mines, who made possible the completion of this work.

Particular acknowledgment is made to John C. Holtz, senior explosives engineer, High Explosives Research Section, Explosives Division, Bureau of Mines, for his continued interest and encouragement and for his valuable advice, criticisms, and suggestions.

The author acknowledges also the helpful comments and criticisms of Dennis W. Keef and Rudolf Kudlich of the Bureau of Mines and the valuable assistance of C. K. White, Explosives Division, Bureau of Mines, who checked certain computations.

VIBRATION CHARACTERISTICS of Three- and Four-Blade Propellers

DURING the early development of aircraft engine-propeller combinations numerous propeller failures occurred which were directly traceable to excitation originating in the aerodynamic interferences existing between the propeller and the adjacent airplane structure. Alleviation of this earliest known propeller vibration was achieved readily by reducing the absolute magnitude of the interference excitation and, from these experiences, certain basic principles of propeller location were evolved which have been used quite successfully for the installation of two- and three-blade propellers. It has been known for some time, however, that these interference conditions were not eliminated, but the problem of aerodynamic excitation was relatively of much less importance than that of reducing the propeller excitation potentialities of the aircraft engine. Since this last problem has proved to be a difficult one, it is not surprising that aerodynamic excitation has received much less attention than it deserved.

■ Engine Excitation

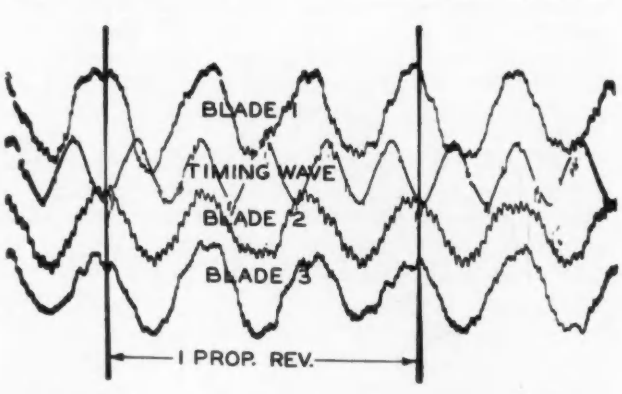
Engine-excited propeller vibration, whether caused directly by the numerous components of gas and inertia torque appearing about the propeller-shaft axis or indirectly by engine-propeller whirls set up by these same engine torques appearing about other axes, has not changed sufficiently in frequency or amplitude on three- and four-blade propellers used in combination with the high-output engines of today to warrant any very detailed discussion within the scope of this paper. One exception to this statement could be made in that the origin of 2½-order engine-propeller whirl on 14-cyl double-row radial engines, first

[This paper was presented at the National Aircraft Meeting of the Society, Washington, D. C., March 13, 1941.]

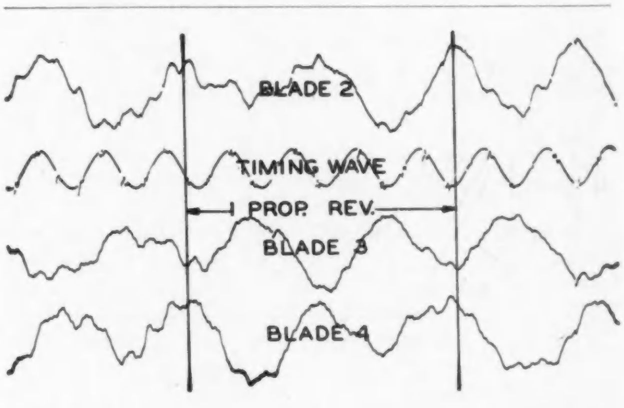
ALTHOUGH the problems involved in developing a satisfactory engine-propeller airplane installation using propellers of four or more blades are admittedly more difficult than those encountered with a three-blade propeller, Mr. Guerke points out, a satisfactory installation can be made "if the airplane designer will assume the responsibility of reducing propeller interference effects." This, of course, may be very difficult if not impossible to accomplish in "pusher" installations, he qualifies. Reporting that two such installations have been made in a normal tractor four-blade installation without recourse to odd blade spacing, he contends that successful installations also can be accomplished in propellers of five or more blades under similar conditions.

Dual-rotation propellers, he believes, will introduce additional propeller design problems of a vibration nature which may require certain compromises in diameter, blade construction, and engine gear ratios before satisfactory installations can be made. Mr. Guerke opines, however, that these compromises can be made without sacrificing airplane performance.

In the technical discussion that precedes the foregoing conclusions, Mr. Guerke takes up engine excitation; aerodynamic excitation; aerodynamically excited three-blade propeller vibration; aerodynamically excited four-blade propeller vibration; methods of reducing or avoiding aerodynamically excited stress; dual rotation; and effects of propeller isolation on aerodynamically excited stress.



■ Fig. 1 – Illustration of Group 1, 3rd-order propeller vibration strain on three-blade propeller



■ Fig. 2 – Illustration of Group 4, 2nd-order propeller vibration strain on four-blade propeller. (Note 180-deg relationship between strain on blades three and four and in-phase relationship between strains on blades two and four)

for HIGH-OUTPUT Engines

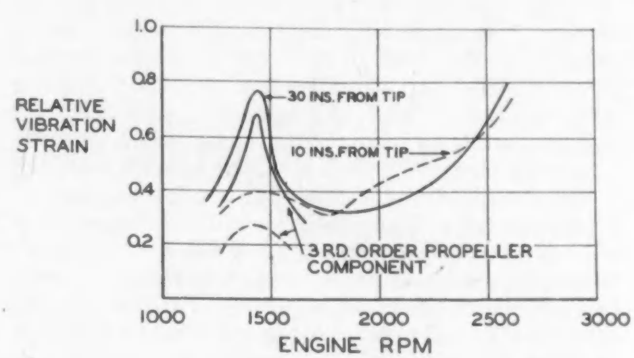
by RALPH M. GUERKE

Project Engineer, Curtiss Propeller Division, Curtiss-Wright Corp.

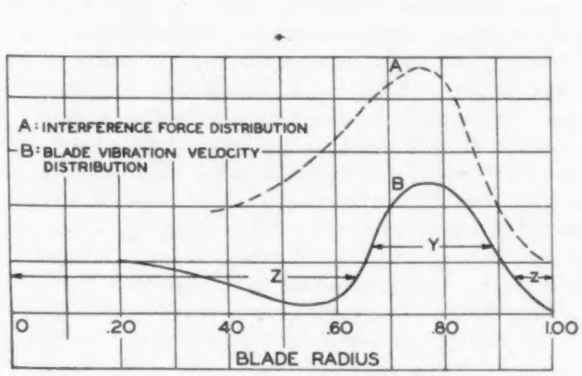
noted in 1937 by G. P. Bentley, and which appeared at certain critical speeds with all propellers or clubs, has been traced satisfactorily and appropriate corrective measures taken to reduce it in those cases where propeller stresses are unsatisfactorily high. The basic excitation for this whirl originated in the $3\frac{1}{2}$-order firing forces from the forward bank of seven cylinders which acted to deflect the support of the forward main crankshaft bearing at a major frequency of $2\frac{1}{2}$-order engine. Since the propeller-shaft rear was also supported by this same bearing, it also suffered deflection relative to the engine frame and a point on the propeller-shaft axis was constrained to follow an orbit in reverse propeller direction of rotation at the exciting frequency of $2\frac{1}{2}$-order engine. Since the engine, propeller, and airplane are vital parts of this system, a change in the mass or rigidity of any of these three components or a change in the connecting elasticity will change the natural frequencies of the several modes of vibration. However, it was found that the natural frequencies of the system were determined mainly by the propeller mass and flexibility. Reduction of the $2\frac{1}{2}$-order whirl forcing function was accomplished by the engine manufacturer in isolating the propeller shaft from the crankshaft through a coupling incapable of transmitting the crankshaft bearing deflections and with this isolation, a reduction in propeller stress of from 40% to 70% was noted.

Aerodynamic Excitation

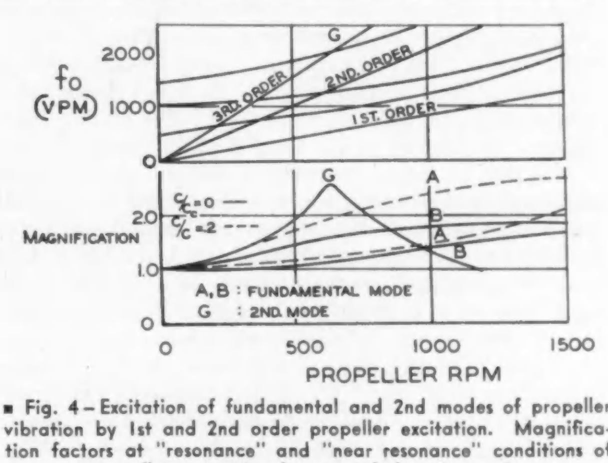
In the case of the single propeller of any number of blades, aerodynamic excitation results from operation of the propeller in a region of non-axial airflow, and from passage of the blade tips through any disturbed region of air whether that region is caused by a part of the aircraft structure or by another propeller. While interference with the slipstream of an adjacently operating propeller might be considered as the strongest excitation of this class, it is infrequently encountered because of early recognition of the resulting severe propeller vibration. Operation of propellers on concentric shafts as in dual-rotation, however, also will lead to severe interference conditions caused by



■ Fig. 5 – Vibration strain at two blade locations in three-blade hollow-steel propeller, 12 ft in diameter, operating in vertical airflow intake test house



■ Fig. 3 – Excitation of blade vibration mode by aerodynamic interference – blade vibration excited at *Y*, damped at *Z*



■ Fig. 4 – Excitation of fundamental and 2nd modes of propeller vibration by 1st and 2nd order propeller excitation. Magnification factors at "resonance" and "near resonance" conditions of propeller operation for several damping ratios

the passage of the blades of one propeller past those of the other. Next in order of severity may be placed those interference effects resulting from passage of the propeller tips through the "down-wash" air from the wing (in the case of the "pusher" propeller) and through the accelerated region of air near the fuselage. While not classified as an "interference" condition, angular airflow through the propeller due to installation conditions or airplane maneuvers constitutes an important source of excitation.

The frequencies of vibration which can be excited in a propeller by these various aerodynamic forcing functions can be classified into four groups which differ mainly in the engine motions associated with them, the types of propeller vibration excited and the damping involved. In Group 1 are placed all vibrations whose frequencies are integral multiples of the number of blade tip interferences per propeller revolution such as 3, 6, 9, and so on, times propeller speed in the case of the three-blade propeller. Blade vibration in this group is of the "in-phase" type in that the stress on all blades is of the same sense and attains a maximum at the same instant. See Fig. 1. Resulting vibratory moments from the blades balance on the propeller shaft. Vibratory forces, however, do not balance and act to produce fore-aft motions of the engine on its mount.

Group 2 Vibrations

Group 2 contains those vibrations whose frequencies are integral multiples of the number of tip interferences per propeller revolution ± 1, such as 2, 4, 5, 7, and so on, times propeller speed for the three-blade propeller. Propeller vibration of this group is distinguished by the fact that it is, in general, of an "out-of-phase" type in which the stress on any one blade is out-of-phase with that on the preceding or following blades by an angle equal to 360 deg divided by the number of blades and that vibratory forces acting on the propeller shaft caused by blade vibration completely balance each other, therefore tending to cause no engine translational vibration. Vibratory moments, however, do not balance and, as a consequence, engine whirl is excited. Excitation of the "minus one" frequencies coincides with the establishment of an engine whirl in which a point on the propeller shaft describes an orbit in the same direction as propeller rotation at a frequency one order of propeller speed higher than the vibration strain frequency in the propeller. Excitation of the "plus-one" frequencies coincides with the establishment of a pronounced "reverse" whirl in which the whirl takes place in a direction opposite to propeller rotation and at a frequency one order lower than that of the propeller vibration. The more nearly resonant the engine whirl, the higher the energy transfer to the engine vibration not only because engine whirl amplitudes will be larger, but the phase angle between the exciting vibratory blade moments and excited whirl will be optimum for that energy transfer. Natural frequencies in whirl of engine-airplane structures, however, are usually quite low in comparison with exciting frequencies and therefore whirl amplitudes are low as a consequence. Thus, while energy transfer to engine motion takes place, to an extent, the actual energy transferred cannot be very large. Since the reverse whirl frequencies are higher, greater energy transfer to the engine motion per propeller revolution should take place than during forward whirl. It is an experimental fact that such whirls are difficult to excite, and it is undoubtedly because of the more effective damping factors acting.

Group 3 contains but one frequency, that of first-order propeller excited as a forced vibration by any of the various interferences or during yawed flight. Resulting vibration forces balance at the propeller hub. Vibratory moments, however, do not balance and in the three- or more blade propeller result in a steady moment on the propeller shaft.

Group 4 contains the frequencies of those modes of propeller vibration which are completely independent of the engine. The outstanding characteristic of this group is that of complete balance at the propeller center of all forces and moments resulting from blade vibration. See Fig. 2.

Energy is introduced into the individual blade vibration at a time when, as in all interference excitations, the blade is in motion away from the aerodynamic forcing function. Maximum energy is introduced when the velocity is maximum, that is, when blade deflection caused by the interference occurs ¼ wave or 90 vibration deg after the blade has passed the interference. In terms of rotation of the blade, this would be $90/K$ deg measured in the direction of propeller rotation from the interference (where K is the order of the frequency as an integral multiple of propeller speed). All blades would, of course, follow the same vibration deflection pattern and would produce a "standing wave" on the propeller disc. Since this "standing wave" can be reproduced on a propeller disc at a frequency any integer times propeller speed, propeller resonance ought to occur theoretically if the number of strain cycles undergone by an individual blade in one revolution coincides with a natural frequency of that blade. However, this condition is approached as a limit for Group 4 vibration only as the number of blades approaches infinity. With a finite number of blades in the propeller Group 4 vibration can occur only at any frequency not equal to one, KB, or $KB \pm 1$ where K is any integer, B the number of blades, since only in these frequencies will the vibratory moments and forces from the individual blades balance at the center of the propeller. Thus, this form of vibration would occur at 2, 6, 10 . . . order propeller in a four-blade propeller; 2, 3, 7, 8 . . . order propeller in a five-blade; 2, 3, 4, 5, 6, 10, 11, 12, 13, 14 . . . order propeller in an eight-blade propeller, and so on.

The vibration stress amplitudes produced in a propeller depend as usual on the excitation amplitude, the damping present, and the proximity of the excitation frequency to a propeller natural frequency. The first variable of excitation amplitude is dependent on the extent to which the individual propeller blade projects into the region of accelerated airflow. However, in this connection consideration also must be given to the mode of vibration excited in the propeller, for it is conceivable that an interference excitation may be so general over the whole blade surface as to make it improbable that energy can be absorbed effectively into that mode from the interference excitation. Thus, although resonance is indicated by the coincidence of the propeller natural frequencies, in higher modes, with the exciting frequencies, only certain portions of the blade surface will be absorbing energy from the interference because of the coincidence of the vibration velocities on these blade sections with the exciting aerodynamic forces. These sections are marked Y on one mode of vibration in Fig. 3 while an excitation of the type mentioned might be represented by the dashed line. However, although energy is introduced into the blade vibration at sections Y, it will

be dissipated to the airstream at an accelerated rate at those sections marked Z in the same vibration mode because of the 180-deg shift in the phase relationship of the vibration velocities of that section to the exciting forces. For these reasons also, the lowest modes of vibration are much more easily excited by the interference excitation. This may be an important consideration in "pusher" installations where the "down-wash" air from the wing acts over a large part of the propeller disc and in dual-rotation where the interference between two blades is more evenly distributed over the lengths of the blades. In addition, since the interference effect is in the nature of an impulse, the excitation will possess a considerable number of harmonic components, each of which will, of course, tend to excite blade vibration.

With the exception of dual-rotation and unusual cases where several interferences occur on the propeller disc, energy is introduced into the blade vibration over a very limited segment of that area swept by the blade. Normal aerodynamic damping takes place over the remaining swept area and, since the effectiveness of this type of damping increases directly as the frequency, it is apparent that the amplitudes of vibration at the higher frequencies will suffer considerable attenuation from this effect alone. Where the vibrating system includes the engine, additional energy will be dissipated in the engine mount and adjacent airplane structure. While this energy dissipation may not be high enough to appreciably alter vibration levels in the airplane, it might constitute a necessary and significant portion of the energy entering the propeller. Elimination of the engine motion might be achieved in certain conditions by allowing inertia forces or moments originating in the engine to oppose the aerodynamically excited moments or forces. However, the penalty for such cancellation would result in higher propeller stresses. Of course, damping of the blade vibration by hysteresis effects in the blade material and mounting will also take place and these contributions to the control of vibration amplitudes should not be minimized. These last factors are not sufficiently large of themselves however, to be relied on exclusively for limitation of blade vibration stresses to reasonable values in those cases where aerodynamic and engine damping are either absent or greatly reduced in their effectiveness.

Coincidence of any aerodynamic excitation frequency with a natural frequency of the propeller constitutes fully as serious a problem as encountered in an engine-excited resonance inasmuch as the conditions are then optimum for maximum energy transfer from the exciting aerodynamic forces to the vibrating blade. Even proximity of these frequencies may constitute a serious problem since this "near-resonance" condition may exist over a wide range of propeller speeds. As an illustration Fig. 4 shows several fundamental and 2nd-mode blade frequencies, corrected for the effects of the centrifugal field, excited by 1st, 2nd, and 3rd-order propeller excitation. Sharply tuned resonant conditions will be noted at G where a 2nd-mode vibration is excited by 3rd-order propeller excitation. Resonance of the fundamental natural frequency with 1st-order propeller excitation can never occur nor can 2nd-mode vibration be excited by 2nd-order propeller excitation, although aerodynamic forces are amplified very appreciably at high propeller speeds because of the proximity of the exciting frequency to the blade fundamental frequency. The reason for the phenomenon is quite obvious upon considering the generally accepted formula for the correction of the non-rotating natural frequencies of a propeller for the effects of a centrifugal force field:

$$f_M = \sqrt{f_o^2 + CN^2}$$

where f_o is the non-rotating natural frequency
N propeller rotational frequency
C a constant dependent on the mode of vibration

Such a formula would indicate that, if resonance were to take place with any frequency an integer (K) times the propeller rotational frequency:

$$K^2 > C.$$

Since the constant C as determined by actual resonance encountered under test conditions appears to have an average value of 1.5 for the fundamental, 4.2 for the 2nd mode, 7.2 for the 3rd, and so on, it is readily apparent that 2nd-order propeller excitation cannot excite a 2nd mode; 3rd order propeller, a fourth, and so on.

The absolute influence of altitude and airplane forward speed on aerodynamically excited vibration stress is difficult to determine analytically inasmuch as the effects are dependent not only on the change in the accelerated airflow region with altitude and velocity, but on the change in the distribution of the aerodynamic load on the blade. Thus, the net effect may vary not only with the installation conditions, but on the airflow characteristics of the airplane under various flight attitudes, and each propeller-airplane combination must be considered a separate problem. To date, although no very appreciable altitude or velocity effect has been noted on 2nd-order propeller excited vibration stress, the possibility that such effects may occur must be considered until tests over a wide range of airplane installations and flight conditions have shown their absolute significance.

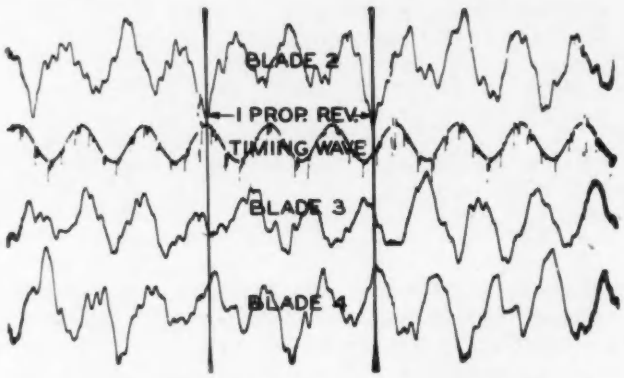
■ Three-Blade Propeller Vibration

Aerodynamically excited vibration in the single three-blade propeller falls into the first three of the four groups discussed. Vibration belonging to the fourth group or that which contains all vibration modes completely dissociated from the engine cannot be excited in this propeller. Thus, any resonance aerodynamically excited by a single interference condition will be subject to damping from the engine as well as to aerodynamic and hysteresis damping.

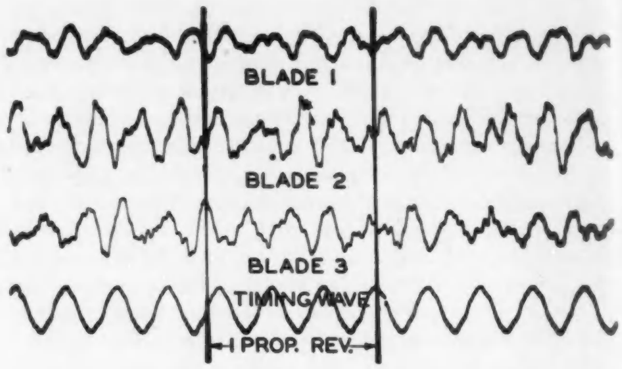
Group 1 vibration is present only in its lowest frequency, that of 3rd-order propeller, mainly because of the extreme effectiveness of aerodynamic damping in reducing amplitudes at the higher frequencies while the absolute amplitude of the vibration present is held to reasonable limits by the combined damping factors. See Fig. 5.

Group 2 vibration is often present in its lowest, "forward" whirl frequency, that of 2nd-order propeller, mainly because aerodynamic damping is less effective at low frequencies. See Fig. 6. Higher frequency vibration resonances caused by both "forward," such as 5th and 8th order propeller and "reverse" whirl components of engine whirl such as 4th and 7th order propeller frequencies, are very effectively damped. As a consequence interference excited stress rarely reaches unsafe amplitudes during flight although high amplitudes are not uncommon in test-stand operation where the airflow is quite turbulent. See "reverse"-whirl illustration, Fig. 7.

Group 3 vibration, or that of 1st-order propeller caused by aerodynamic interference, is usually present to a limited extent but never of sufficient amplitude to give rise to unsatisfactory conditions. However, when caused by angu-



■ Fig. 6 – Illustration of Group 2, 3rd-order propeller vibration strain on four-blade propeller



■ Fig. 7 – Illustration of Group 2, 4th-order propeller vibration strain on three-blade propeller

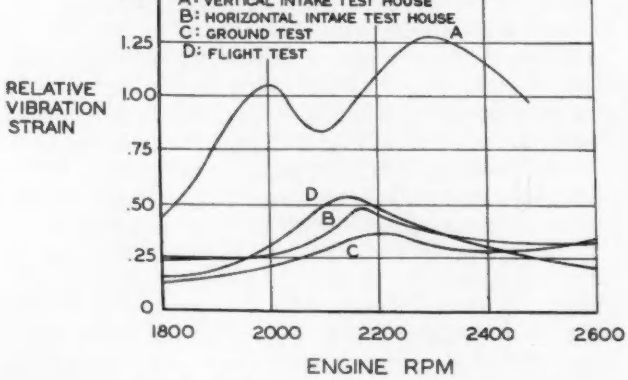
lar airflow through the propeller due to installation conditions or airplane maneuvers, such as yaw or side-slip, the resulting vibration stress amplitudes may reach undesirably high values.

■ Four-Blade Propeller Vibration

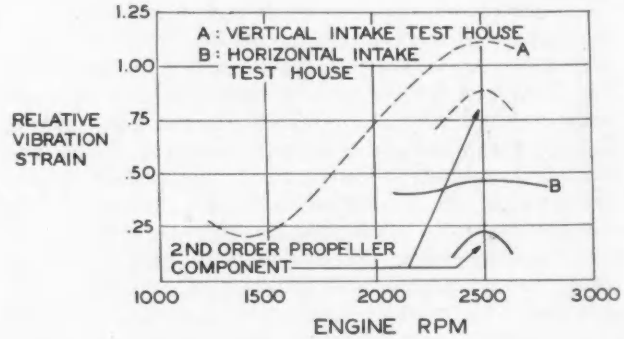
Aerodynamically excited vibrations in the four-blade propeller are quite similar in general characteristic to those excited in the three-blade propeller with the major exception that vibrations falling under the Group 4 classification can be excited. In addition, a minor difference exists in that Group 1 and 2 vibrations have relatively lower amplitudes than in the three-blade propeller primarily because the frequencies are higher and more effectively damped.

As previously expressed, the fourth group of vibrations can neither influence the engine nor be influenced by it. With this statement and the conclusion that the frequencies involved are 2, 6, 10 . . . $(2 + 4N)$ order propeller, all interest in the problem would cease but for the experimental fact that the lack of damping that the engine contributes in Group 1 and 2 vibration is absent in Group 4 in the low-frequency region where aerodynamic damping is also relatively low in amplitude. Thus, we find that 2nd-order propeller vibration is very strong because of inadequate damping while the 6th and higher frequencies are again sufficiently damped and satisfactorily low in amplitude.

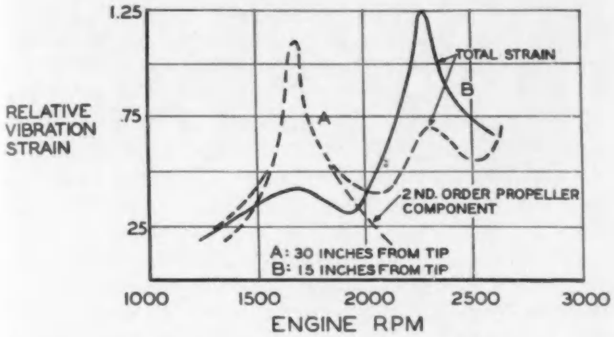
Since the first experience with 2nd-order propeller vibration, the phenomenon has been studied on a number of four-blade propellers under stand, airplane ground, and full-flight test conditions. Comparative propeller vibration strains under these various conditions for a steel propeller 13½ ft in diameter are given in Fig. 8, for an 11 ft 2 in. steel propeller in Fig. 9, and for a 13-ft aluminum-alloy propeller in Fig. 10. It will be noted that, under the worst interference conditions encountered in a test house with vertical airflow intake, the 2nd-order propeller vibration extended over a wide region. This effect was attributed to an erratic variation in amplitude and position of the aerodynamic excitation on the propeller disc. This phenomenon was mainly due to a marked dependency of airflow turbulence in the test house on the wind direction and velocity in the region of the air intake stack under conditions of high engine power and speed. Despite the fact that the propeller diameter was but slightly larger than half the orifice diameter of the test house, a considerable turbulence was set up in the air in passing from the vertical to the horizontal sections of the house. The most affected region was located at the top of the orifice and vibration records showed that the air load on the blade tended to be increased as it passed into this region. It will



■ Fig. 8 – Vibration strain in four-blade hollow-steel propeller, 13½ ft in diameter, under various operating conditions



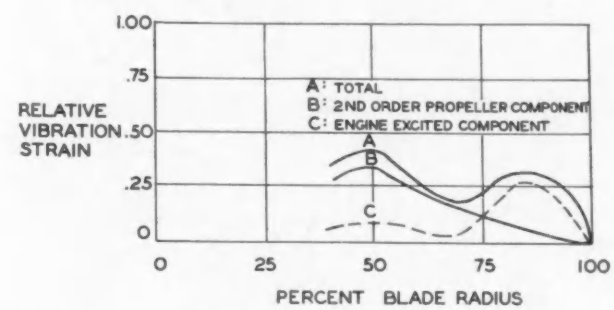
■ Fig. 9 – Vibration strain in four-blade hollow-steel propeller 11 ft 2 in. in diameter under various operating conditions



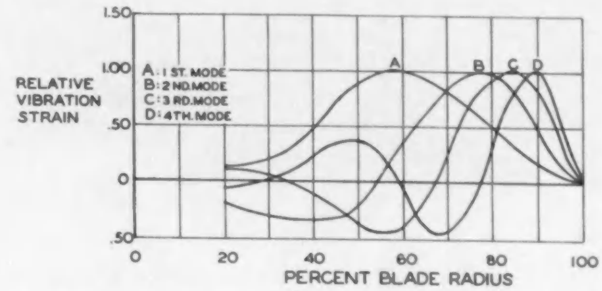
■ Fig. 10 – Vibration strain at two blade locations in four-blade aluminum-alloy propeller, 13 ft in diameter, operating in vertical airflow intake test house

be noted that a marked improvement in the propeller strain condition coincided with a change in test house to the horizontal air-inflow type and this improvement was maintained during subsequent flight tests of this propeller on a twin-engine airplane of the bombardment type with a propeller-tip fuselage clearance of approximately 10 in. During these flight tests, the distribution of vibration strain at the resonance engine speed of 2200 rpm is shown in Fig. 11. It will be noted that two regimes of blade stress exist and are caused separately by aerodynamic interference and by engine excited whirl or gas torque forcing functions which, in overlapping, tend to maintain blade strain constant over a large portion of the blade. It also will be noted that, because of this distribution, complete cancellation of 2nd-order propeller stress would only effect a 30% reduction in the maximum blade vibration stress even though the stress amplitude at the half-radius blade location would be reduced to 20% of its maximum value. The ratio of the frequencies of these two vibrations was about four to one, and modes excited in the propeller were fundamental and fourth.

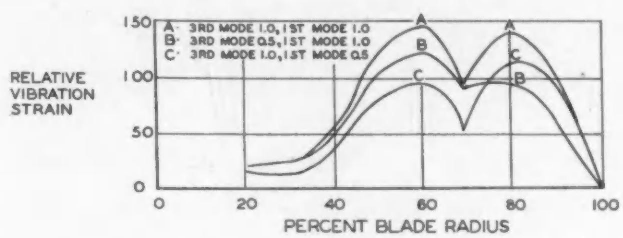
Representative vibration strain distributions for four of the lowest modes of propeller vibration are shown in Fig. 12. If, under certain excitation conditions, the fundamental and fourth modes are combined in various proportions, or the second and fourth, first and third the total stress (see Figs. 13-15) in the "overlap" region might exceed reasonable values of vibration strain. Such condi-



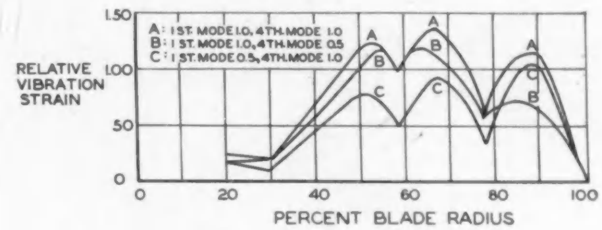
■ Fig. 11 – Vibration strain distribution during flight on one blade of four-blade propeller, 13½ ft in diameter, at 2200 rpm



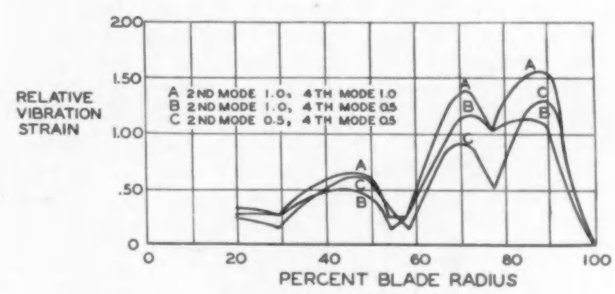
■ Fig. 12 – Representative vibration strain distribution in several symmetrical modes



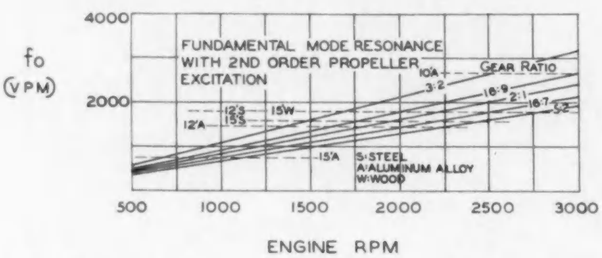
■ Fig. 13 – Strain distributions resulting from combining representative first and third symmetrical mode vibration strains in various proportions



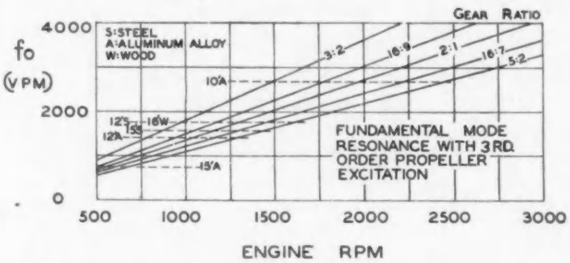
■ Fig. 14 – Strain distributions resulting from combining representative first and fourth symmetrical mode vibration strains in various proportions



■ Fig. 15 – Strain distributions resulting from combining representative second and fourth symmetrical mode vibration strains in various proportions



■ Fig. 16 – Fundamental mode resonance with 2nd-order propeller excitation



■ Fig. 17 – Fundamental mode resonance with 3rd-order propeller excitation

tions of multiple resonance have occurred in the past in three-blade propeller experience and occasionally have resulted in unsatisfactory propeller operating conditions, especially in those cases where two modes are involved in which the strain-distribution patterns with blade length are reasonably identical. It will be noted, however, that the strain distribution in the fundamental mode is sufficiently unlike distributions in the higher modes that the total stress under "overlap" conditions will only be moderately increased over the amplitude of either resonance. Therefore, although second-order propeller vibration stress probably will be encountered in a large number of four-blade propeller installations, the propeller airworthiness will not be affected materially even if the resonance so excited coincides with engine-excited resonance provided that:

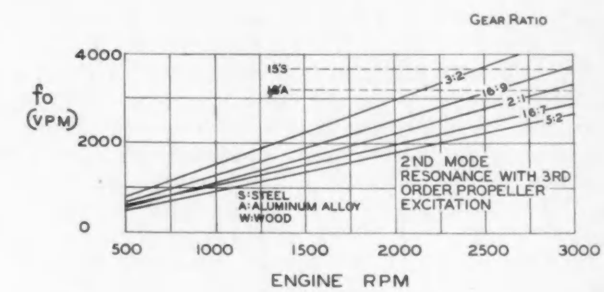
1. The strain condition in the blade without aerodynamic interference is not "borderline,"
2. The strain condition in the blade due to 2nd-order propeller excitation is not "borderline,"
3. The engine excited or higher order interference excited resonance is of fourth mode or higher.

It is also very apparent that exact coincidences cannot be tolerated if strong second or third modes are excited. Since 2nd-order propeller excitation cannot excite a higher mode than the fundamental, it would thus be necessary to establish an engine-excited second- or third-mode vibration in the propeller of fairly high level at an engine speed near the resonance caused by 2nd-order propeller excitation. While this last phenomenon is by no means rare, previous experience would indicate that lower amplitudes and therefore less trouble would be encountered with resonances in these modes than with those excited directly or indirectly in higher modes by the higher harmonics of gas torque.

■ Reducing Aerodynamically Excited Stress

Several methods of reducing vibratory stress due to aerodynamic interferences are obviously possible. In the first and oldest method, the propeller diameter may be changed. This would effect minor changes in the lowest natural frequencies of the propeller but would be more apt to alter completely the amplitude of the interference excitation on the blades. This plan would be most efficacious if the accelerated airflow region were to be relatively thin as occurs near the airplane fuselage since a reduction in propeller diameter would remove effectively the blade tip from the excitation region. However, in "pusher" propeller installations, a reasonable reduction in diameter might have little effect on propeller stress inasmuch as the interference between the propeller and the "down-wash" air from the wing is much more general over the propeller disc. In this case, an increase in diameter of the propeller or a blast of air directed into the propeller disc at another carefully selected point might effect a second interference which would balance the strain caused by the first. However, such a correction probably would effect cancellation at only a few airspeeds and under certain conditions of flight.

Change of both diameter and engine gear ratio would constitute a second method and also a more appropriate solution to the "pusher" problem since the position of the resonance in the engine speed range would be changed. See Figs. 16 to 18. It will also be noted that a change in propeller design and blade material without, however, changing diameter or engine gear ratio, might change the natural frequencies of the propeller sufficiently to correct



■ Fig. 18 – Second mode resonance with 3rd-order propeller excitation

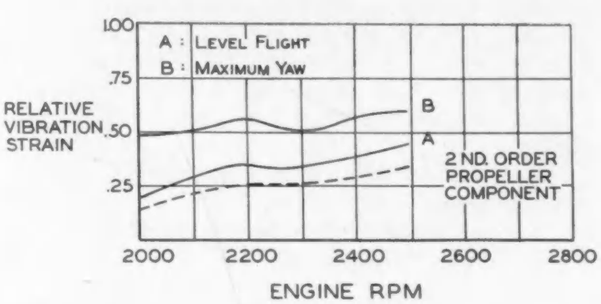
an unfavorable second-order propeller resonance location. See Fig. 16.

In order better to illustrate the effects of diameter, material, or gear ratio on the location of the interference-excited resonance in the engine speed range, several simple predictions were made. With a 15-ft diameter aluminum-alloy propeller, a 15-ft diameter hollow-steel and a 16-ft diameter composition-wood, the engine speeds for resonance with 2nd-order propeller excitation would be approximately 1150 rpm, 2400 rpm, 2750 rpm respectively, for a gear ratio of 16:7 and 1250 rpm, 2500 rpm, and 3000 rpm, respectively, for a gear ratio of 5:2. Resonance with 12-ft diameter aluminum-alloy and hollow-steel propellers would occur at engine speeds of 1750 rpm, and 2150 rpm, respectively, for a gear ratio of 16:9; 1950 rpm and 2400 rpm, respectively, for a gear ratio of 2:1.

A third method of reducing or avoiding aerodynamically excited stress would call for radical redesign of the propeller in order to change the damping characteristics of the system. In a four-blade propeller the normal 90-deg blade

spacing might be changed to 60 deg-120 deg, or to an infinite number of other spacings such as 72 deg-108 deg, and so on. All such spacings have as a primary purpose the breaking up of the 2nd-order propeller excitation pattern or the introduction of engine damping into the vibration. With the 60 deg-120 deg spacing, the engine damping introduced into the 2nd-order, propeller vibration reaches a maximum of any blade spacing and is about equivalent to the engine damping occurring with a two-blade propeller with the same blade design, vibration frequency, and amplitude. The major penalty for introduction of this damping into 2nd-order propeller vibration is the introduction of 3rd and 4th-order propeller vibration into the propeller. These last vibrations are engine-damped to the same extent as that excited by 2nd-order propeller excitation but, due to their higher frequencies, they may reasonably be expected to be controlled effectively by aerodynamic damping.

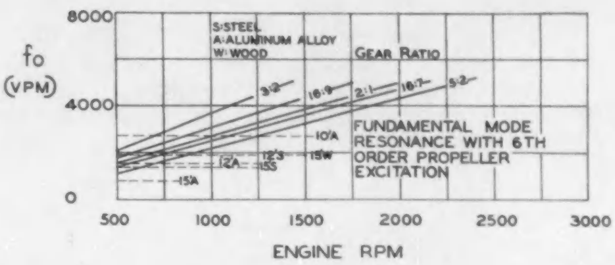
Figs. 16 to 18 summarize the situation with regard to the four-blade propeller. It will be noted that the natural frequencies of the propellers are so placed as to make it difficult to escape aerodynamically excited resonance. However, the condition of resonance need not be a severe one if due attention is paid to the location of the propeller. Tractor installations require a fuselage clearance which is dependent largely on the individual installation and, while experimentation to date indicates that a 10-in. clearance is adequate, further work must be accomplished before a definite minimum can be established. A single-engine installation will, by reason of small propeller diameter, have natural frequencies placed above the operating range of the engine-propeller combination but, since the interference excitation in these installations is quite small, blade vibration stresses caused by the interference condition will be small as a consequence. Curve *A* in Fig. 19 shows the maximum vibratory stress in a four-blade propeller of fundamental natural frequency of about 3000 cpm on an airplane of the fighter type with a 3:2 geared engine. Curve *B* shows the additional 1st-order propeller vibratory stress introduced under conditions of maximum yaw obtainable by the pilot and, at the higher engine speeds, at velocities in excess of 300 mph.



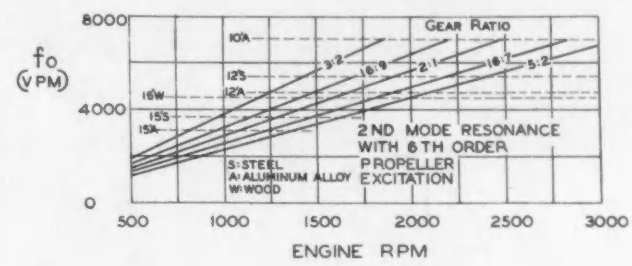
■ Fig. 19 – Vibration strain in four-blade aluminum-alloy propeller, 10 ft 3 in. in diameter, under level flight conditions (A) and maximum yaw conditions obtainable (B) in single-engine airplane of "fighter" type

■ Dual Rotation

In general, the conclusions reached in discussion of single three- and four-blade propellers apply equally well to the individual propeller in the dual-rotation unit. The excitation arising from the passage of a blade of one propeller past a blade of the other would tend to excite frequencies of vibration 6, 12, 18 . . . order propeller in the three-blade dual-rotation propeller, 8, 16, 24 . . . order propeller, in the four-blade dual-rotation propeller. The excitation would also load the blades much more uniformly over their length than an external interference. Reference again to Fig. 3 will indicate why such blade interference would be more likely to excite a fundamental mode of vibration than a 2nd or even higher mode. However, since excitation occurs at every blade passage, a smaller harmonic excitation content will be much more effective in maintaining the vibration than when the interference occurs but once per propeller revolution. Aerodynamic damping could only take place between such interferences and would be relatively ineffective. Resonance, as a consequence, will be very strong since the dual-rotation unit acts as a balanced system of forces and moments and engine damping will be negligible. Under uniform blade spacing conditions, all resonances with blade fundamental frequencies will be high and resonances with higher vibration modes very probably so. Figs. 20 and 21 graphically indicate the expected positions of these resonances in the engine-propeller operating range in various combinations of propeller natural frequency and engine gear ratio.



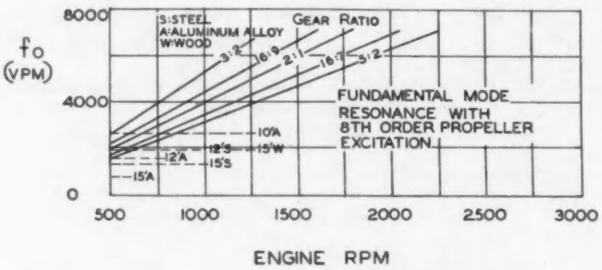
■ Fig. 20 – Fundamental mode resonance with 6th-order propeller excitation



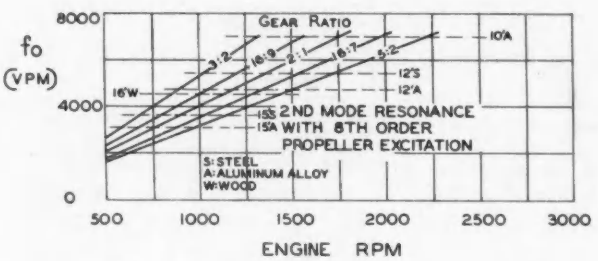
■ Fig. 21 – Second mode resonance with 6th-order propeller excitation

The four-blade propeller in the dual-rotation unit can again be designed with reference to its blade spacing to effect at least a partial damping of the interference excited vibration. If a 60 deg-120 deg blade spacing were again used in place of the normal 90-deg interval, the 8th-order propeller excitation frequency would be completely cancelled to be replaced by 4th- and 12th-order propeller frequencies. Partial aerodynamic damping in the dual-rotation unit would be present at both these frequencies which in effectiveness would be about equal to that of two two-blade propellers of the same design vibrating with the

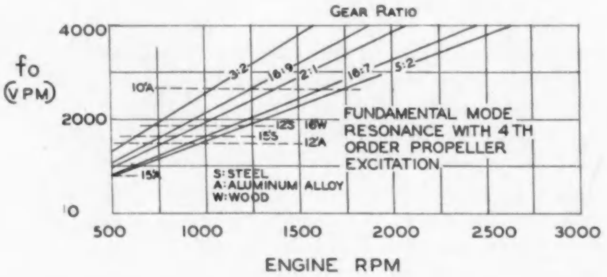
same amplitude and at the same frequencies. On the individual blade, this aerodynamic damping would amount to about 50% of the normal value on the blade while engine damping would be negligible. Since the change of excitation frequency would be very appreciable in such an arrangement and the damping very helpful in controlling vibration amplitude, this method may offer a valuable means toward making a satisfactory installation when severe resonant conditions are anticipated in the operating region with dual-rotation propellers with four equispaced blades. See Figs. 22-25. It will be noted that, in the four-blade propeller with normal 90-deg blade intervals, resonances on a 2:1 geared engine with 8th-order propeller excitation occur at 1180 rpm for the 12-ft diameter aluminum alloy propeller and 1410 rpm for the same diameter steel propeller. In the propeller with the 60 deg-120 deg blade spacing, resonances on the same engine with 4th-order propeller excitation occur at 2680 rpm for the 12-ft diameter aluminum-alloy propeller and above 3000 rpm for the 12-ft diameter steel propeller. All of these resonances involved the 2nd mode of propeller vibration.



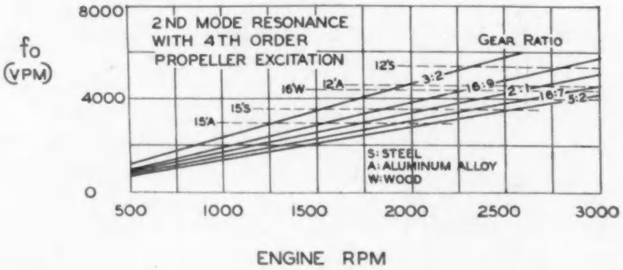
■ Fig. 22 – Fundamental mode resonance with 8th-order propeller excitation



■ Fig. 23 – Second mode resonance with 8th-order propeller excitation



■ Fig. 24 – Fundamental mode resonance with 4th-order propeller excitation



■ Fig. 25 – Second mode resonance with 4th-order propeller excitation

■ Effects of Propeller Isolation

For some time, it has been believed that complete isolation of the propeller from torsional and whirl vibration of the engine would constitute the only ideal solution to the propeller vibration problem. Aside from the design difficulties and disadvantages of a considerable increase in weight, there now seems to be some doubt as to the general desirability of that isolation. It is undeniable that engine excitation of propeller vibration would be completely absent, but it is not generally realized the extent to which the absence of engine damping would affect propeller vibration stresses caused by aerodynamic interferences. As a consequence, 2nd, 3rd, and 4th-order propeller vibration would be much more severe on the three-blade propeller and third- and fourth-order propeller vibration on the four-blade propeller with any blade spacing.

The insertion of flexibility into the propeller shaft as a first step toward complete propeller isolation also might lead to unsatisfactory vibration conditions if, as a consequence, the lower modes of propeller vibration are excited which are near or identical to those modes excited by aerodynamic interferences. A satisfactory propeller-shaft design would prevent the coincidence of these engine and aerodynamically excited resonances by isolating from the propeller, all engine alternating torques above ½-order engine in frequency.

■ Conclusion

The problems involved in developing a satisfactory engine-propeller airplane installation using propellers of four or more blades are admittedly more difficult than those encountered with a three-blade propeller. However, experiences gained so far with these propellers seem to indicate that a satisfactory installation can be made if the airplane designer will assume the responsibility of reducing propeller interference effects. This, of course, may be very difficult if not impossible to accomplish in "pusher" installations. However, if it can be accomplished in a normal tractor four-blade propeller installation without recourse to odd blade spacing, and two such installations have been made, it also can be accomplished in five or more blade propellers under similar installation conditions. Dual-rotation propellers will introduce additional propeller design problems of a vibration nature which may require certain compromises in diameter, blade construction, and engine gear ratios before satisfactory installations can be made. However, it is believed that these can be made without sacrificing airplane performance.

S·A·E Journal

DECEMBER 1941

SOCIETY OF AUTOMOTIVE ENGINEERS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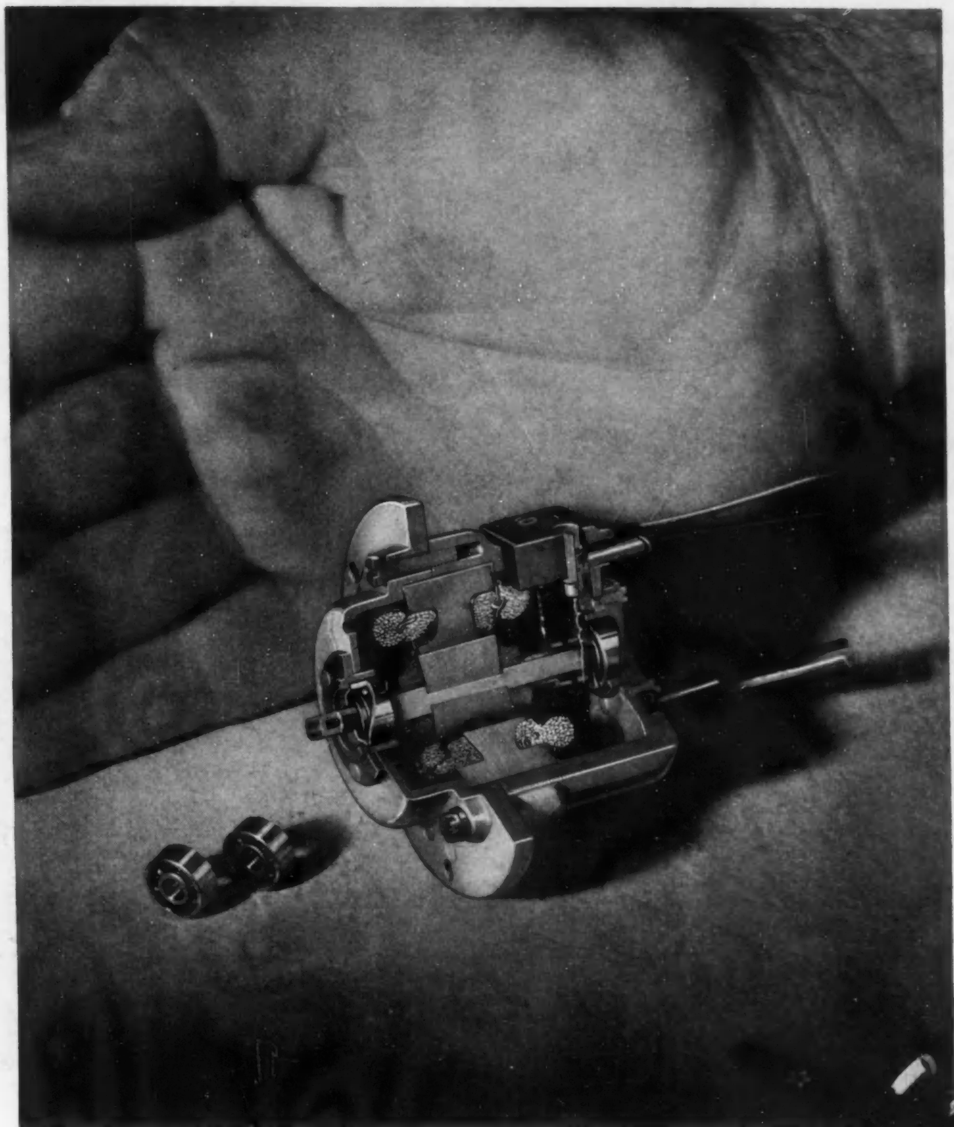
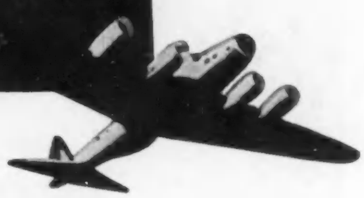
PATENTED
In every assembly location where SPEED NUTS or SPEED CLIPS have replaced conventional fastenings, immense weight reduction has been made. In many cases they actually saved 70% to 80% of the weight. That is why we say BUILD LIGHTER with SPEED NUTS.
SPEED NUTS are manufactured from high carbon spring steel and accurately heat treated. They are the only one piece fastening devices with a double spring tension lock. Severe vibration tests proved that SPEED NUTS are four times tougher in resistance to vibration loosening. And SPEED NUTS are applied so fast that time saved frequently exceeds 50%. This is why we say ASSEMBLE FASTER with the SPEED NUT SYSTEM.
Defense production lines move faster at lower cost with the SPEED NUT SYSTEM and deliver a completed product that stands the punishment of extreme vibration. Send us your assembly details now and we will mail samples and engineering data promptly.
TINNERMAN PRODUCTS, INC.
2063 Fulton Road
Cleveland, Ohio
IN CANADA:
Wallace Barnes Co., Ltd., Hamilton, Ontario
IN ENGLAND:
Simmonds Aerocessories, Ltd., London
OVER A BILLION IN USE—OVER 800 SHAPES AND SIZES





Nerves for BOMBERS

● Working in pairs, as transmitter and receiver, literally hundreds of these tiny Pioneer* Autosyn Motors in a large bomber tell the pilot the exact functioning of remote, vital parts of the ship.

Like the human nervous system, these motors must be quick-acting, sensitive. Hence, the use of two precise, ultra-sensitive New Departure ball bearings on each rotor shaft. These bearings are delivered to assemblers in oil-filled glass vials, untouched and uncontaminated by human hands.

This is only one phase of the defense effort for which New Departure is working night and day.

New Departure, Division of General Motors, Bristol, Connecticut.

** Pioneer Instrument, Division Bendix Aviation Corporation.*

New Departure

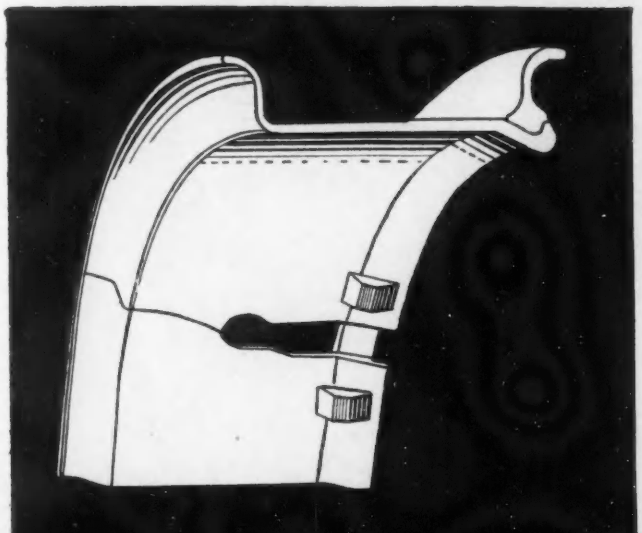
BALL BEARINGS FOR DEFENSE

3051

SAE JOURNAL, December, 1941, Vol. 49, No. 6. Published monthly by the Society of A tomotive Engineers, Inc. Publication office at 56th and Chestnut Streets, Philadelphia, Pa. Editorial and advertising departments at the headquarters of the Society, 29 West 39th Street, New York, N. Y. $1 per number, $10 per year; foreign $12 per year; to members 50 cents per number, $5 per year. Entered as second-class matter, Feb. 16, 1933, at the Post Office at Philadelphia, Pa., under the Act of Aug. 12, 1912. Acceptance for mailing at special rate of postage provided for in Section 1103 Act of Oct. 3, 1917, authorized on Jan. 14, 1928.

WORLD'S MOST POPULAR RIMS

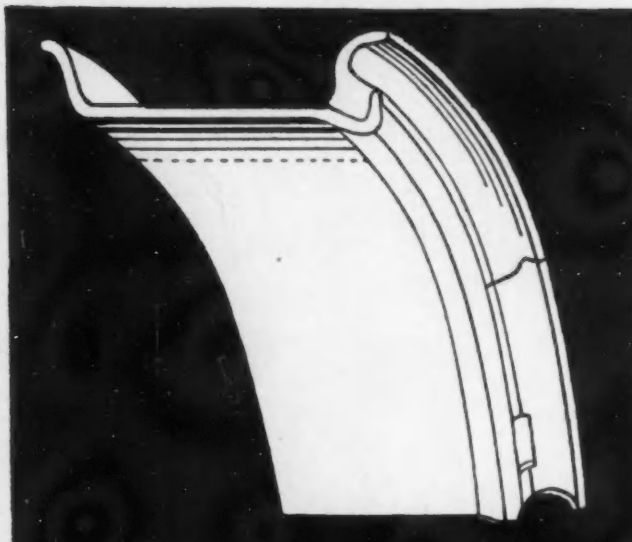
for simplicity • ease of operation • strength • safety



Goodyear TYPE "K-28" RIM

For use as demountable rim only, on trucks, trailers, buses

Safest demountable rim made. Split base — mechanically locked between wheel and continuous side ring.



Goodyear TYPE "L" RIM

made for use with disc type wheels

A solid-base rim with split interlocked side ring. Also built with drivers for use as a demountable rim.

THE unchallenged popularity of Goodyear Type "K-28" and Type "L" rims is due to these facts:

They are the only two types of rims that are commercially and mechanically practical in ALL standard sizes for use on trucks, trailers and buses.

Both types give a full circumferential seat for tire beads – rings cannot float and injure tire.

They require no special tools to remove or apply – just an ordinary tire iron.

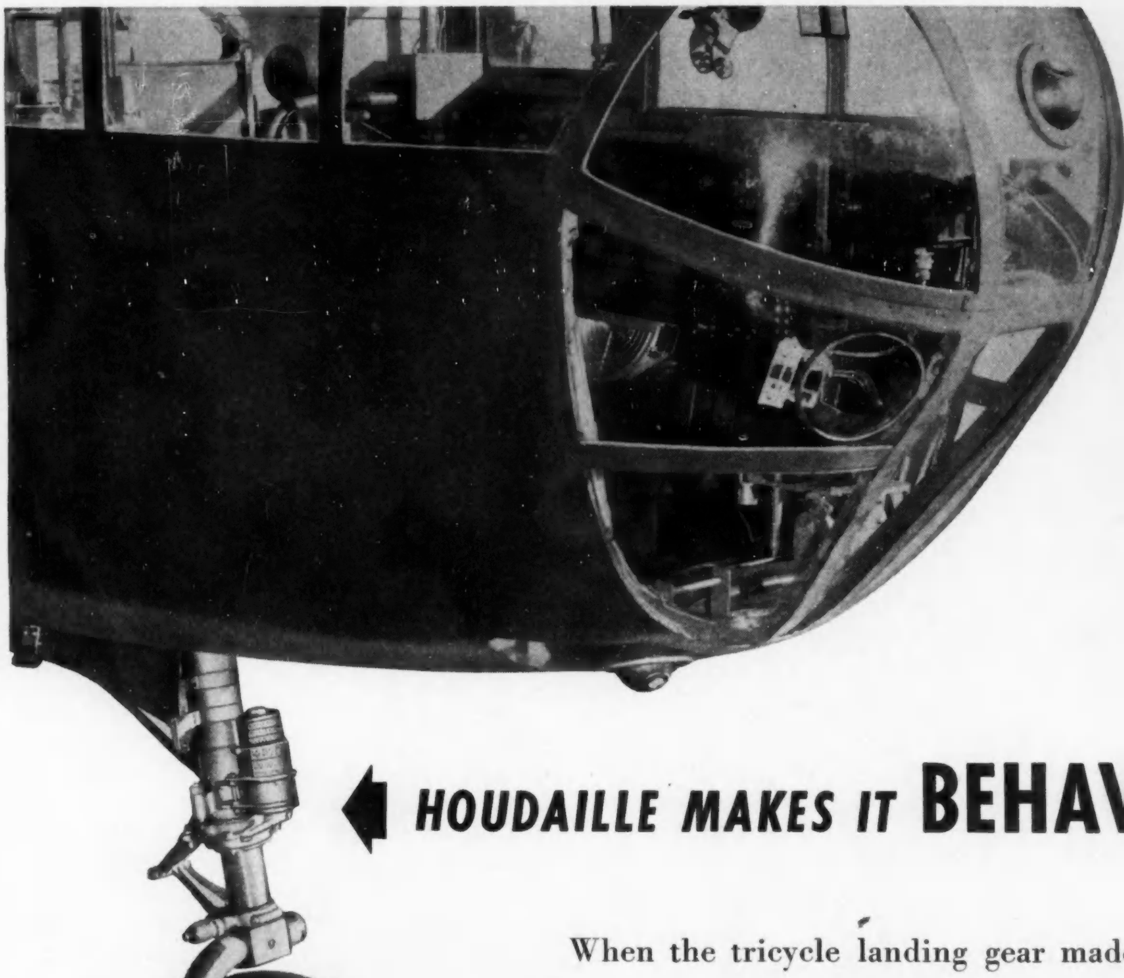
They are the quickest, simplest rims to operate – prevent tire abuse from pounding, save labor and time.

Both types are made from hot-rolled special steel for extra strength and long life. They are produced in Goodyear's own steel-fabricating plant and embody our years of experience as the world's largest tire builder.

Goodyear rims are standard equipment on many types of motor vehicles and are approved for use on motor transport and ordnance equipment of the United States Army. Specify them on your next order.

THE GREATEST NAME IN RUBBER

GOODYEAR



HOUDAILLE MAKES IT BEHAVE

When the tricycle landing gear made its re-appearance on modern high-speed airplanes, designers knew that uncontrolled pivoting of the castering wheel might easily cause serious difficulty.

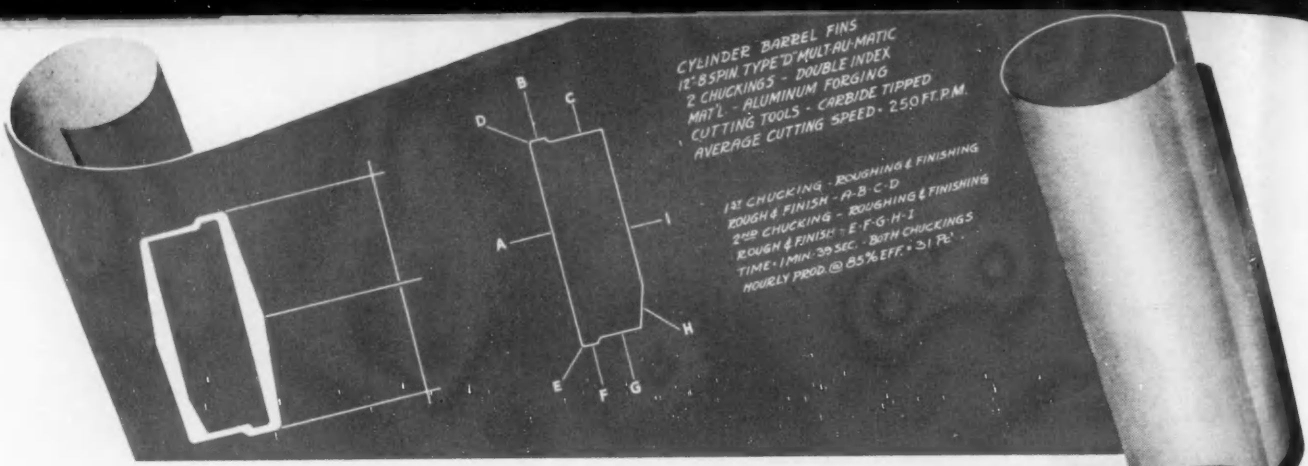
Accordingly, Houdaille engineers who have specialized for a quarter century in precision hydraulic equipment developed the Houdaille Shimmy Dampener as an integral part of the nose wheel strut.

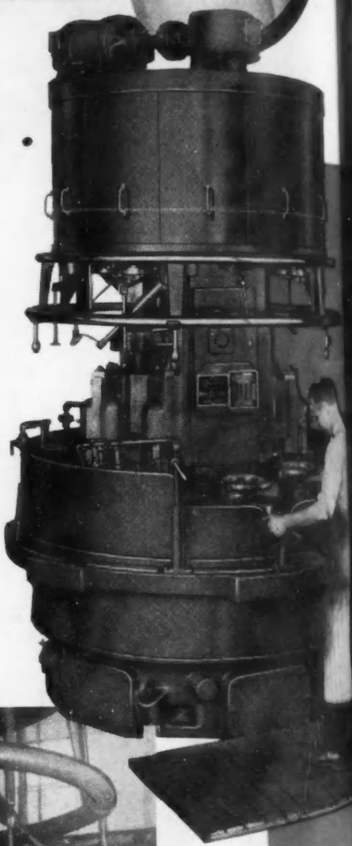
The extensive facilities of the Houde Engineering Corporation are engaged in manufacturing these and other precision parts in impressive quantities for the aviation industry.

HOUDE ENGINEERING CORPORATION

BUFFALO, N. Y.

A DIVISION OF HOUDAILLE-HERSHEY CORPORATION





Finish Machined . . .

32 PER HOUR

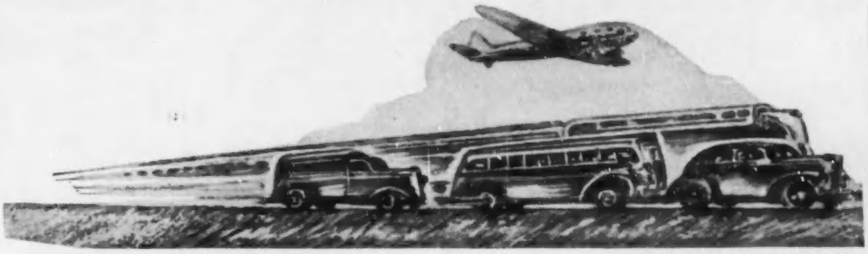
These forged aluminum airplane engine cylinder barrel fins are machined all over by a Type "D" Bullard Mult-Au-Matic, employing the double indexing feature, which rough and finish machines the 9 surfaces at the rate of 32 parts per hour.

This is a typical Bullard contribution to National Defense work. Any manufacturer with parts of this general character to produce, can use the Mult-Au-Matic method to increase his production. Time studies will be made on request.

THE BULLARD COMPANY BRIDGEPORT, CONNECTICUT

SAE JOURNAL *Pre-Prints*

THE SOCIETY
OF
AUTOMOTIVE
ENGINEERS,
INC.
29 W. 39TH ST.,
NEW YORK



News of the
JANUARY
Issue

By Norman G. Shidle

Washingtonia

COPPER is still tops in tightness – with threats to prohibit its use in automobile radiators growing. If a prohibition does come "suddenly" six or eight months hence, OPM officials will consider that industry has had plenty of warning – as *they* believe it did on bright work elimination.

* * *

Right or wrong, nobody can convince OPMers that industry is justified in hearing only official orders. They keep on expecting industry to tune in actively on "suggestions" as well . . . And some automotive executives and engineers now are saying that only by working actively on "suggestions" is there any chance left to have some say about how your own business is going to be run . . . a sad thought . . . but . . . could be.

* * *

Heard around the New Social Security Building recently:

OPM stands for . . . "O Promise Me"
OPM stands for . . . "Other Peoples' Money"
OPM – "A madhouse . . . which is run by the inmates."

Which only goes to show that the same kind of people work there as go there. Cooperation will always be possible between groups with a variety of interests as long as everybody retains some sense of humor.

* * *

An order is now being written at OPM to prohibit (except by special permission) use of scrap iron or steel out of which machine tools may be made.

This is indicative of (a) an acute shortage of steel scrap which may result in *less* steel output in 1942 than in 1941, and (b) emphasis soon to be placed on collection *and segregation* of all metal scrap as already is common with aluminum scrap.

* * *

OPM is conferring with the U. S. Mint to see about the chances of taking all the nickel out of nickels – as a nickel conservation measure. An act of Congress would be required.

* * *

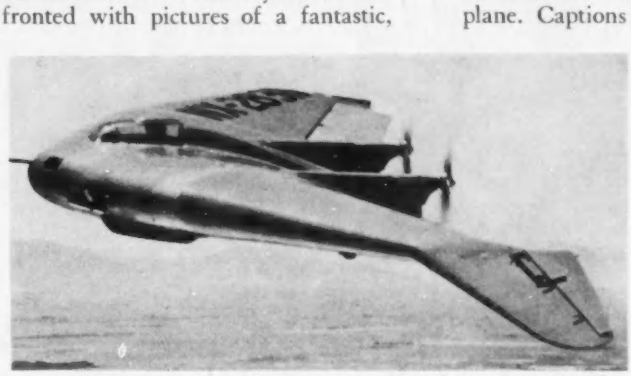
From now on, priorities for materials will not be granted for any project – defense or civilian – until it has been given a "strip-tease" for critical materials elimination by OPM Conservation Division – Julius Rosenwald, chief.

* * *

Our none-too-large tungsten supply may be importantly augmented if President Roosevelt signs the revised Federal Aid highway bill now coming through Congress – or if he had signed the earlier bill, for that matter. This bill, if passed, will permit the opening of new roads in the West necessary to tap new tungsten deposits, OPM officials say.

ENGINE-IN-WING DESIGN REDUCES PLANE DRAG 15%

NEWSPAPER readers received a distinct shock recently when confronted with pictures of a fantastic, bat-like wing that flies through the air with the ease of a bona fide airplane. Captions assured Mr. and Mrs. America that, even though tail surfaces and fuselage were missing, the strange craft was an airplane. In fact, reports said, Northrop Aircraft, Inc., was secretly developing the design as a radical new type of transport airplane.



Press Association, Inc.

The Northrop "Flying Wing," shown, is said to be designed to exceed 500 mph, climb almost straight up, and generally outfly the latest pursuit ships. Reportedly, the plane is equipped with tricycle landing gear, two three-bladed pusher propellers and two 120-hp engines capable of sending it into high altitudes

Technical details of the Northrop "Flying Wing" are still veiled by government censorship, but the story of submerging engines in airplane wings (one of the main drag-reducing features of the Northrop design) can be told, and will be . . . in the January SAE Journal. H. H. Ellerbrock, Jr., mechanical engineer, National Advisory Committee for Aeronautics, describes the thrilling NACA engine-in-wing tests, started back in the summer of 1940, which provided background for the radical

(Concluded on page 10)

The
FEBRUARY
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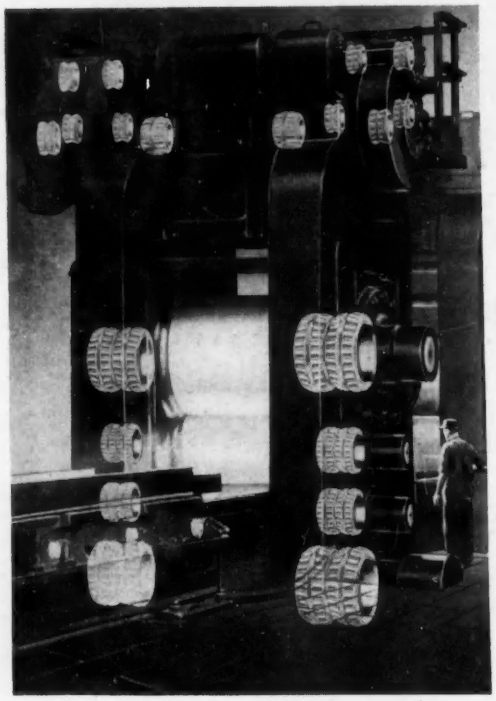
will be the

BIG

ANNUAL MEETING

NUMBER

Roller Bearings Conserve Power for Defense



Application points of Timken tapered-roller bearings in a modern 4-high steel rolling mill stand. Loads are measured in millions of pounds per bearing. A special type of lubricant had to be developed to withstand the tremendous pressures

THE recent OPM edict restricting the use of electrical power has emphasized the important fact that power is just as essential to our production of planes, ships, tanks, and guns, as is, for example, steel or copper. Hence engineers in defense plants are investigating all possible ways and means for conserving this vital commodity.

One significant means of conserving power in industry is through use of antifriction bearings, and, as defense piles additional burdens on high-speed production, the advantages of these versatile parts become of greater importance.

For example, roller-bearing applications to machine tools and steel mills have virtually revolutionized equipment in these industries . . . in some cases, doubling output as compared with the former plain bearing installations. So states Timken's vice president, T. V. Buckwalter, in a thought-provoking January SAE Journal article entitled, "Roller Bearings and National Defense."

Recounting some of the amazing uses to which roller bearings are being put in defense, author Buckwalter says: "The roller bearing has improved industrial production to an almost unbelievable extent and has provided a capacity for producing the implements of war that cannot be matched by any other nation."

Engine-In-Wing Design Reduces Plane Drag 15%

(*Concluded from page* 9)

design work being done today.

The increases in speed and payload or bomb-load, made possible by enclosing engines in the wing, have long intrigued aircraft designers; but cooling and accessibility problems have heretofore held up development. Mr. Ellerbrock shows how these problems can be licked. Here is specific technical information for aircraft engineers on engines that make possible wing designs which reduce drag 15% on small planes and 5% on large planes. . . . Watch for "Cooling Characteristics of Submerged Light Aircraft Engines."

Distributor Efficiency Cut By Fuel Differences

IF an engine manufacturer wanted to get the most power from his engine, he would take the best fuel on the market and build a distributor that would give maximum performance with that fuel. That's the ideal set-up. Such a method, however, is not feasible with the average car engine, because:

1. Most car owners use more than one type of fuel. This means distributor adjustments for each fuel for highest efficiency, because of differences in fuel characteristics;

2. Most car owners are more interested in "knockless" operation than maximum power or fuel economy.

Hence, the modern distributor is a compromise. It must take the poorest fuel without objectionable "knock," yet give good performance on the best fuel.

In the January SAE Journal, J. T. Fitzsimmons, Delco-Remy Division, General Motors Corp., explains the engineering compromises that must be made to build the best type of distributor under present conditions. He sees no increase in the accuracy of the modern distributor until fuels are more nearly standardized as to "knock."

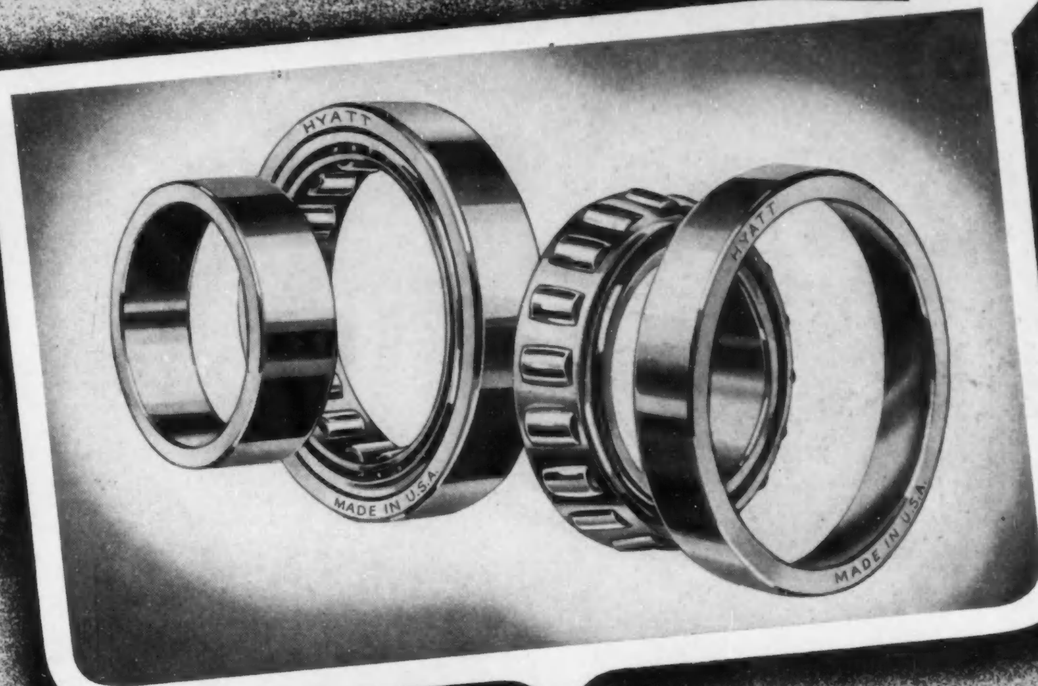
Safety Is What You Make It

A POWERFUL five-ton truck hurtles through the night, its headlights cutting a wedge in the blackness. In the dimly-lighted cab, the driver fights to keep awake. He has struggled with this terrible drowsy feeling for miles; tried to stir his lethargic brain with swigs of black coffee from a thermos bottle. But it's no use. His eyelids are like lead. Should pull off the road for a spell, but there is a schedule to meet . . . The drone of the wheels, stuffy air . . . His eyes close . . . for a second. . . .

Crash!

Who's to blame? The Driver? Perhaps. But, the mechanic at the garage could be responsible, or even the man at the drawing board . . . the man who designed the truck cab and controls. If a cab is poorly ventilated against engine fumes; if the roof is so low that the driver must hunch over; if controls are hard to operate; if visibility is bad – all these things impose a tremendous burden on a long-haul driver's efficiency and endurance.

J. Willard Lord, safety engineer, Atlantic Refining Co., has a good deal of potent comment to make on the relation of vehicle design to safety in modern truck operation . . . in the January SAE Journal. More attention must be given to safety features, Mr. Lord declares. Mirrors, brakes, seating, arrangement of controls, headlighting, bumpers, all need careful scrutiny to meet today's pressing requirements. Fleet engineers will find stimulating answers to their safety problems and vehicle designers will receive valuable tips from "Design Elements Affecting Safety."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MADE IN U.S.A.
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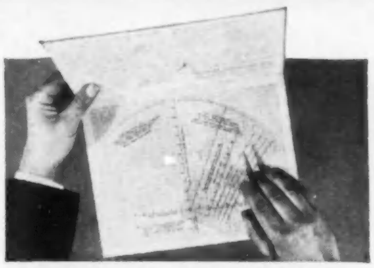


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precision seals, manufactured to rigid specifications, tested, and accepted—used on Wright Aeroplane Engines at U. S. Naval Factory, and doing their part in the country's national defense job.
Protect your equipment with Milpaco Oil Seals. Let an experienced Milpaco man tell you about the advantages of Milpaco Oil Seals.
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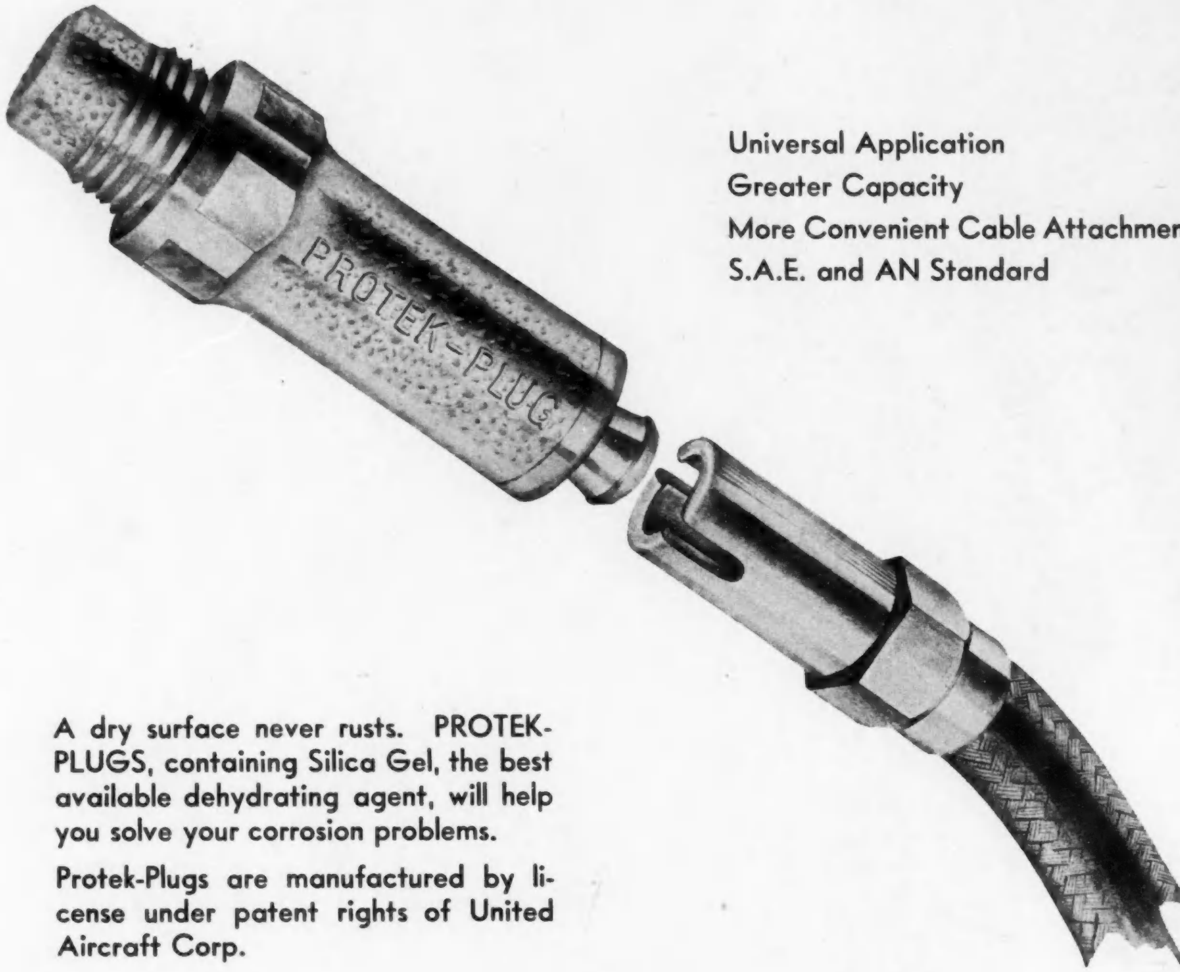
BOWER ADVANTAGES
Propeller Shaft
Ask for it...
An interesting full color, 4-page folder featuring Bower
BOWER ROLLER BEARING CO.

AN *Improved* PROTEK-PLUG*

THE ANTI-CORROSION TREATMENT FOR AIRCRAFT ENGINES AND ACCESSORIES



Universal Application
Greater Capacity
More Convenient Cable Attachment
S.A.E. and AN Standard

A dry surface never rusts. PROTEK-PLUGS, containing Silica Gel, the best available dehydrating agent, will help you solve your corrosion problems.

Protek-Plugs are manufactured by license under patent rights of United Aircraft Corp.

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PROTEK-PLUGS are a product of

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Manufacturers of Aircraft Carburetors, Fuel Pumps and Accessories.

CONSOLIDATED B-24'S FABRICATED WITH BOOTS SELF-LOCKING NUTS

80,000 POUNDS LIGHTER—THANKS TO A "MAGIC" NUT

A single fleet of planes, being built for the United States Government by Consolidated, will carry 80,000 pounds★ less dead weight, solely through the use of Boots Self-Locking Aircraft Nuts. As one result, they can lift that much more fire-power, armament or fuel.

Boots Nuts are of sheet metal construction—have all the required tensile strength of old-fashioned nuts. They "outlast the plane," and can be reused as often as required in the airplane's maintenance.

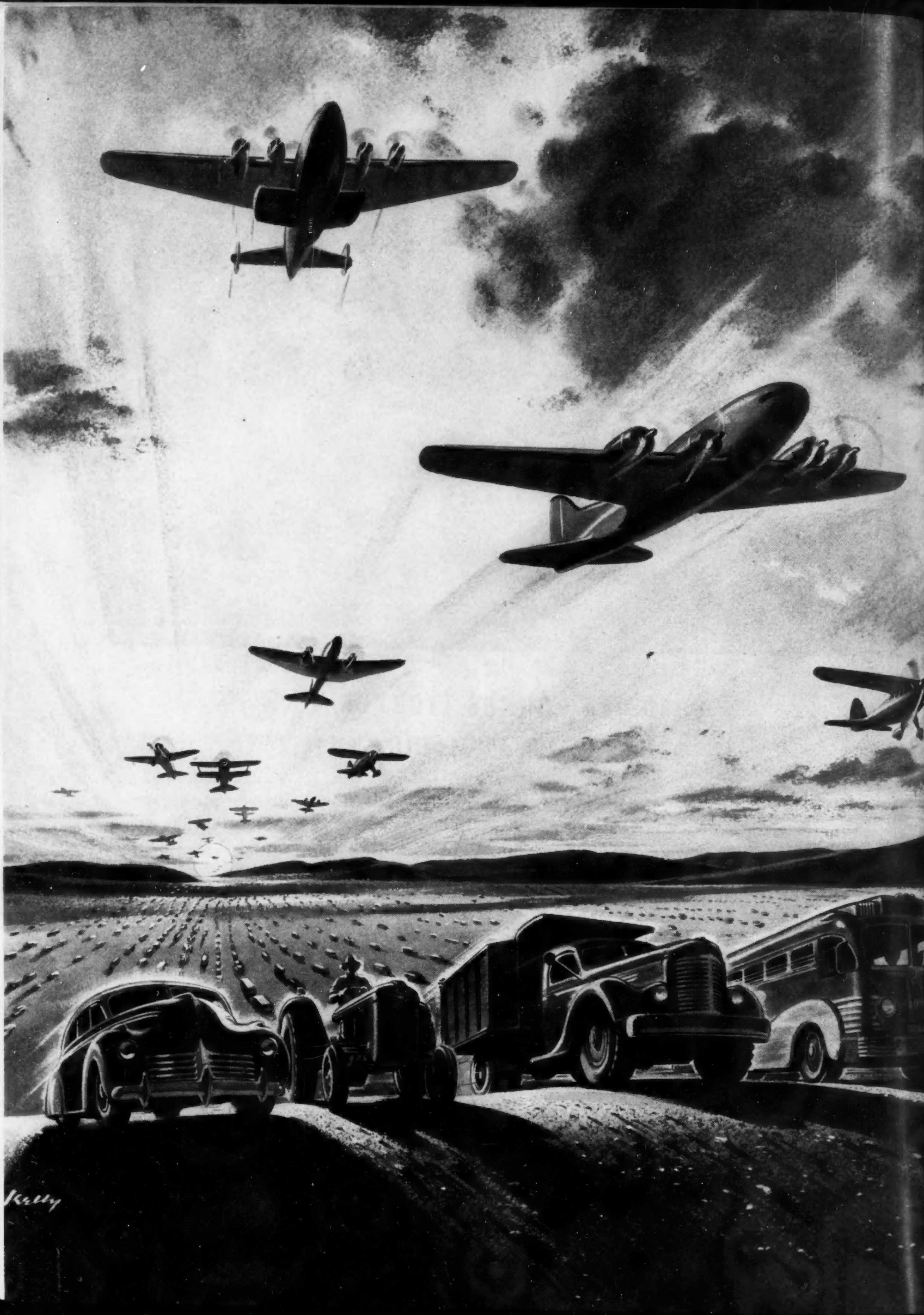
The only one-piece, all-metal, self-locking nut to pass the rigid tests of the Army, Navy and Civil Aeronautics Authority.

★Consolidated's Chief Engineer figures a weight saving of 40 to 80 lbs. per B-24 due to use of Boots vs. nuts used in the past.

BOOTS

AIRCRAFT NUT CORPORATION

NEW CANAAN, CONNECTICUT


Kelly

Airlines Save Years in Defense

HOLDING back the clock . . . turning travel hours into work time, the airlines are helping this country's defense production overtake in months a head start of years. Key men transported across the country or over oceans in a matter of hours—materials, plans, blueprints on hand when and where needed—these are the contributions of the airlines.

Organized and equipped for normal peacetime traffic, the airlines were abruptly assigned a key role in defense. They accepted the job calmly and without show—training personnel, extending routes, and increasing schedules without relaxing either vigilance or quality of service. No gauge can measure America's debt to the airlines for the thousands of productive hours that otherwise would have been lost forever.

WRIGHT AERONAUTICAL CORPORATION
A Division of Curtiss-Wright Corporation • Paterson, New Jersey

3000 Man-years Saved in 1941

For each 100 miles flown, 1 hour and 42 minutes is saved over the next fastest mode of travel. Flying over 1,500,000,000 passenger miles, the airlines have saved 3000 man-years in 1941. Designed to speed traffic is this new Airlines Terminal building in midtown Manhattan.

Born in a Forge Shop

PURSUIT plane, combat plane, bomber, or peaceful airliner, all have their auspicious beginnings in the fire and thud of the powerful drop hammer. From the massive propeller hub, crankshaft and crankcase forgings to the tiny clevis forgings, wherever utmost strength and dependability with minimum weight are vital, there you will find the drop forging . . . more than likely produced on a Chambersburg hammer . . .

CHAMBERSBURG ENGINEERING CO. • CHAMBERSBURG, PA.

THE CECOSTAMP • A NEW METHOD OF PRODUCING AIRPLANE STAMPINGS

A NEW, high-production, easily controlled, impact-type drop stamp, designed by Chambersburg engineers after a close study of aircraft manufacturing requirements. In the rapid production of drop stampings from stainless steel, high strength aluminum alloys and other metals of low ductility, the CECOSTAMP has taken its place with the newer tools and techniques made necessary by this great industry.

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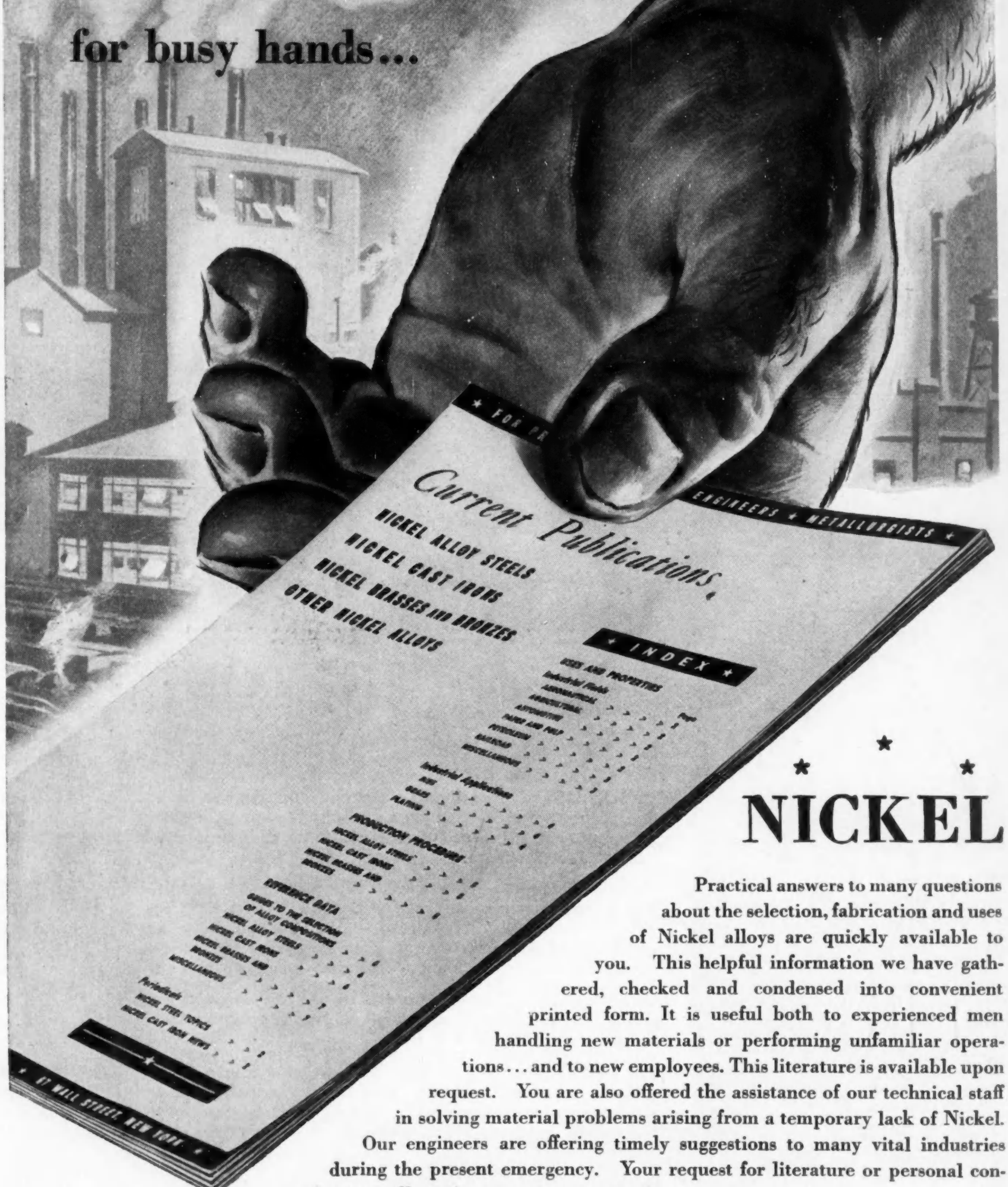
HAMMERS • CECOSTAMPS • PRESSES

PRACTICAL helps for busy hands...



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NICKEL

Practical answers to many questions about the selection, fabrication and uses of Nickel alloys are quickly available to you. This helpful information we have gathered, checked and condensed into convenient printed form. It is useful both to experienced men handling new materials or performing unfamiliar operations... and to new employees. This literature is available upon request. You are also offered the assistance of our technical staff in solving material problems arising from a temporary lack of Nickel. Our engineers are offering timely suggestions to many vital industries during the present emergency. Your request for literature or personal consultation will receive our prompt attention.

THE INTERNATIONAL NICKEL COMPANY, INC. 67 WALL STREET NEW YORK, N. Y.

THE GLASS THAT *gives you Better Vision*

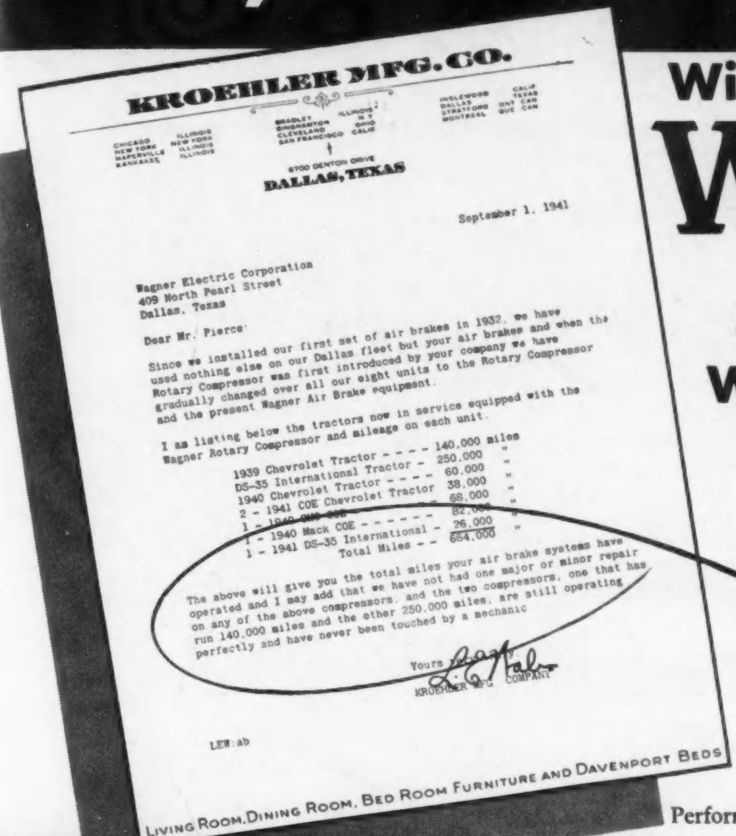
● Good plate glass permits you to see things as they are. L·O·F Hi-Test Safety PLATE is that kind of glass—with ground and highly polished surfaces—a safety sandwich, with two lights bonded together by a strong, tough, transparent plastic.

L·O·F Hi-Test Safety PLATE reduces eyestrain . . . makes riding and driving safer and more enjoyable. Motor cars equipped with L·O·F Safety Plate offer maximum protection, comfort and uninterrupted vision.

The familiar L·O·F trademark—"The Mark of Quality"—identifies this glass. Car salesmen use it as evidence of the finer materials in the car they sell.

LIBBEY · OWENS · FORD
HI-TEST *Safety Plate* GLASS

664,000 Safety Miles



KROEHLER MFG. CO.

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INGLEWOOD CALIF. / DALLAS TEXAS / STRATFORD ONT. CAN. / MONTREAL QUE. CAN.

6700 DENTON DRIVE
DALLAS, TEXAS

September 1, 1941

Wagner Electric Corporation
409 North Pearl Street
Dallas, Texas

Dear Mr. Pierce:

Since we installed our first set of air brakes in 1932, we have used nothing else on our Dallas fleet but your air brakes and when the Rotary Compressor was first introduced by your company we have gradually changed over all our eight units to the Rotary Compressor and the present Wagner Air Brake equipment.

I am listing below the tractors now in service equipped with the Wagner Rotary Compressor and mileage on each unit.

1939 Chevrolet Tractor - - - -	140,000	miles
DS-35 International Tractor -	250,000	"
1940 Chevrolet Tractor - - - -	60,000	"
2 - 1941 COE Chevrolet Tractor	38,000	"
1 - 1940 [illegible]	68,000	"
1 - 1940 Mack COE - - - - - -	82,000	"
1 - 1941 DS-35 International -	26,000	"
Total Miles - -	664,000	

The above will give you the total miles your air brake systems have operated and I may add that we have not had one major or minor repair on any of the above compressors, and the two compressors, one that has run 140,000 miles and the other 250,000 miles, are still operating perfectly and have never been touched by a mechanic

Yours very truly,
L. E. Waller
KROEHLER MFG. COMPANY

LEW:ab

LIVING ROOM, DINING ROOM, BED ROOM FURNITURE AND DAVENPORT BEDS

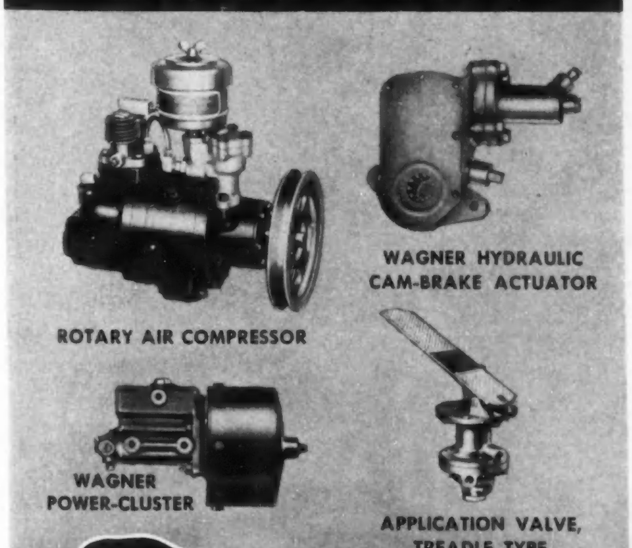
With WAGNER *Air Brakes*

Without any repairs

"We have not had one major or minor repair on any of the above compressors, and the two compressors, one that has run 140,000 miles and the other 250,000 miles, are still operating perfectly and have never been touched by a mechanic."

Simple to Install

All parts that make up the Wagner air brake systems are simple in design and easy to install.



Performance of this kind is not unusual when Wagner Air Brakes — the air brakes with the rotary compressor — are on the job.

Thousands of truck owners and fleet operators have already installed Wagner air brakes on their rolling equipment and this increasing popularity makes Wagner air brakes an extra profit item that manufacturers and dealers should not overlook.

There are three Wagner Air Brake Systems: (1) The Wagner Hydrair Brake, the system that combines air power with hydraulic actuation; for commercial vehicles equipped with cam brakes, (2) the Wagner Air-Hydraulic Brake for commercial vehicles equipped with internal hydraulic brakes, and (3) the Wagner Straight-Air Brake, for commercial vehicles equipped with straight-air-actuated cam brakes.

Write for new Air Brake Booklet KU-50. It explains and illustrates the complete line of Wagner Air Brakes.

K41-9

WAGNER ELECTRIC CORPORATION 50TH 1891 • 1941 ANNIVERSARY ST. LOUIS, MO.

Wagner Electric Corporation
6400 Plymouth Avenue, Saint Louis, Mo., U.S.A.
BRAKES MOTORS TRANSFORMERS FANS

FEDERAL
for Precision
FEDERAL FAN AND WATER-PUMP BEARING
THE FEDERAL BEARINGS CO., INC.
Makers of Fine Ball Bearings
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Detroit Office: 2640 Book Tower • Cleveland Office: 402 Swetland Building
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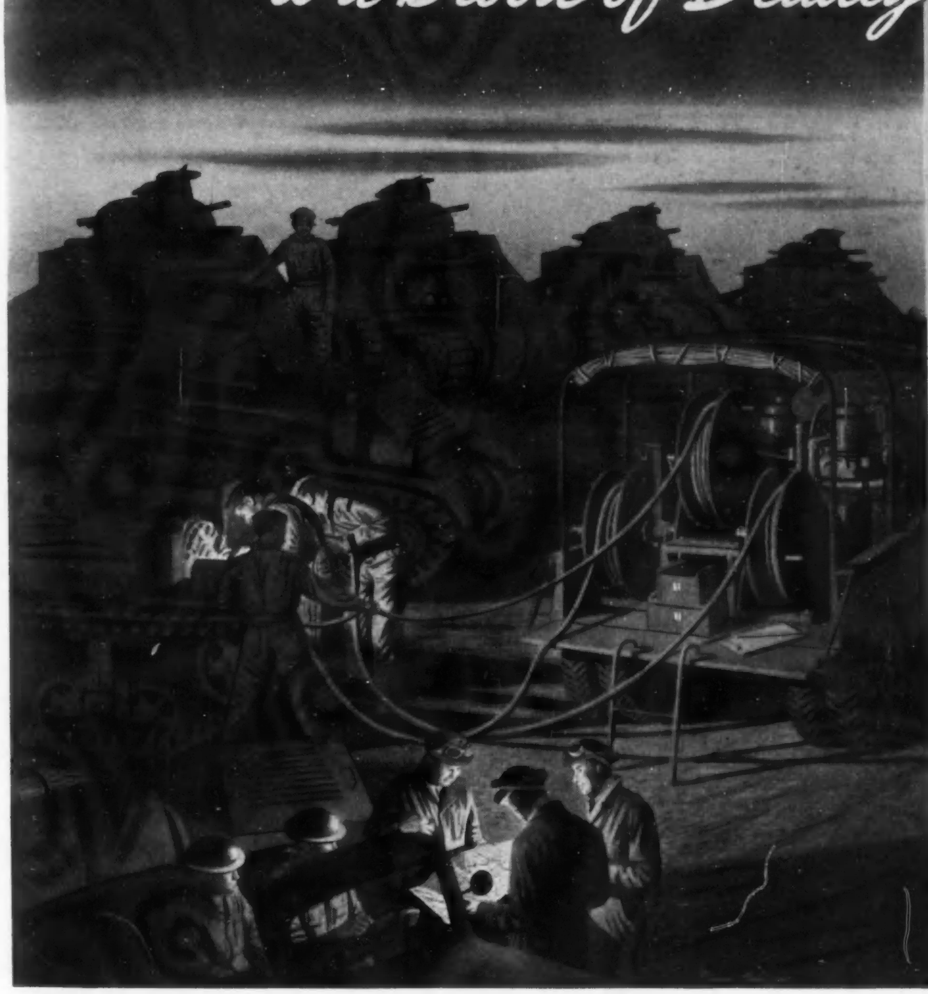
TRADE MARK

100 HORSEPOWER NURSE-MAID...

to a Brood of Deadly Monsters



SOLDIERS of Uncle Sam's Tank Corps learn about problems unknown in civilian life. For instance, a motorist takes to the highway without a worry, knowing there will be service stations at all main crossroads. But . . .

The whole countryside is "highway" for a tank brigade. Those deadly monsters work up giant appetites for grease, oil, and fuel. But there are no service stations in a battle zone.

Because contractors operate trucks and tractors and power shovels on projects remote from service stations, their needs were met by mounting Alemite Power Lubrication Equipment on trucks and providing every needed service *on the job*. So the Army found the answer, ready-made, developed by Stewart-Warner to do a similar peacetime job.

Thus another Stewart-Warner product, engineered for peace, *fits smoothly into defense needs* —another instance of the readiness with which America's automotive industry meets America's emergency requirements.

Another Product of
STEWART
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STEWART-WARNER

SPEEDOMETERS • TACHOMETERS • AMMETERS • VOLTMETERS • COMPLETE INSTRUMENT PANELS • THERMO-ELECTRIC GAUGES • MECHANICAL GAUGES • ELECTRIC FUEL PUMPS ELECTRIC WINDSHIELD WIPERS • SOUTH WIND GASOLINE CAR HEATERS • DIE CASTINGS • ALEMITE LUBRICATION EQUIPMENT • HOOD HARDWARE • INTERIOR HARDWARE

1844 DIVERSEY PARKWAY, CHICAGO • GENERAL MOTORS BUILDING, DETROIT

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HE KNOWS HOW TO USE THEM!

He knows that Form-A-Gasket Number 1 is a paste that dries fast and sets hard. That it is used principally for making permanent assemblies and for building up uneven or warped surfaces. That it withstands highest pressure and greatest strain.

He knows that Form-A-Gasket Number 2 is a paste that dries slowly, remains pliable and disassembles very easily. That it is principally used for general assembly work. That it preserves all types of solid gaskets from heaviest copper to thinnest cork.

He knows that Aviation Form-A-Gasket is a liquid that quickly changes to a paste and does not dry nor run when subjected to heat. That it is used on cylinder head gaskets to prevent water seepage, corrosion and head seizure. That it is extensively used on machined surfaces and screw thread connections.

PERMATEX COMPANY, INC., Sheepshead Bay, N. Y., U. S. A.

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Gibson Co., Wm. D., Div. of Assoc. Spring Corp.

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Suppressors, Radio, Spark Plug
Erie Resistor Corp.

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Switches, Starting
Electric Auto-Lite Co.

Tachometers
Stewart-Warner Corp.

Tappets
Wilcox-Rich Div. of Eaton Mfg. Co.

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Titeflex Metal Hose Co.

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Titeflex Metal Hose Co.

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Thompson Products, Inc.
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Felters Co.

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Wicks, Felt
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Felters Co.

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Elastic Stop Nut Corp.

The addresses of companies listed in this index can be obtained from their current advertisements indexed on page 94.

WANTED

AIRPLANE POWER PLANT ENGINEER

Familiar with mock up practice. Attractive position open to experienced applicant interested in pursuing this form of powerplant design development and control for an experimental powerplant design group. Applications not solicited from persons now employed in aircraft manufacturing industry. Apply by letter to Installation Department.

PRATT & WHITNEY AIRCRAFT
EAST HARTFORD, CONNECTICUT

WANTED

FLIGHT TEST ENGINEER

Capable of supervising the assembly of experimental aircraft power plant installations, test instruments and special test equipment, and familiar with both airplane and power plant flight test procedure. Must be able to set up complete flight programs and supervise recording of data in flight. Applications not solicited from persons now employed in aircraft manufacturing industry. Apply by letter to Installation Department.

PRATT & WHITNEY AIRCRAFT
EAST HARTFORD, CONNECTICUT

WANTED

ENGINEERING STRESS ANALYST

Capable of independent design and analysis of aircraft structures relating to power plant installations. Apply by letter to Installation Department. Applications not solicited from persons now employed in aircraft manufacturing industry.

PRATT & WHITNEY AIRCRAFT
EAST HARTFORD, CONNECTICUT

WANTED

ENGINEER

Familiar with aircraft power plant design, fabrication and testing and capable of composing, editing and preparing technical reports, bulletins and manuals relating to this art. Apply by letter to Installation Department. Applications not solicited from persons now employed in aircraft manufacturing industry.

PRATT & WHITNEY AIRCRAFT
EAST HARTFORD, CONNECTICUT

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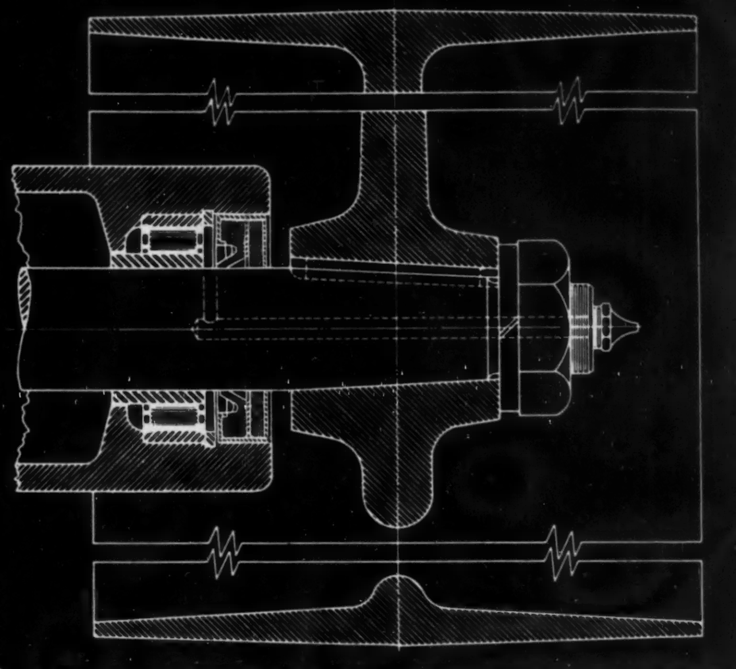

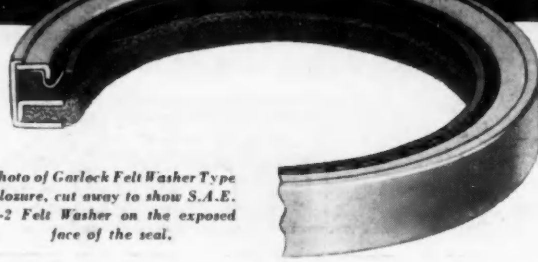

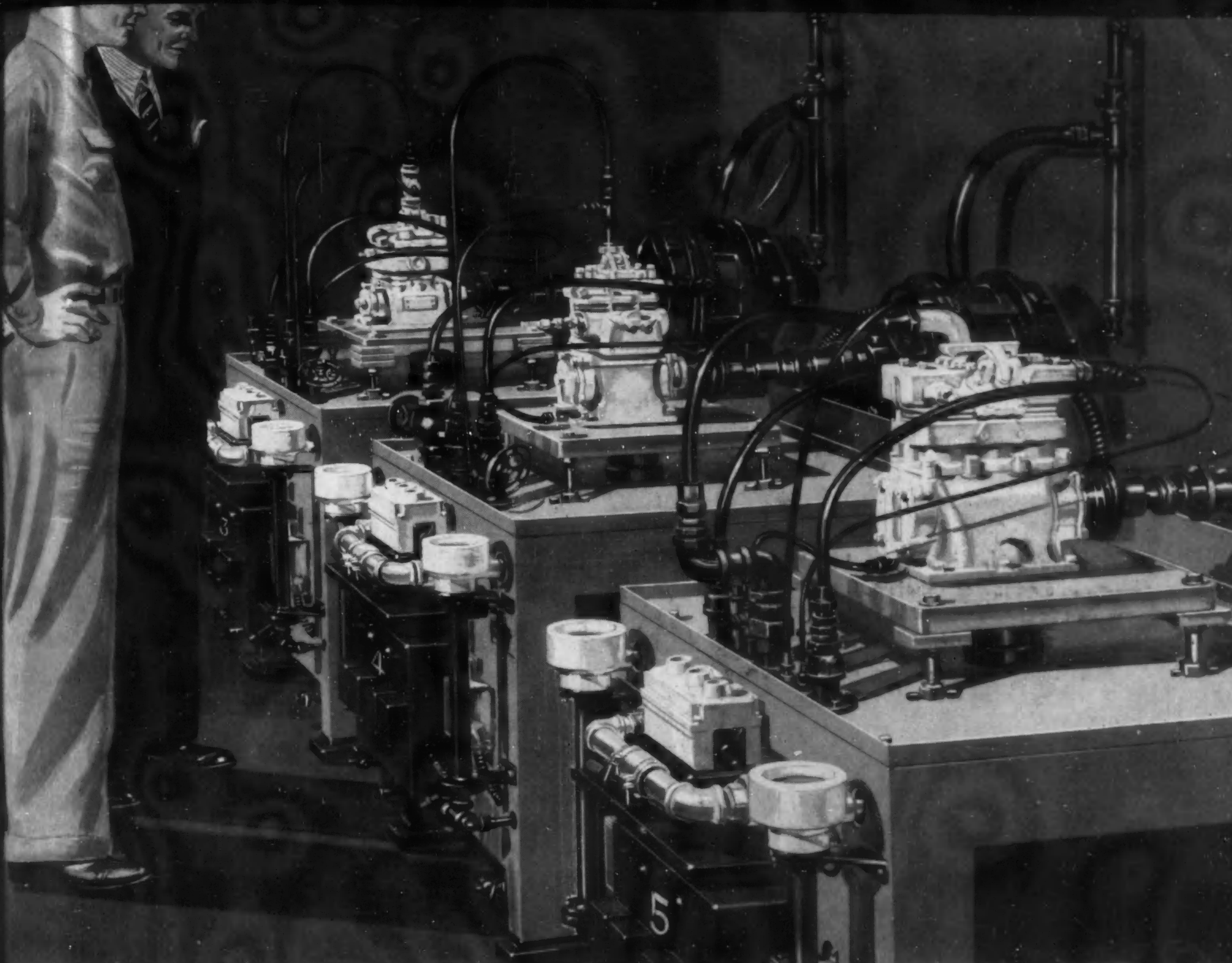

We're making History

Students of history can't keep pace with the change in boundaries or the industrial progress being made today in behalf of National Defense. This program will be recorded in the history books of tomorrow as an unprecedented achievement.

For our part, we're glad for the opportunity to contribute the experience we have gained from years of leadership in the production of electric furnace steel. This experience and our rapidly increasing facilities are being devoted to National Defense needs.

THE TIMKEN ROLLER BEARING COMPANY, CANTON, OHIO
Steel and Tube Division